Process Engineering Equipment Handbook

Claire Soares

McGraw-Hill

New York Chicago San Francisco Lisbon London Madrid
Mexico City Milan New Delhi San Juan Seoul
Singapore Sydney Toronto

Library of Congress Cataloging-in-Publication Data

Soares, Claire.
Process engineering equipment handbook / Claire Soares.
p. cm.
Includes index.
ISBN 0-07-059614-X (acid-free paper)
1. Chemical plants—Equipment and supplies. I. Title.

TP157.S658 2002
660′.283—dc21 2001045228

McGraw-Hill
A Division of The McGraw-Hill Companies



Copyright © 2002 by The McGraw-Hill Companies, Inc. All rights reserved. Printed in the United States of America. Except as permitted under the United States Copyright Act of 1976, no part of this publication may be reproduced or distributed in any form or by any means, or stored in a data base or retrieval system, without the prior written permission of the publisher.

1 2 3 4 5 6 7 8 9 0 CCW/CCW 0 7 6 5 4 3 2 1

ISBN 0-07-059614-X

The sponsoring editor for this book was Kenneth P. McCombs, the editing supervisor was Stephen M. Smith, and the production supervisor was Sherri Souffrance. It was set in New Century Schoolbook by Best-set Typesetter Ltd., Hong Kong.

Printed and bound by Courier Westford.

McGraw-Hill books are available at special quantity discounts to use as premiums and sales promotions, or for use in corporate training programs. For more information, please write to the Director of Special Sales, McGraw-Hill Professional, Two Penn Plaza, New York, NY 10121-2298. Or contact your local bookstore.

This book is printed on acid-free paper.

Information contained in this work has been obtained by The McGraw-Hill Companies, Inc. ("McGraw-Hill") from sources believed to be reliable. However, neither McGraw-Hill nor its authors guarantee the accuracy or completeness of any information published herein and neither McGraw-Hill nor its authors shall be responsible for any errors, omissions, or damages arising out of use of this information. This work is published with the understanding that McGraw-Hill and its authors are supplying information but are not attempting to render engineering or other professional services. If such services are required, the assistance of an appropriate professional should be sought.

Contributors

A

ASME Specifications

ASME: 345 East 47 Street, New York, NY 10017 USA

Abrasives

Abrasive Technology, Inc.: 8400 Green Meadows Drive, Westerville, OH 43081 USA
Sandusky-Chicago Abrasive Wheel Co., Inc.: 1100 W. Barker Avenue, Michigan City, IN 46360 USA
National Metal Abrasive, Inc.: P.O. Box 341, Wadsworth, OH 44282 USA

Acid Rain

Environment Canada: Environment Canada Enquiry Centre, Ottawa, Ontario K1A 0H3 Canada

Acoustic Enclosures, Turbine

Altair Filters International Limited: Omega Park, Alton, Hampshire GU34 2QE England

Actuators

J.M. Voith GmbH: P.O. Box 1940, D-89509 Heidenheim, Germany
Voith Turbo GmbH & Co., KG: P.O. Box 1555, D-74555 Crailsheim, Germany

Air Filtration

Altair Filters International Limited: see above

Air Pollution Control

Alstom (formerly ABB Power Generation): Finspong 61282 Sweden; Hasselstrasse 16, CH-5401 Baden, Switzerland; 5309 Commonwealth Center Parkway, Midlothian, VA 23112 USA

B

Bearings

Demag Delaval: 840 Nottingham Way, Trenton, NJ 08638 USA
Revolve Magnetic Bearings, Inc.: Calgary, Alberta, Canada

Boilers

Environment Canada: see above

Brakes

J.M. Voith GmbH: see above

C

CFD (Computational Fluid Dynamics)

Fluent Inc.: Lebanon, NH USA

Carbon; Carbon-Graphite Mix Products

Advance Carbon Products: 2036 National Avenue, Hayward, CA 94545 USA

Carbon Dioxide (CO_2); CO_2 Disposal

Vatenfall: S162 87 Stockholm, Sweden

Cement; Portland Cement

Environment Canada: see above

Centrifuges

Dorr-Oliver Inc.: 612 Wheelers Farm Road, Milford, CT 06460 USA

Chemical Complex; (Petro)Chemical Complex; Chemical Plant

Petrochemcial Company of Singapore: Singapore

Chemicals

ARCO Chemical Company: 3801 West Chester Avenue, Newton Square, PA 19073-2387 USA

Chemicals (Toxic), Handling

ARCO Chemical Company: see above

Chillers; Crystallizers; Chemical Separation Method; Alternative to Distillation/Fractional Distillation

Armstrong Engineering Associates: P.O. Box 566M, West Chester, PA 19381-0566 USA

Cogeneration

Alstom: Finspong 61282 Sweden

Compressors

Sulzer-Burckhardt: Winterthur, Switzerland
Demag Delaval: see above
Aerzener Maschinenfabrik, GmbH: D2358 Aerzen, Germany

Condensers

Alstom: Hasselstrasse 16, CH-5401 Baden, Switzerland; D6800 Mannheim 1, Germany

Condition Monitoring (CM); Condition-Monitoring System(s) (CMS); Engine Condition Monitoring (ECM); Engine Condition–Monitoring System(s) (ECMS)

Claire Soares Inc.: P.O. Box 540213, Dallas, TX 75354 USA

Control Systems; Controls

Sulzer-Burckhardt: see above
Voith Turbo GmbH & Co., KG: P.O. Box 1555, D-74555 Crailsheim, Germany
Voith Safeset A.B.: Ronningev. 6, S-82434 Hudliksvall, Sweden

Controls, Retrofit

Petrotech Inc.: 108 Jarrel Drive, P.O. Box 503, Belle Chase, LA 70037 USA
Demag Delaval: see above
J.M. Voith GmbH: see above

Conveyors

Sandvik Process System, Inc.: USA
Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Coolant; Engine Coolant

ARCO Chemical Company: see above

Cooling; Cool, Products That (Air Conditioners); Liquid-Cooled Air Conditioners

Thermoelectric Cooling America Corporation (TECA): USA

Cooling Towers

The Marley Cooling Tower Company: Marley and Lone Elm Roads, Olathe, KS 66061 USA

D

Drying

ARCO Chemical Company: see above

E

Ecological Parks; Industrial Ecological Parks

Environment Canada: see above

Ecosystem

Environment Canada: see above

Electric Motors; Electric Motor Controls

Reliance Electric Company: Cleveland, OH USA

Emissions; Air Emissions

Environment Canada: see above

Engines, Gas

Cooper-Bessemer Reciprocating: Grove City, PA USA

Environmental Accountability

Kodak: USA
Cultor: Finland

Environmental Economics

AssiDomän: Sweden

Exhausters, Centrifugal Gas

Ansaldo: Milan, Italy

Expansion Joints

Townson Expansion Joints: United Kingdom

Explosion; Explosion Hazard Analysis; Explosion Hazards

Eutech Engineering Solutions Ltd.: Billingham, Cleveland TS23 4YS England
H.M. Principal Specialist Inspector, Health & Safety Executive, Quay House, Quay Street, Manchester M3 3JB England

F

Fans, Centrifugal

Ansaldo: see above
Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Filters

Peerless Manufacturing Company: 2819 Walnut Hill Lane, Dallas, TX 75229 USA

Forest Products

AssiDomän: see above

Fuel Gas Conditioning System(s)

Peerless Manufacturing Company: see above

Fuel Systems; Fuel Flow Control

J.M. Voith GmbH: see above
Whittaker Controls, Inc.: 12838 Saticoy Street, North Hollywood, CA 91605 USA

Fuels, Alternative; Fuels, Gas Turbine

Bechtel Power Corporation: Gaithersburg, MD 20878 USA

G

Generators; Turbogenerators

Alstom: see above

H

Heat Exchangers

Armstrong Engineering Associates: see above

Heat Pumps; Heat Pumps, Geothermal; Heating Systems with a Renewable Energy Source

Enertran: Canada

L

Life-Cycle Assessment (LCA) (of Turbomachinery)

Claire Soares Inc.: P.O. Box 540213, Dallas, TX 75354 USA

Liquid Natural Gas (LNG)

Peerless Manufacturing Company: see above

Lubrication

Demag Delaval: see above

M

Measurement

Demag Delaval: see above

Metallurgy; Metallurgical Repair; Metallurgical Refurbishment

Liburdi Engineering: Hwy. 400, Dundas, Ontario, Canada

O

Oil Sands; Synthetic Crude; Tar Sands; Shale

Syncrude Canada Limited: Ft. McMurray, Alberta, Canada

Oxygen Analysis

Rosemount Analytical: Orville, OH USA

Ozone

Environment Canada: see above

P

Pollutants, Chemical; Pollutants, (from) Chemical Processes; Pollutant Indicators; Pollutants, Toxic; Pollutants, Toxic Chemicals

Environment Canada: see above

Power Transmission

Demag Delaval: see above
MAAG Gear Company: Switzerland
J.M. Voith GmbH: see above

Pulp and Paper

AssiDomän: see above

Pulsation Dampeners

Peerless Manufacturing Company: see above

Pumps

Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.
Demag Delaval: see above
Sulzer Pumps: USA
Goulds Pumps: USA

R

Refineries, Petroleum

Environment Canada: see above

S

Seals; Gas Seals

Revolve Magnetic Bearing, Inc.: see above

Separators

Peerless Manufacturing Company: see above

Stacks

Altair Filters International Limited: see above

T

Tanks

A.O. Smith Engineered Storage Products Company: 2101 S. 21st Street, Parsons, KS 67357 USA
Enraf: England

Turbines, Gas

Alstom: see above
ASME: see above

Turbines, Steam

Demag Delaval: see above
Peerless Manufacturing Company: see above

Turbochargers

Demag Delaval: see above

Turboexpanders

Demag Delaval: see above

U

Ultrasonic Cleaning

Sonics: USA

V

Vaporizers; Vaporizor Applications

Armstrong Engineering Associates: see above

W

Waste Management

Environment Canada: see above

About the Author

A registered professional engineer in Texas and Alberta, Canada, Claire Soares graduated with a B.Sc.Eng. in 1972 and an M.B.A. in 1993. Her career began in computational fluid dynamics working for Brian Spalding in Imperial College, London, on the COBALT project. She then spent about two years working on developing structural patents for the marine and the power distribution industries in England and Canada. Her rotating machinery career began in earnest at the oil sands Syncrude site in Fort McMurray, Alberta, in 1975. Four years later, she moved to Esso Resources and conventional oil and gas production until the "oil patch sat on its tail" at the end of 1981. She then accepted a three-year commission with the Canadian Air Force as Propulsion Systems Manager for all transportation engines in the Department of Defence Transport Command in Canada. She took charge of six helicopter engine fleets, as well as projects related to selecting replacements for two of those fleets. After that, she moved to the United States to start work as a senior engineer for Ryder Airline Services Division (ASD was also called Aviall and, before that, CooperAirmotive). At that point ASD was the largest independent overhaul facility for airline engines in the world, with a shop capacity of about 1000 engines a year. She ran technical support on second shift for 250 mechanics and their supervisors on the JT8D, JT3D, and CFM 56 engine lines. Three years later, she was made manager of the V2500 engine repair program, at that time the first and only designated facility for this engine in the United States. Two years later, after the engine line was up and running, she left to become an independent consultant, trainer, and writer. She has now lived and worked on four continents. Ms. Soares organizes one to two conference sessions annually for the International Gas Turbine Division of the American Society of Mechanical Engineers (ASME) and has done so since 1985. In May 2001 she was appointed to Fellow grade by ASME.

Process Engineering Equipment Handbook is her fourth book. The first, *Process Plant Machinery*, Second Edition, was coauthored with Heinz Bloch, P.E. This work and *Environmental Technology and Economics: Sustainable Development in Industry* helped provide the present handbook with its broad perspective. *Turboexpanders and Process Applications*, her third book, also coauthored with Heinz Bloch, was released in 2001. All her books are used when appropriate as course instruction material for her own and others' courses. Ms. Soares writes extensively for technical journals, such as *Petroleum Economist*, *Asian Electricity*, and *International Power Generation.*

She also writes for more general audiences, with some television screenplays and articles for various international newspapers and magazines to her credit. A published photographer, she writes poetry and has staged multimedia performances of her work for organizations such as the city of Dallas. She is a scuba diver and licensed commercial pilot, and also enjoys swimming and hiking.

Preface

If you picked up this book you are probably one of those lucky people who run plants. Either a thinly spread engineer (branch of specialty is irrelevant), a newly promoted technical manager, or a harassed technologist or senior mechanic, who just was told: "See that plant out there? You're in charge of making it work!" Even if you've been in plants for years, that's enough to make your innards rumble. If you have just stepped out of school, into your first plant, or into a totally different plant from the previous one you were at, your reaction might be more severe, especially if you belong to one of the numerous organizations with no budget for rotating machinery specialists (who look after what moves a process through its paces) or environmental specialists (who make certain you don't get fined or jailed, good intentions notwithstanding, as you run your plant). At this point, I should explain where I fit in with your agenda.

Twenty-some years ago, some heated arguments on the subject of how much I wanted to be a rotating machinery specialist took place in Canada's wild and woolly north. I was fencing with my boss, a process engineer, who was recommending I join his field. It was what my company needed, he asserted. I thought it needed both of us doing what we loved best. My career bears witness to the fact that I won the match, in the short- and long-term.

Time since has brought a few things forcibly home to me. To start with, the more I dealt with plant machinery in any form, the more I accepted that process conditions could affect the performance of that machinery at least as much as actual mechanical characteristics. In operations, repair and overhaul, or retrofit design and reengineering, what keeps people like me a step ahead of the manufacturer's field service representatives is knowledge of the process and familiarity with the controls that govern the entire system. In turn, the process engineer who gets handed a plant to run must acquire some basic knowledge of my bread and butter, the machinery that makes everything move up, down, or around. In large facilities, such as the ones I was fortunate enough to spend time in, there generally are in-house rotating machinery specialists. Often, though, the process engineer is not that lucky and gets everything—process components, machinery, controls, and all.

Life handed me an education (after formal degree acquisition) in rotating machinery specialization and the environmental technology that goes with it (yes, we machinery "cranks" run the stuff that turns out arguably 80 percent of the gunk in the universe). While doing this, I worked with scores of process engineers, control engineers, and various other specialists on a variety of projects that were among the most high profile in the world in their own right. It was "arranging to be in the right place—an operating plant—to get the best education in the best school in the world."

After all, curriculum, undergraduate or otherwise, is not necessarily any comfort. In my day they rarely taught this stuff to process, chemical, or mechanical engineers

at universities. They still don't. That leaves all the young engineers in the same boat—without any practical guide for reference.

My editor at McGraw-Hill was keen that they should have one, and one that was easy to read. We soon found we were on the same page on the subject of readability. We do not like technical material that sounds more intellectual than it absolutely has to, and we do like many diagrams, photographs, tables, and figures.

I add two other ingredients to my books and courses: (1) information on items (such as condition monitoring and specialized controls) that will help the engineer optimize cost-effective operations and (2) information that will help the engineer stay out of trouble with legislators, particularly environmental legislators (regardless of whether the legislation is current or impending). Fines levied for ignoring emissions or pollutant statutes may not be high enough to be a deterrent in themselves when weighed against a process plant's gross production revenue. They can, however, whittle away at profits while adding to overall costs per operating hour. Frequently, though, environmental equipment can actually result in machinery's attaining longer times between overhauls. Also, the loss of goodwill —that priceless commodity on annual reports—is immense if environmental standards are not followed.

In this competitive age, plants do vie for national, state, or provincial quality control awards. Clever managers can turn those into longer customer lists. Attaining these awards is not something that many accountants, lawyers, and MBAs, who run major corporations but may have little or no technical exposure, can pull off without their engineers. It is the engineers who are likely to be the key figures in putting together the action framework for what will buy their firm new or continued goodwill. Environmental accounting plans, holistic management of resources and waste products, environmental policy, waste and toxic management—they mean pretty much the same thing and they are not a feature at all in many other process engineers' reference books.

It is painfully evident that the emphasis given to waste and toxic management varies globally. It reaches a high in Sweden and Norway, England is fast developing an aggressive proactivity in this vein, and Canada has excellent technology, which may or may not get enforced to the appropriate extent depending on the political balance of power at any time. The United States has some large loopholes that are surprising for a country so advanced; shared emissions legislation is one. And yet, it's in the area of waste and toxic management that companies receive the most vocal and widespread media coverage (and loss of business) when exposed. Some of the world's youth appear to have a sense of resources running low and therefore a need to conserve them. In these days of increasing international joint ventures, the gaps between all these preferences is fast diminishing and the stable point for the resultant system can tend to reflect the highest standards among the partners.

One could argue that subjects that infringe on environmental and waste management turf belong in another handbook and with another kind of engineer. That is not entirely true though this is becoming a specialist field. The reasons for this statement are rooted in profit margins. If environmental considerations and waste products can be integrated into production in a way that what might have been a hazard or waste now contributes to revenue, this is obviously preferable to that hazard or waste being isolated with its own disposal/neutralization system that does not contribute revenue. Some examples are biomass waste in pulp and paper production, formerly disposed of, that can be converted to gaseous fuel for a turbine (*see* Pulp and Paper) and chemical by-products in complex downstream petrochemical plastics production, otherwise waste, that can now also be used as turbine fuel. The controls and system modifications that assist incorporation of these profitable adaptions into process plants are given some space.

I have also included some basic information on specific controls and monitoring systems. They are a fact of life on a process engineer's turf; the ones I have highlighted have a proven track record for adding profit margins to processes by minimizing downtime or fluctuations.

Similarly, a process engineer may have to make decisions related to turbomachinery performance or capacity that are affected by metallurgical processes. Included is some information on common critical alloys used in today's plants.

This book contains information on the major components and basic systems, including instrumentation and controls, and some optimization techniques that I wish I had had when I landed, albeit happily, in my first major plant. It also contains examples, drawn from knowledgeable sources, of action plans that have kept various process companies in good standing and high esteem with their public and governments worldwide. Selected extracts of the technology that are the bases of these policies are also included. These examples and technology extracts are frequently missing from engineering handbooks; I would be doing users of this handbook a disservice to leave out this information.

Increasingly process plants are becoming small power producers. Governments are now beginning to offer incentives to small power producers. The Thai government, which buys the excess power from Esso's Sriracha refinery, is just one such example. The Alberta, Canada, government buys excess power from Syncrude Canada Limited, which produces 170,000 barrels of crude oil a day. The British power network buys excess power from Elf Acquitaine's Flotta terminal, which collects North Sea petroleum products.

In other words, this book aims to provide a process engineer with:

- Knowledge of the basics the process engineer will meet up with
- Enough knowledge to help the process engineer optimize operation safety, efficiency, and profit margins
- Information about environmental systems and avoiding trouble with the law
- Tools to integrate the plant's operation with other services, such as power production and waste management, to further optimize profits and minimize losses due to interruptions in services provided by external companies

Claire Soares
claire_soares@compuserve.com

Introduction

The contemporary process engineer has to be an all-around generalist. This handbook contains basic information on items that cause or assist chemical reactions, such as chillers. In today's environment, information on additional sectors is also required to help the plant engineer function.

To begin with, besides the components in a plant that produce required chemical and physical state changes, such as fractionating distillation columns and reactors, the plant engineer needs to know the process plant machinery that transports and delivers raw material and products. It is this machinery that is very often the bane of the engineer's existence. The good news is that with a little knowledge one can keep most of it running.

When that happens, the process plant engineer may have to troubleshoot equipment if the plant does not have a rotating machinery engineer. That may be why so many of the process plant engineers I talked with asked me to include material on condition monitoring and life-cycle (of machinery components) assessment. These two items alone can save a plant a huge amount of its costs per plant operating hour, if properly utilized. I have included some of my notes from two of my basic courses on these subjects.

Interestingly enough, plant problems are common at system interfaces—at expansion joints, rather than at what they connect, and at gearboxes, couplings, and torque measurers, rather than the parent items of machinery they link. Also, certain accessories can be weak points if improperly applied. Air filtration can protect a machine from icing and erosive particles or it might build up excessive pressure drop and penalties on turbine output power. The quantity of information provided on these items, such as air filtration and power transmission equipment, reflects this fact.

Environmental technology and economics is another area now integral to a process engineer's world. Without this knowledge, the company could be fined and management imprisoned. Even if this is not an issue, environmental savvy can buy a corporation an inestimable amount of goodwill and a high profile in proactive community service. It has resulted in national quality control awards and other recognition. This may mean that corporation is preferentially selected as a product or services supplier. Environmentally sound cohabitation of industrial real estate in what are termed ecological industrial parks can also save utility costs.

Most importantly, however, taken over the life of a plant, environmentally sound methodology saves in overall operating costs per unit time. What seems like cleanliness and politically correct "extras" saves in parts longevity. This is not common knowledge in the engineering world. In fact, one common motivation for fitting environmental accessories is that operators are aware that if they wait until

use of same is forced upon them, retrofits of items such as flue gas desulphurization cost as much as 300 percent more than installation with initial construction.

Process plant engineers will probably also have to run their own in-house power plant at some point. They will have to manage interfaces with the power plant and their main process facility. The power is primarily for their own needs, but they frequently sell their excess power to the national grid. Even more interesting is the fact that process fluids that would otherwise be waste can sometimes be used as fuel. Examples include flue gas from a mining process used as a heat source and waste biomass from pulp and paper production gasified to use for steam production for running a steam turbine. In turn, waste heat from turbines is used for process purposes. Excess steam from the heat recovery steam generator in the independent power plant at Kuala Langat, Malaysia, is used in the owner's neighboring mill. Heat recovery schemes in Alberta, Canada, are being used to provide heat to vegetable greenhouses.

This cooperative effort between power generation and process technology is underlined by three major factors. First, the major oil companies, such as Shell, Amoco, and Esso, are now actively involved in major independent power production as a policy—the fuel they produce has a certain market as power production fuel. Second, the drive toward environmentally economic technology to dampen the strain on the world's sagging natural resources has played a major part in integrating the process engineer's world with that of power production. Last but not least, internal power production gives process engineers a much better handle on avoiding the brownouts, fluctuations, and power cuts that an external utility supplier can cause. They can also have the flexibility to use products within the plant for "unusual" fuel when they work with manufacturers who will accommodate their requirements. The contemporary competitive business environment is making it much easier for an engineer to negotiate this adaptability from a manufacturer.

With this complex, mobile, and fascinating framework, know that one could never fit representative information on everything a process engineer might need in one volume. The compromise I have struck is to cover the basics succinctly and spend effort on items that the process engineers I talk to and work with have very little information on, but ask about all the time. Not surprisingly, these items can cost the most in terms of cost per operating hour. Costs include lost production time, machinery problems, troubleshooting efforts, useless expense in overdesigned machinery condition monitoring systems, premature parts replacement, and power supply problems.

Then there was the matter of highly specialized process sectors, such as pulp and paper, agriculture, and food processing. This book hits the highlights with these industries and others, and a specifications and standards section is provided for the process engineer to use in conjunction with this book. All the common items in any process industry, such as pumps, motors, couplings, controls, and so forth, are in this book. As terminology for an item varies so much between industries, it is a good idea to read through the Contents and the Index a few times to get the most out of this book.

Last, but not least, although this book might be more for process engineers in plant operations, I have included some design information where I felt it would assist logic. Design and specification work gains from operations and maintenance exposure.

Contents

Preface xi
Contributors xv
Introduction xxi

A A-1

ASME Specifications A-1
Abrasives A-1
Accident Management A-9
Accountability, Environmental (*see* Environmental Accountability)
Acid Rain A-9
Acoustic Enclosures, Turbine A-10
Actuators A-37
Aerfoils; Airfoils (*see* Metallurgy; Turbines)
Agitators A-38
Agriculture; Agricultural Product Processing A-48
Air Filtration; Air Inlet Filtration for Gas Turbines A-49
Air Pollution Control A-83
Air Purification; Air Sterilization A-101

B B-1

Balancing; Onspeed Balancing of a Rotor B-1
Balancing Problems, Troubleshooting (Turbomachinery) (*see* Condition Monitoring)
Batteries (*see* Cells)
Bearings B-1
Blades and Vanes (for a Turbine) (*see* Metallurgy; Turbines)
Boilers B-11
Borescopes B-11
Brakes B-11
Brick-Lined Process Equipment B-23
Briquette Machines B-28

C C-1

CAD/CAM C-1
CFD (Computational Fluid Dynamics) C-1
Carbon; Carbon-Graphite Mix Products C-7
Carbon Dioxide (CO_2); CO_2 Disposal C-13
Castings (*see* Metallurgy)
Cells C-22
Cement; Portland Cement C-22
Centrifuges C-34
Ceramics C-38

Chemical Cleaning C-38
Chemical Complex; (Petro)Chemical Complex; Chemical Plant C-38
Chemicals C-41
Chemicals (Toxic), Handling C-42
Chillers; Crystallizers; Chemical Separation Method; Alternative to Distillation/ Fractional Distillation C-62
Chimneys (*see* Stacks)
Cleaning C-78
Clutches (*see* Power Transmission)
Coatings (*see* Ceramics; Metallurgy)
Cogeneration C-78
Coker (*see* Oil Sands)
Color Coding C-81
Columns (*see* Towers and Columns)
Combustor(s); Low NO_x Combustor (*see* Turbines)
Compressors C-81
Condensers C-233
Condition Monitoring (CM); Condition-Monitoring System(s) (CMS); Engine Condition Monitoring (ECM); Engine Condition–Monitoring System(s) (ECMS) C-253
Control Regulators (*see* Actuators)
Control Systems; Controls C-331
Controls, Retrofit C-354
Conversion Tables (*see* Some Commonly Used Specifications, Codes, Standards, and Texts)
Conveyors C-408
Coolant; Engine Coolant C-408
Coolers, Dairy C-412
Cooling; Cool, Products That (Air Conditioners); Liquid-Cooled Air Conditioners C-413
Cooling Towers C-423
Corrosion; Anticorrosion Coatings C-426
Couplings (*see* Power Transmission)
Crushers C-426

D D-1

Dampeners (*see* Pulsation Dampeners)
Desalination D-1
Dialysis; Electrodialysis D-2
Distillation; Fractional Distillation D-2
Diverter; Diverter Damper; Diverter Valve; Flapper Valve D-3
Doctor D-3
Drives D-3
Drum; Knock-Out Drum; Knock-Out Vessel (*see* Separators)
Drying D-4
Ducting; Ducting and Joints D-18

E E-1

ECM (Engine Condition Monitoring) (*see* Condition Monitoring)
ECMS (Engine Condition–Monitoring Systems) (*see* Condition Monitoring)
Ecological Parks; Industrial Ecological Parks E-1
Ecosystem E-1
Ejectors E-2
Electric Motors; Electric Motor Controls E-3
Emergency Power Generation (Packages) (*see* Engines, Gas)
Emissions; Air Emissions E-36
Engines, Gas E-38
Environmental Accountability E-40

Environmental Air Monitoring (*see* Emissions)
Environmental Economics **E-49**
Evaporative Coolers (*see* Chillers; Coolers, Dairy)
Exhaust Stacks (*see* Stacks)
Exhausters, Centrifugal Gas **E-56**
Expansion Joints **E-57**
Explosion; Explosion Hazard Analysis; Explosion Hazards **E-67**
Extraction, Liquid-Solid **E-87**

F **F-1**

Fans, Centrifugal **F-1**
Filters **F-5**
Flare Stacks **F-12**
Floating Covers **F-12**
Fluidized Bed **F-12**
Forest Products **F-12**
Fuel Gas Conditioning System(s) **F-24**
Fuel Systems; Fuel Flow Control **F-29**
Fuels, Alternative; Fuels, Gas Turbine **F-37**

G **G-1**

Gas Turbine Cleaning or Washing (*see* Turbines)
Gas Turbines (*see* Turbines)
Gearboxes (*see* Power Transmission)
Gears (*see* Power Transmission)
Generators; Turbogenerators **G-1**
Grinding (*see* Abrasives; Some Commonly Used Specifications, Codes, Standards, and Texts)
Grinding Wheels (*see* Abrasives)

H **H-1**

Hazards (*see* Color Coding; Explosion; Some Commonly Used Specifications, Codes, Standards, and Texts)
Heat Exchangers **H-1**
Heat Pumps; Heat Pumps, Geothermal; Heating Systems with a Renewable Energy Source **H-1**
Heat Treatment (*see* Metallurgy)
Heaters, Electric **H-6**
High-Speed Drive Systems (*see* Power Transmission)
Hot Isostatic Pressing (or HIPing) (*see* Metallurgy)
Hydraulic Filters (*see* Filters)

I **I-1**

Industrial Ecological Park (*see* Ecological Parks)
Industrial Ecology (*see* Ecological Parks)
Inlet Ducts and Silencers (*see* Acoustic Enclosures, Turbine; Air Filtration; Ducting)
Instrumentation (*see* Condition Monitoring; Measurement)
Insulation (*see* Some Commonly Used Specifications, Codes, Standards, and Texts)
Irradiation, Food Product **I-1**

L **L-1**

LCA (*see* Life-Cycle Assessment)
LNG (*see* Liquid Natural Gas)
Laser Cutting, Drilling, Machining, Welding (*see* Metallurgy)

Life-Cycle Assessment (LCA) (of Turbomachinery) L-1
Liquid Eliminators (*see* Separators)
Liquid Natural Gas (LNG) L-21
Lubrication L-24

M M-1

Magnetic Bearings (*see* Bearings)
Measurement M-1
Metallurgy; Metallurgical Repair; Metallurgical Refurbishment M-31
Metering, Fluids; Metering Pumps (*see* Fuel Systems)
Mist Eliminators (*see* Separators)
Mixers (*see* Agitators; Centrifuges)
Monitoring (*see* Condition Monitoring)
Motors (*see* Electric Motors)

N N-1

Noise and Noise Measurement (*see* Acoustic Enclosures, Turbine)
Noise Silencing and Abatement (*see* Acoustic Enclosures, Turbine)
Nondestructive Testing (FP1, MP1, X Ray) (*see* Metallurgy)
Nozzles N-1

O O-1

Oil Analysis O-1
Oil Sands; Synthetic Crude; Tar Sands; Shale O-1
Oxygen Analysis O-1
Ozone O-4

P P-1

Packaging P-1
Paper (*see* Pulp and Paper) P-1
Pipe (*see* Some Commonly Used Specifications, Codes, Standards, and Texts) P-1
Plastics P-1
Pollutants, Chemical; Pollutants, (from) Chemical Processes; Pollutant Indicators; Pollutants, Toxic; Pollutants, Toxic Chemicals P-7
Portland Cement (*see* Cement) P-72
Power Production; Power Production In-House; IPP; SPP P-72
Power Transmission P-73
Pulp and Paper P-204
Pulsation Dampeners P-214
Pumps P-214

R R-1

Reactors; Chemical Reactors R-1
Refineries, Petroleum R-1
Refrigerant(s) R-10
Regenerator R-19

S S-1

Seals; Gas Seals S-1
Separators S-10
Snubbers (*see* Pulsation Dampeners)

Stacks S-29
Steam Generator and Steam Supply S-38

T T-1

Tanks T-1
Temperature and Pressure Sensors (*see* Measurement)
Thermal Insulation (*see* Some Commonly Used Specifications, Codes, Standards, and Texts)
Thin-Film Processors (*see* Chillers)
Torque Converters, Measurements, and Meters (*see* Power Transmission)
Towers and Columns T-40
Toxic Substances (*see* Pollutants, Chemical)
Transportation, of Bulk Chemicals, of Large Process Equipment T-40
Triple Redundancy T-41
Turbines, Gas T-43
Turbines, Steam T-76
Turbochargers T-112
Turboexpanders T-114

U U-1

Ultrasonic Cleaning U-1

V V-1

Valves (*see* Control Systems)
Vanes (*see* Metallurgy)
Vaporizers; Vaporizer Applications V-1
Vents and Flame Arrestors V-31
Vibration Measurement (*see* Condition Monitoring)

W W-1

Waste Heat Recovery (*see* Cogeneration)
Waste Management W-1
Welding (*see* Metallurgy)

Some Commonly Used Specifications, Codes, Standards, Texts Sp-1
Index In-1

A

ASME Specifications

A list of specifications available from the American Society of Mechanical Engineers is available in the section Some Commonly Used Specifications, Codes, Standards, and Texts at the end of the book before the index. This list provides additional detail on items that may not be covered in great depth here.

Abrasives

Abrasives is a term given to various materials with different physical formats (such as aggregate, grains, shot, particles bonded with an adhesive, and so forth) that are used to wear down surfaces to desired dimensions or surface finishes or for some other purpose. They may be used in their "raw" state, such as with shot or glass in shot or glass-bead peening operations. Or they may be used in conjunction with adhesives and fillers to make belts, wheels, and tool surfaces.

Sometimes peening operations address more than surface finish. Glass-bead peening has been used to add a compressive stress layer to the surface of gas-turbine compressor wheels to bring the net tensile stress level down to tolerable levels. As alloy metallurgy improved, machinery component operation totally under the stress endurance curve was possible, and such applications were phased out. They are worth mentioning, however, as they can contribute to puzzling failure modes if they are wrongly applied during design or repair and overhaul.

Abrasives may also be combined with adhesives to make grinding wheels, belts, or other components for precision grinding in sophisticated manufacturing or repair machinery such as a blade-tip grinder. A *blade-tip grinder* is basically a combination of a grinder and a computer or some form of computer numerical control (CNC) that is used to grind airfoil blade tips to arrive at precise turbine-wheel-assembly dimensions.

Product improvements now have produced superabrasives for high-precision processes that are generally used in conjunction with CNC and computer-aided design/computer-aided manufacturing (CAD/CAM) programming and equipment.

Different types of grinding materials allow larger depths of cut (decreased work time) with less metallurgical workpiece (heat soak) damage. This lowers overall production costs.

In the continued drive for improved environmental impact in production, some companies are using water coolants rather than oil coolants.

The applications of abrasives in process technology are too numerous to mention; however, some basic information on the common facets of abrasives in various states and process formats follows. Sources are as acknowledged.

CNC Applications*

Precision-plated grinding wheels are now replacing standard wheels (that use adhesive bonding with abrasive grit) in some applications. These are suitable for

* Source: Abrasive Technology, Inc., USA. Adapted with permission.

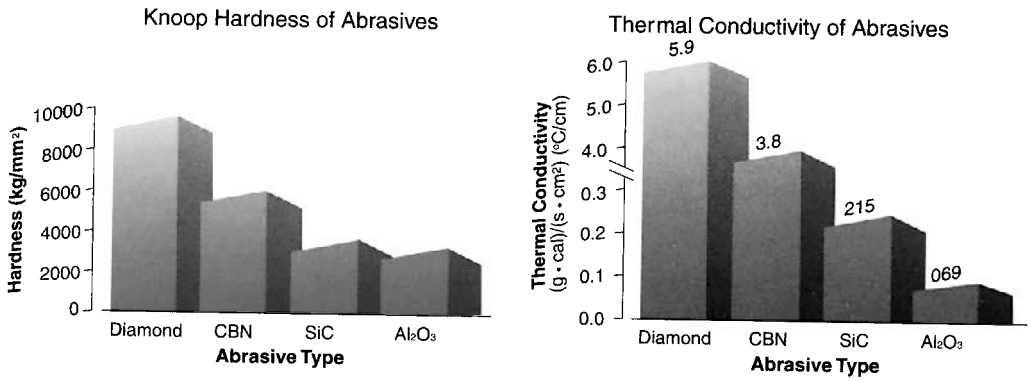


FIG. A-1 Physical characteristics of specific abrasives. (Source: Abrasive Technology, Inc.)



FIG. A-2 CAD/CAM workstation. (Source: Abrasive Technology, Inc.)

use on nickel- and cobalt-based superalloys, engineered ceramics, and standard ferrous alloys. (See Figs. A-1 to A-3.) Advantages of these wheels include:

- The ability to manufacture tight-tolerance complex forms.
- Lower initial cost, compared to vitrified and metal bonded superabrasive wheels.
- Freer cutting, resulting in higher material removal rates, less power, and reduced thermal damage to the workpiece.
- The ability to hold form or profile from first to last cut.
- Reduction or elimination of time associated with dressing, setups, and wheel changes.
- Safe, high-speed operation due to a steel core.
- The ability to strip and replate the core.

Precision techniques are required to measure the finished products, including CNC coordinate measuring machine (CMM) capability with associated CAD/CAM features, laser micrometers, form-scan geometry gauge, optical comparators, and other digital gauges.

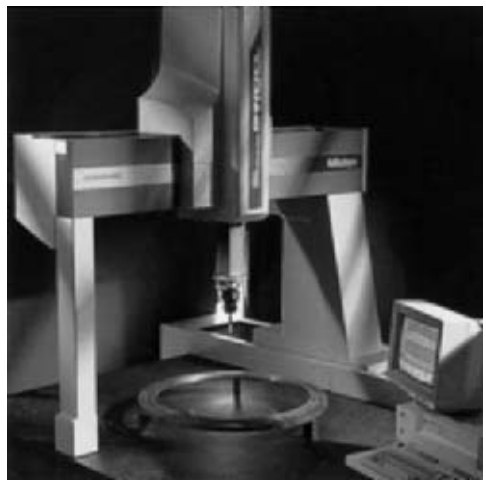


FIG. A-3 Coordinate measuring machine (CMM). (Source: Abrasive Technology, Inc.)

Standard Grinding Wheels*

Technical data standard on abrasives

Wheel symbols and markings. All responsible grinding-wheel manufacturers use a marking system established by the American Standards Association. This marking system describes the makeup of an abrasive wheel in a manner somewhat similar to the way a chemical formula describes the nature of a chemical compound. It indicates by a standard system of letters and numbers the important elements used in the manufacture of the wheel and, to a degree, the amounts and manner of their combination.

Any change in the marking represents a change in wheel characteristics and produces a corresponding change in the action of the wheel.

Familiarity with wheel markings, along with an understanding of the characteristics of the material to be ground, helps to determine the kind of work that a particular wheel is capable of grinding.

Elements of the marking system and their significance. (See Table A-1.) The first "station" specifies the particular type of abrasive material in the wheel. A complete line of abrasive grains suitable for grinding practically any material are available.

The chief difference in the types of abrasive involve their "toughness" and their "friability." The dictionary definition of "friable" is "easily crumbled or reduced to a powder"; certainly abrasives are not easily crumbled or reduced to a powder. The difference is one of degree and both tough and friable types have their applications.

YA	51A	Tough aluminum oxide for heavy-duty work. Best general purpose grain.
AA	52A	Semifriable. Frequently specified for precision grinding on average steels.
TA	53A	Practically pure aluminum oxide. White color, friable, and cool cutting.
	50A	Combination of semifriable and white friable grain.

* Source: Sandusky-Chicago Abrasive Wheel Co., Inc., USA.

TABLE A-1 Summary of Major Abrasive-Type Characteristics

			Sandusky and Chicago	TA 53A	60 60	K K	5 5	V V	E BH			
Sandusky and Chicago Abrasive Types			Grain Sizes	Grades	Structure			Bond Process		Sandusky and Chicago Bond Codes		
	ALUMINUM OXIDE		16	F	2			V—Vitrified			*VITRIFIED*	
					dense						Aluminum Oxide	
YA	Regular	51A	20	G	3						*High heat*	
AA	Semifriable	52A	24	H						R	Red	A
TA	Friable	53A	30	I	4 std.			B—Resin		E	Clear	BH
BB	Off white	–	36	J						E+	Blue	–
32A	Gray	55A	46	K	5					R+	Dark red	–
PA	Pink	12A	60	L	6			R—Rubber			*Low heat*	
					open							
RA	Ruby	RA	70	M	7					–	Red	V
AAT		54A	80	N	8					–	Clear	B
	Combinations											
–		50A	90	O							*Silicon Carbide*	
			100	P	9					A	Clear	C
	SILICON CARBIDE		120	Q	10 very						*RESINOID*	
											Aluminum Oxide	
C	Black	49C	150	R	11 porous					B	Brown	B1
												B2
GC	Green	49CG	180	S	12							BXF
			220	T							*Silicon Carbide*	
										B	Brown	B5
			320	U								B15
											REINFORCED RESIN	
										BF	Brown	BR
											RUBBER	
										HR	Hard	HR
										SR	Soft	SR

AAT	54A	Combination of tough and friable. Used on fairly heavy work but of precision character.
32A	55A	Gray color polycrystalline grain. Excellent for surface and tool and cutter grinders.
PA	12A	Light pink colored grain. Gives superior results on “problem” high alloy steels.
RA	RA	High chrome, ruby colored. Surface grinding and internal wheels.
WTC	WTC	Ceramic abrasive with polycrystalline microstructure. High stock removal rates and long life with vitrified or resin bonds.
ZA	ZA	Zirconium oxide abrasive. High stock removal on mild, stainless, and high alloy steels.
C	49C	Black silicon carbide. Standard for grinding most nonferrous and nonmetallic materials. Best material for use on ordinary cast irons.
GC	49CG	Green silicon carbide. A little more friable than regular black. Usually used in carbide tool grinding.
CA	CA	Combination of aluminum oxide and black silicon carbide and a resin bond. Used in plugs and cones when grinding a variety of materials including both steel and cast iron with the same grinding wheel.

The second "station" specifies the grit size. This represents the approximate number of openings per linear inch in the final screen used to size the grain. Obviously the larger the number, the finer the grain. For example, 60 grit size is one that will pass through a screen with 27 openings per inch and be held on a screen with 33 openings per inch. The most commonly used grit sizes fall in the range of 24 to 120.

The third "station" in the wheel marking is a letter that specifies the grade or relative holding power of the bond that holds the grains in place. With a given type of bond it is the amount used in the wheel that determines the wheel's grade or "hardness." When the amount of bond is increased, the size of the globules of bond (bond posts) connecting each abrasive grain to the adjacent ones is increased. This larger bond post is naturally stronger, thereby increasing the "hardness" of the wheel. The term "hardness" with respect to abrasive wheels has nothing to do with the hardness of the abrasive material itself, but rather with the relative shearing and impact forces necessary to dislodge a grain from the wheel. Both "soft" and "hard" wheels are necessary and have their specific uses.

The ideal grade for any particular grinding application is the one that will hold the grains in place until they become dull or glazed and then release them allowing new sharp grains to take their place. This may be any place on the scale between the very soft G, H, or I grades to the very hard S, T, or U grades depending upon the material being ground, the nature of the results desired, and the various conditions of speed, pressure, etc., coincident to the operation.

The fourth "station" in the marking system describes the "structure" (relative bulk density) of the wheel. When the abrasive grains are tightly pressed together and interlocked the wheel has a denser structure, which is indicated by a lower structure number, such as 3 or 4. When the wheel is rather loosely pressed and has a more porous nature (more free cutting but necessarily less durable), it has a higher structure number, such as 7 or 8.

The standard structure at which most small wheels are manufactured is 4. This standard density is supplied when the structure number is omitted from the wheel marking. Structure numbers higher (more open) than 8 require the addition of a burnout material to the grinding wheel mix. Open structure wheels are soft and should only be used when necessary on difficult to grind materials.

The fifth "station" in the marking system specifies the basic type of bond used in the wheel. These are:

V *Vitrified:* A glass or porcelain fired to a high temperature.
B *A synthetic resin:* Usually a phenolic thermosetting type. ("B" originates from the old term "Bakelite.")
R *Rubber:* Either natural or synthetic oil resistant type.
E *Shellac:* The natural material. Has very limited applications.

The sixth "station" is used to designate the particular bond used in the wheel. These are designed with various characteristics to give the resulting wheel certain physical properties desirable for the different applications.

Fundamentals of wheel selection. There are many factors that enter into the proper selection of a wheel. Probably the most important things to consider when choosing a wheel for the first time on any job are:

A. Material to be ground and its hardness.
B. Amount of stock to be removed.
C. Finish required.
D. Wheel speed or tool speed.

E. Area of grinding contact.
F. Whether grinding is done dry or with a coolant.
G. Severity of the grinding operation.
H. Type and condition of grinding machine being used.

A. The material being ground affects the selection of the type of *abrasive*, the *grit size*, and the *grade*.

Aluminum oxide is used to grind steel and steel alloys. If the material is heat sensitive or a "high-speed" steel, use friable grain. For difficult materials, such as tool steels high in vanadium, pink or ruby grain is suggested.

Silicon carbide is used to grind cast iron, nonferrous metals, and nonmetallic materials such as glass.

Very hard and brittle materials require relatively fine grain, soft or ductile materials a coarser grain.

Use soft grade wheels on hard materials. A harder grade may be used on more easily penetrated materials, which have a lesser dulling effect on the grain.

B. Amount of stock to be removed affects the choice of *grit size* and *bond*.

Use coarse grit for rapid stock removal and finer grits for harder materials. Use vitrified bond for fast cutting and resinoid or rubber for high-speed machines and higher finish.

C. Finish required influences selection of *bond* and *grain size.* Generally speaking a resin-bonded or rubber-bonded wheel gives a better surface finish but not as good dimensional accuracy as a vitrified wheel.

Grain size is a major factor in the surface finish but the method of dressing the wheel also plays an important part. Fine finishes can be obtained with relatively coarse wheels if the wheel is dressed carefully with a diamond and grinding infeed is properly adjusted. A poor finish can result even with a fine wheel if the grinding techniques used are wrong.

A general rule of thumb relating required RMS finish with grit size under average conditions is as follows.

RMS Finish	Grit Size
32	46
20–32	54
15–20	60
10–15	80

D. Wheel speed affects the choice of *bond*. Vitrified wheels are not to be run in excess of 6500 surface feet per minute except in very special cases. High speeds require resin or rubber bonds.

E. Area of grinding contact affects the choice of *grit size* and *grade*. Large contact area indicates coarse grit size and small contact area a finer size. Small contact area requires a harder grade, the smaller the harder.

F. Grinding wet or dry influences the choice of grade. If coolant is used, wheels one or two grades harder may be used without burning the work. Much depends on the amount and efficiency of coolant reaching the area of contact.

G. The severity of the grinding operation influences the choice of *abrasive type* and to some extent the wheel *grade*. A tough abrasive, regular aluminum oxide, would be used for grinding steel under severe conditions. A milder friable abrasive would be used for light grinding on hard or heat-sensitive steels. Intermediate

types, semifriable or a combination, would be used under conditions of average severity. Where rough conditions such as deburring or removing rough scale exist, it is usually necessary to use a harder grade and coarser grit than the nature of the material would call for under normal grinding conditions.

H. The type and condition of the machine being used will influence the choice of *bond* and *grade*. If the machine is of a portable type one may assume that the wheel will have rougher usage than if used on a fixed base machine. If, in addition, it is a high-speed machine, a resin-bonded wheel is mandatory. Generally speaking, portable machines require harder grade wheels than permanently fixed machines.

Peening; Shot Peening*

For illustrative purposes, some standard information on certain types of shot follows.

An example of standard product information

Standard abrasive meets Society of Automotive Engineers (SAE) requirements for size, chemical composition, hardness, microstructure, and physical properties.

Size: See previous chart.

Chemistry

Carbon	.85%–1.20%	
Manganese		
	S70–S110	.35%–1.20%
	S170	.50%–1.20%
	S-230 and larger	.60%–1.20%
Silicon		.40%–1.50%
Sulfur		.05% max
Phosphorus		.05% max

Hardness

S-40/48RC
M 48/55RC
HH 55/62RC
FH 63/68RC

Microstructure: Uniform martensite tempered to a degree consistent with the hardness range with fine, well-distributed carbides, if any.

Physical characteristics: Reference SAE J-827.

Specialty products

Precise media for the most exacting requirements, including roll etch grit for the steel industry, special cutting grit for the granite industry, and military specification shot (MIL-13165-C) for precision peening applications, and customized products to meet a customer's unique requirements.

Packaging

Drums: Nonreturnable 55-gallon steel drums palletized on expendable wooden pallets.

---

*Source: National Metal Abrasive, Inc., USA. Adapted with permission.

Shot 2000 lb net
Grit 1667 lb net sizes G-12 through G-40
1500 lb net sizes G-50 through G-120

Bags/cartons: 50 lb net, multiwall paper bags—40 bags per carton (2000 lb net) strapped to disposable wooden pallets.

Bulk: Custom bulk packaging available.

Typical material safety data sheet

SECTION I PRODUCT IDENTIFICATION

D-U-N-S No.: 14-421-8252

Product Names: Perma-Steel Shot, Perma-Steel Grit
Common Name: Cast Steel Shot, Cast Steel Grit
Chemical Family: Ferrous

SECTION II HAZARDOUS INGREDIENTS

CHEMICAL NAME	CAS NUMBER	%WEIGHT	OSHA PEL – TWA	ACGHI TLV – TWA
Iron	1309-37-1	>96	10 mg/m^3 as oxide fume	5 mg/m^3 as oxide fume
Manganese	7439-96-5	0.35–1.3	C* 5 mg/m^3 as dust	C* 5 mg/m^3 as dust
			C* 5 mg/m^3 as fume	C* 1 mg/m^3 as fume
Carbon	1333-86-4	0.8–1.3	None established	None established
Silicon	7440-21-3	0.4–1.2	15 mg/m^3 (as nuisance dust)	10 mg/m^3

SECTION III PHYSICAL DATA

Melting Point: 1371–1482°C
Evaporation Rate: Not Applicable
Boiling Point: 2850–3150°C
Specific Gravity (at 60°F): >7.6
Solubility in Water: Not Applicable
Vapor Pressure: Not Applicable
Vapor Density: Not Applicable
Percent Solid by Weight: 100%
pH: Not Applicable
Appearance and Odor: Shot is near spherical. Grit is angular. Both are gray to blue in color with slight metallic odor.

SECTION IV FIRE AND EXPLOSION HAZARD DATA

Flash Point: Not Applicable
Flammability Limits: Not Applicable
Autoignition Temperature (solid iron exposed to oxygen): 930°C
Cast Steel Shot and Grit will not burn or explode. The solid form of material is not combustible. Fire and explosion hazards are moderate when material is in the form of dust and exposed to heat or flames, chemical reaction, or contact with powerful oxidizer.
Fire Extinguishing Method: Use dry chemicals or sand to exclude air.

---

* C means ceiling limit, limits that shall not be exceeded, even for a short time.

Accident Management

Accidents are generally categorized according to levels of severity. These levels would be as follows or subdivisions thereof, depending on the nature of the process. The accident may be one that:

1. Results in death for nonplant personnel (public fatality).
2. Kills one or more plant personnel (major accident).
3. Kills the operator (plant fatality).
4. Injures personnel (time loss) and/or damages property (asset loss).
5. Is prevented by operator's action (near miss).
6. Is prevented by policy/legal area restrictions and personnel procedures and/or manually activated equipment.
7. Is prevented by automatic systems/equipment. The systems may provide passive (monitoring) capability or active real-time control.

When a process engineer provides information to help determine a plant's insurance premiums, he or she will find that those premiums decrease as the accident level goes from 1 to 7 in the previous list. Planning to avoid accidents is thus a major function of the process engineer's duties. Any manual actions/procedures, such as training on plant simulators, required to help avoid accidents should also be practiced or drilled until they become easy to follow when actually required.

To help avoid the possibility of human error, automation in plant-safety systems is now an area of practicing triple redundancy. Triple redundancy was formerly observed only with control of high-performance military aircraft systems. Now, increasingly, it pays off with land-based control systems that are safety, operational, or control in scope. *See* Triple Redundancy.

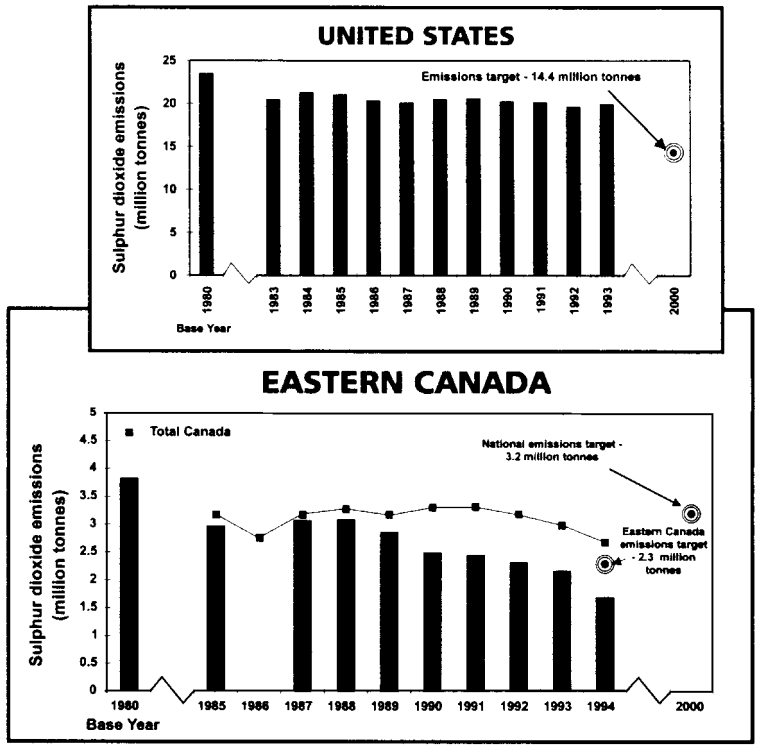
Accountability, Environmental (*see* Environmental Accountability)

Acid Rain

Acid rain is precipitation polluted by acidification with atmospheric pollutants. These pollutants include emissions of oxides of nitrogen (NO_x), oxides of sulfur (SO_x), and hydrogen chloride radicals. Various strengths of nitric acid, sulfuric acid, and hydrochloric acid result. Key indicators of acid rain include emission levels of NO_x and SO_x, wet sulfate deposits, and trends in acidity in lakes and other freshwater bodies. An increase in emissions that increases the level of any of these indicators will bring environmental, regulatory, and potentially public and special-interest group pressures to bear on a plant. Sample measurements of these indicators on a national or provincial scale are illustrated in Figs. A-4 to A-7.

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.



Notes:
(a) Eastern Canadian emissions data for 1986 are unavailable.
(b) U.S. emission estimates converted to metric tons (1 ton = 0.91 metric tons).

Canadian sources:
Eastern Canadian emissions: Environment Canada. *Annual Report on the Federal-Provincial Agreements for the Eastern Canada Acid Rain Program*, 1990 to 1994.

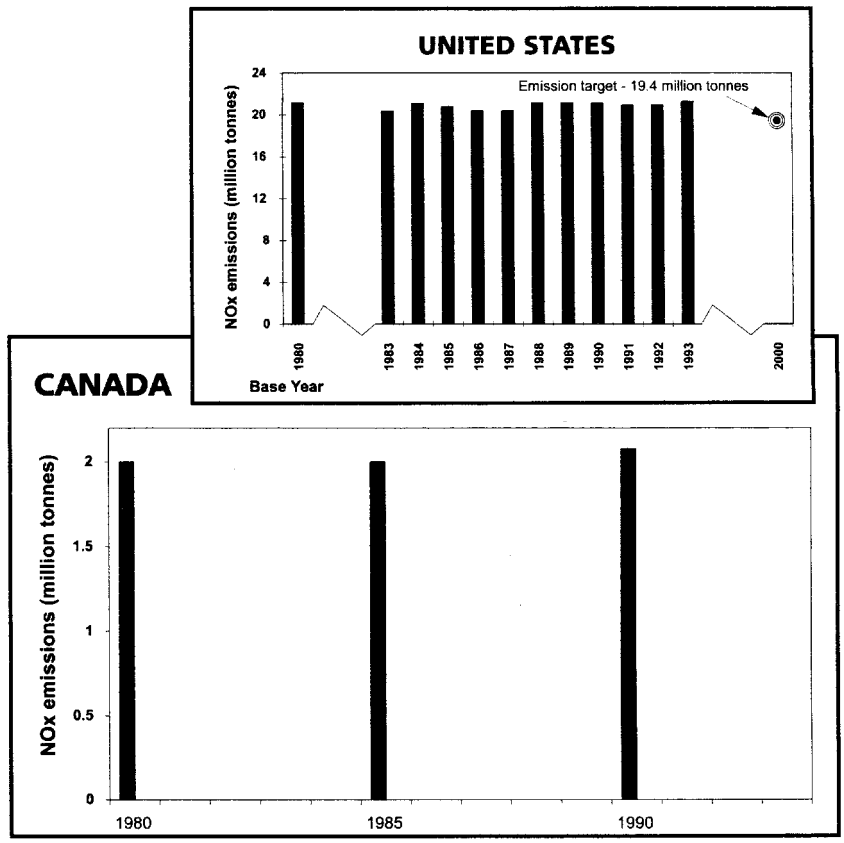
Total Canadian emissions: Pollution Data Branch, Environmental Protection Service, Environment Canada.

U.S. source:
U.S. EPA. 1994. *National Air Pollutant Emission Trends, 1990–1993*. Office of Air Quality Planning and Standards, Research Triangle Park, North Carolina.

FIG. A-4 United States and eastern Canada SO_x emission trends. (Source: Environment Canada SOE 96-2, Spring 1996.)

Acoustic Enclosures, Turbine

Enclosures around noisy rotating machinery, particularly gas turbines, provide protection against noise and help contain risk from situations such as small gas leaks. The design of acoustic gas turbine enclosures is summarized in this section in text extracts from two papers on the subject. These extracts give the reader all the salient points to watch for if specifying or buying an acoustic enclosure. It also provides the reader with basic knowledge of how acoustic sound intensity measurements are conducted generally (i.e., for any kind of equipment). Note also that these designs were built for offshore applications where weight has to be minimized.



Notes:
(a) U.S. emission estimates converted to metric tons (1 ton = 0.91 metric tons).
(b) Canadian emission estimates are preliminary.

Canadian source:
Pollution Data Branch, Environmental Protection Service, Environment Canada.

U.S. source:
U.S. EPA. 1994. *National Air Pollutant Emission Trends, 1990–1993.* Office of Air Quality Planning and Standards, Research Triangle, North Carolina.

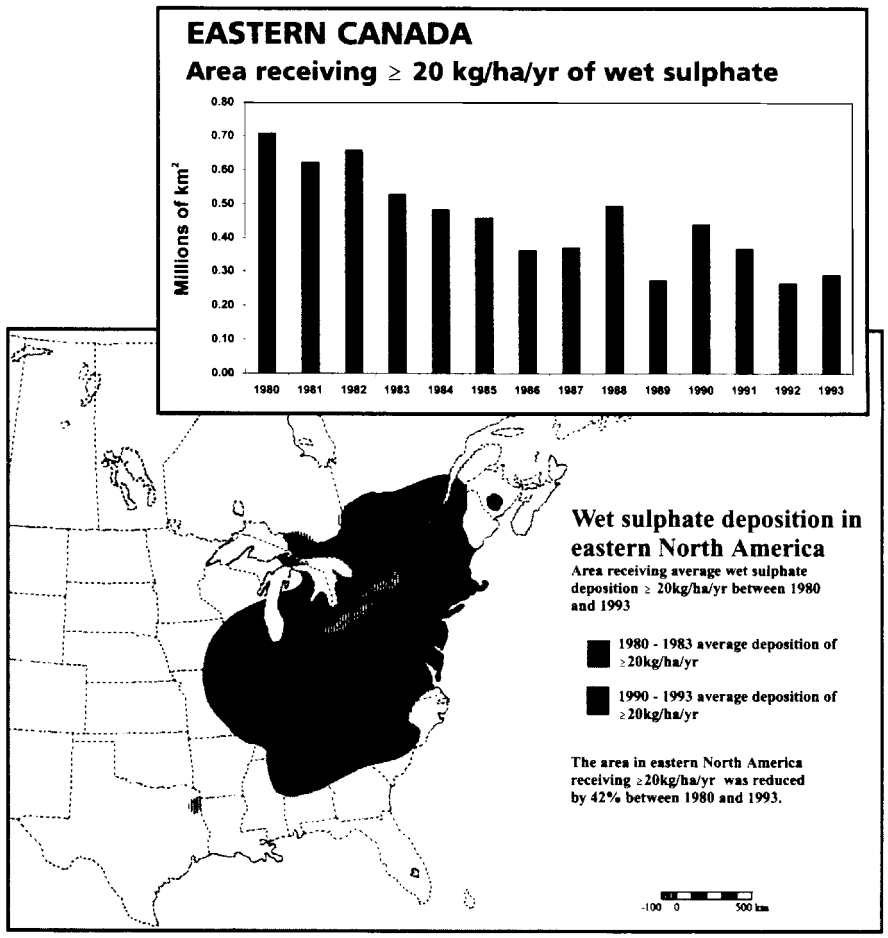
FIG. A-5 United States and Canada acid rain NO_x emission trends. (Source: Environment Canada SOE 96-2, Spring 1996.)

Applications of Sound Intensity Measurements to Gas Turbine Engineering*

Nomenclature

A = total absorption in receiving room
I, I_0 = intensity, pW/m^2
L_1, L_2 = sound pressure levels, dB
LI = sound intensity level, dB

*Source: Altair Filters International Limited, UK; also, this section is adapted from extracts from a paper published in *ASME Journal of Engineering for Gas Turbines and Power*, Vol. 113, October 1991.



Notes:

(a) Wet sulphate deposition represents the weight of sulphate in precipitation and is an indicator of acid rain. The target value of 20 kg/ha/yr of wet sulphate was derived from limited data available in the early 1980s and was based mainly on sport fish loss, which occurs at a pH level of 5.3 and below. More recent studies suggest that a pH of 6.0 is needed to protect most aquatic organisms.

(b) The NAtChem Data Base was used for the storage and analysis of acid precipitation data

(c) National figures on dry sulphate deposition and all nitrate deposition are not available.

Source of data:

Measurements and Analysis Research Division, Atmospheric Environment Service, Environment Canada, Downsview, Ontario.

Map produced by:

State of the Environment Directorate, Environment Canada.

FIG. A-6 Wet sulfate deposits: eastern North America. (Source: Environment Canada SOE 96-2, Spring 1996.)

LK = reactivity index

P, P_0 = pressure, Pa

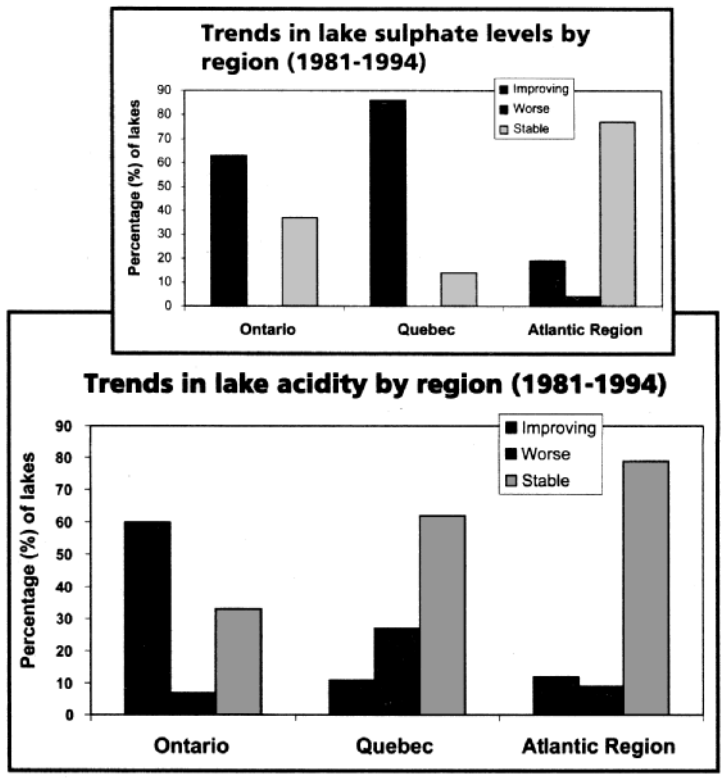
P_a, P_b = pressure, Pa

S = surface area of the test panel, m^2

W, W_0 = power, pW

$r, \Delta r$ = distance, m

t = time, s



Notes:

(a) For acidity: Increasing pH or alkalinity implies an "improving" condition, decreasing pH or alkalinity implies a "worse" condition, no trend means "stable."
(b) For sulphate: Decreasing trends imply "improving," increasing trends imply "worse," no trend means stable.
(c) Number of lakes: Atlantic Region, 75; Quebec, 37; Ontario, 90.
(d) Atlantic Region refers only to the provinces of Nova Scotia and Newfoundland.
(e) Most of the provincial data sets consistently spanned the period from 1981 to 1993/1994. However, some Ontario data sets were slightly longer or shorter than this time period.

Sources:

Methodology: Clair, T.A., *et al.* 1995. "Regional precipitation and surface water chemistry trends in southeastern Canada (1983–1991)." *Canadian Journal of Fisheries and Aquatic Sciences, 52(1): 197–212.*

Data:

Atlantic Region: T.A. Clair, Environment Canada. Quebec: A. Bouchard, Environment Canada. Northwestern Ontario: E.J. Fee, and P. Campbell, Fisheries and Oceans Canada. Algoma: D.S. Jeffries, National Water Research Institute. Sudbury: W. Keller, Ontario Ministry of Environment and Energy. Muskoka/Haliburton: P.J. Dillon, Ontario Ministry of Environment and Energy.

FIG. A-7 Trends in lake sulfate levels (North America). (Source: Environment Canada SOE 96-2, Spring 1996.)

u = particle velocity, m/s

ρ = density, kg/m^3

Gas turbines that are supplied to the oil and power industries are usually given extensive acoustic treatment to reduce the inherent high noise levels to acceptable limits. The cost of this treatment may be a significant proportion of the total cost of the gas turbine installation. In the past it has been difficult to determine if the acoustic treatment is achieving the required noise limits because of a number of

operational problems. These problems include: the presence of other nearby, noisy equipment, the influence of the environment, and instrumentation limitations. Traditionally, sound measurements have been taken using a sound level meter that responds to the total sound pressure at the microphone irrespective of the origin of the sound. So, the enforcement of noise limits has been difficult because of uncertainties concerning the origin of the noise.

Recent advances in signal processing techniques have led to the development of sound intensity meters that can determine the direction, as well as the magnitude, of the sound, without the need for expensive test facilities. These instruments enable the engineer to determine if large equipment, such as gas turbine packages, meet the required noise specification even when tested in the factory or on site where other noise sources are present.

There are, of course, limitations in the use of sound intensity meters, and there are some differences of opinion on measurement techniques. Nevertheless, the acoustic engineer's ability to measure and identify the noise from specific noise sources has been greatly enhanced.

In this section, the differences between sound pressure, sound intensity, and sound power are explained. Measurement techniques are discussed with particular references to the various guidance documents that have been issued. Some case histories of the use of sound intensity meters are presented that include field and laboratory studies relating to gas turbines and other branches of industry.

Fundamental concepts

Sound pressure, sound intensity, and sound power. Any item of equipment that generates noise radiates acoustic energy. The total amount of acoustic energy it radiates is the *sound power*. This is, generally, independent of the environment. What the listener perceives is the *sound pressure* acting on his or her eardrums and it is this parameter that determines the damaging potential of the sound. Unlike the sound power, the sound pressure is very dependent on the environment and the distance from the noise source to the listener.

Traditional acoustic instrumentation, such as sound level meters, detects the sound pressure using a single microphone that responds to the pressure fluctuations incident upon the microphone. Since pressure is a scalar quantity, there is no simple and accurate way that such instrumentation can determine the amount of sound energy radiated by a large source unless the source is tested in a specially built room, such as an echoic or reverberation room, or in the open air away from sound reflecting surfaces. This imposes severe limitations on the usefulness of sound pressure level measurements taken near large equipment that cannot be moved to special acoustic rooms.

Sound intensity is the amount of sound energy radiated per second through a unit area. If a hypothetical surface, or envelope, is fitted around the noise source, then the sound intensity is the number of acoustic watts of energy passing through 1 m^2 of this envelope (see Fig. A-8). The sound intensity, I, normal to the spherical envelope of radius, r, centered on a sound source of acoustic power, W, is given by:

$$I = \frac{W}{4\pi r^2} \tag{1}$$

Clearly, the total sound power is the product of the sound intensity and the total area of the envelope if the sound source radiates uniformly in all directions. Since the intensity is inversely proportional to the distance of the envelope from the noise source, the intensity diminishes as the radius of the envelope increases. But as this

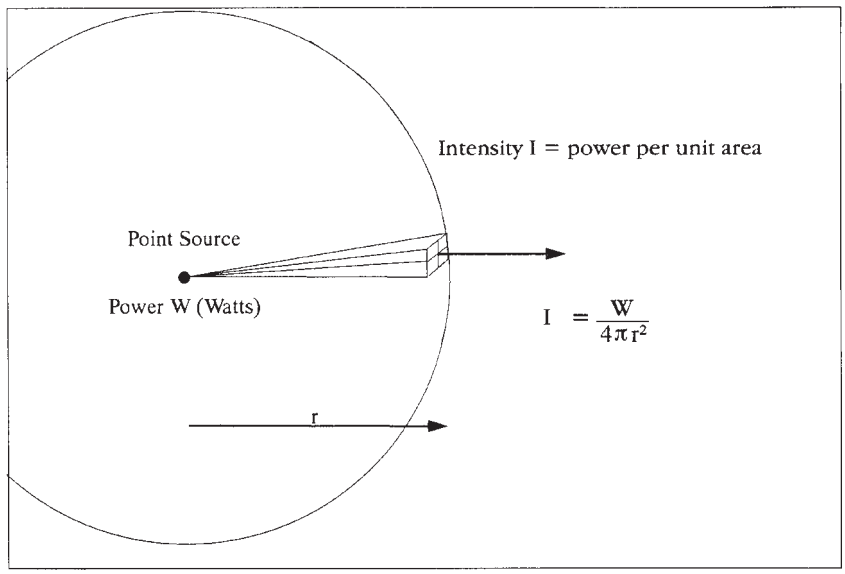


FIG. A-8 The intensity level from a point sound source. (Source: Altair Filters International Limited.)

distance increases, the total area of the envelope increases also, so the product of the intensity and the surface area (equal to the sound power) remains constant.

When a particle of air is displaced from its mean position by a sound wave that is moving through the air there is a temporary increase in pressure. The fact that the air particle has been displaced means that it has velocity. The product of the pressure and the particle velocity is the sound intensity. Since velocity is a vector quantity, so is sound intensity. This means that sound intensity has both direction and magnitude.

It is important to realize that sound intensity is the time-averaged rate of energy flow per unit area. If equal amounts of acoustic energy flow in opposite directions through a hypothetical surface at the same time, then the net intensity at that surface is zero.

Reference levels. Most parameters used in acoustics are expressed in decibels because of the enormous range of absolute levels normally considered. The range of sound pressures that the ear can tolerate is from 2×10^{-5} Pa to 200 Pa. This range is reduced to a manageable size by expressing it in decibels, and is equal to 140 dB.

The sound pressure level (SPL) is defined as:

$$\text{SPL} = 20 \log_{10}\left(\frac{P}{P_0}\right) \quad \text{dB (re. } 2 \times 10^{-5} \text{ Pa)}$$

Likewise, sound intensity level (SIL) and sound power level (PWL) are normally expressed in decibels. In this case,

$$\text{SIL} = 10 \log_{10}\left(\frac{I}{I_0}\right) \quad \text{dB (re. } 1\,\text{pW/m}^2\text{)}$$

$$\text{PWL} = 10 \log_{10}\left(\frac{W}{W_0}\right) \quad \text{dB (re. } 1\,\text{pW)}$$

The relationship between sound pressure level and sound intensity level. When the sound intensity level is measured in a free field in air, then the sound pressure level and sound intensity level in the direction of propagation are numerically the same. In practice most measurements of the sound intensity are not carried out in a free field, in which case there will be a difference between the sound pressure and intensity levels. This difference is an important quantity and is known by several terms, such as reactivity index, pressure–intensity index, *P-I* index, phase index, or LK value. This index is used as a "field indicator" to assess the integrity of a measurement in terms of grades of accuracy or confidence limits. This will be considered in more detail later in this section.

Instrumentation

Sound intensity meters. A sound intensity meter comprises a probe and an analyzer. The analyzer may be of the analog, digital, or FFT (fast Fourier transform) type. The analog type has many practical disadvantages that make it suitable only for surveys and not precision work.

Digital analyzers normally display the results in octave or $^1/_3$ octave frequency bands. They are well suited to detailed investigations of noise sources in the laboratory or on site. Early models tended to be large and heavy and require electrical main supplies, but the latest models are much more suited to site investigations.

FFT analyzers generate spectral lines on a screen. This can make the display very difficult to interpret during survey sweeps because of the amount of detail presented. Another disadvantage of FFT-based systems is that their resolution is generally inadequate for the synthesis of $^1/_3$ octave band spectra.

Sound intensity probes. There are several probe designs that employ either a number of pressure microphones in various configurations or a combination of a pressure microphone and a particle velocity detector. The first type of probe uses nominally identical pressure transducers that are placed close together. Various arrangements have been used with the microphones either side by side, face to face, or back to back. Each configuration has its own advantages and disadvantages.

If the output signals of two microphones are given by P_a and P_b, then the average pressure, P, between the two microphones is:

$$P = {}^1\!/_2\,(P_a + P_b) \tag{2}$$

The particle velocity, u, is derived from the pressure gradient between the two microphones by the relationship:

$$u = -\int \frac{1}{\rho} \cdot \frac{\partial p}{\partial r} \cdot dt$$

$$u = -\frac{1}{\rho} \int \frac{(P_b - P_a)}{\Delta r} \cdot dt \tag{3}$$

Since sound intensity, I, is the product of the pressure and particle velocity, combining equations 2 and 3 gives the intensity as

$$I = -\left(\frac{P_a + P_b}{2\rho\Delta r}\right) \int (P_b - P_a) \cdot dt \tag{4}$$

Figure A-9 shows a two microphone probe, with a face-to-face arrangement, aligned parallel to a sound field. In this orientation the pressure difference is

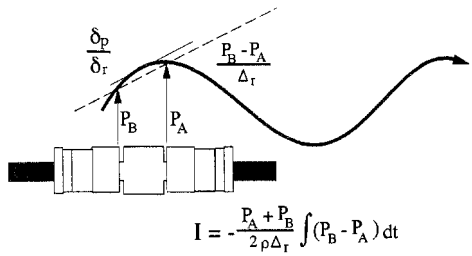


FIG. A-9 The finite difference approximation of sound intensity for a two microphone configuration. (Source: Altair Filters International Limited.)

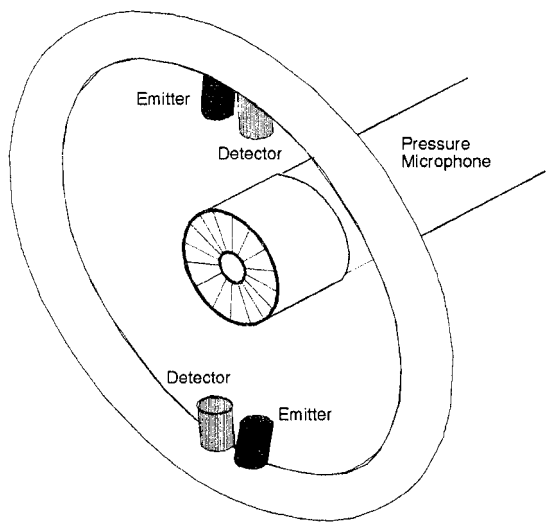


FIG. A-10 Schematic representation of a pressure/velocity probe. (Source: Altair Filters International Limited.)

maximized, and so is the intensity. If the probe is aligned so that the axis of the two microphones is normal to the direction of propagation of the sound wave, then the outputs of the two microphones would be identical in magnitude and phase. Since the particle velocity is related to the difference between the two pressures, P_a and P_b, then the intensity would be zero.

The second type of probe combines a microphone, to measure the pressure, and an ultrasonic particle velocity transducer. Two parallel ultrasonic beams are sent in opposite directions as shown in Fig. A-10. The oscillatory motion of the air caused by audio-frequency sound waves produces a phase difference between the two ultrasonic waves at their respective detectors. This phase difference is related to the particle velocity component in the direction of the beams. This measure of particle velocity is multiplied directly by the pressure to give the sound intensity.

Guidelines and standards in sound intensity measurements and measurement technique

Guidelines and standards. Work began in 1983 on the development of an international standard on the use of sound intensity and the final document is about to be issued. Further standards are expected dealing specifically with instrumentation.

TABLE A-2 Uncertainty of the Determination of Sound Power Level (ISO/DP 9614)

Octave Band Center Frequencies, Hz	1/3 Octave Band Center Frequencies, Hz	Standard Deviations, dB		
		Class 1	Class 2	Class 3
63–125	50–160	2	3	4
250–500	200–630	1.5	2	4
1000–4000	800–5000	1	1.5	4
	6300	2	2.5	4
A-Weighted (50–6300 Hz)		1	1.5	4

NOTES:
1. Class 1 = Precision Grade, Class 2 = Engineering Grade, Class 3 = Survey Grade.
2. The width of the 95% confidence intervals corresponds approximately to four times the dB values in this table.

In the absence of a full standard the only guidance available was the draft ISO standard (ISO/DP 9614) and a proposed Scandinavian standard (DS F88/146).

The ISO document ISO/DP 9614 specifies methods for determining the sound power levels of noise sources within specific ranges of uncertainty.

The proposed test conditions are less restrictive than those required by the International Standards series ISO 3740-3747, which are based on sound pressure measurements. The proposed standard is based on the sampling of the intensity normal to a measurement surface at discrete points on this surface. The method can be applied to most noise sources that emit noise that is stationary in time and it does not require special purpose test environments.

The draft document defines three grades of accuracy with specified levels of uncertainty for each grade. Since the level of uncertainty in the measurements is related to the source noise field, the background noise field, and the sampling and measurement procedures, initial procedures are proposed that determine the accuracy of the measurements. These procedures evaluate the “Field Indicators” that indicate the quality of the sound power measurements. These field indicators consider, among other things:

- The pressure–intensity index (or reactivity index)
- The variation of the normal sound intensities over the range of the measurement points
- The temporal variation of the pressure level at certain monitoring points

The three grades of measurement accuracy specified in ISO/DP 9614, and the associated levels of uncertainty, are given in Table A-2.

The Scandinavian proposed standard (DSF 88/146) was developed for the determination of the sound power of a sound source under its normal operating conditions and in situ. The method uses the scanning technique whereby the intensity probe is moved slowly over a defined surface while the signal analyzer time-averages the measured quantity during the scanning period.

The results of a series of field trials by several Scandinavian organizations suggested that the accuracy of this proposed standard is compatible with the “Engineering Grade,” as defined in the ISO 3740 series.

The equipment under test is divided into a convenient number of subareas that are selected to enable a well-controlled probe sweep over the subarea. Guidance is given on the sweep rate and the line density. Measurement accuracy is graded according to the global pressure–intensity index, LK. This is the numerical difference between the sound intensity level and the sound pressure level. If this

field indicator is less than or equal to 10 dB then the results are considered to meet the engineering grade of measurement accuracy. As this field indicator increases in value, the level of uncertainty in the intensity measurement increases. When the LK value lies between 10 and 15 dB the measurement accuracy meets the “Survey” grade.

Measurement techniques. The precise measurement technique adopted in a particular situation depends on the objectives of the investigation and the level of measurement uncertainty that is required.

(A) Subareas. It was mentioned earlier that the total sound power is the product of the intensity and the surface area of the measurement envelope around the noise source. In practice, most noise sources do not radiate energy uniformly in all directions so it is good practice to divide the sound source envelope into several subareas. Each subarea is then assessed separately, taking into account its area and the corresponding intensity level. The subarea sound powers can then be combined to give the total sound power of the source.

The number, shape, and size of each subarea is normally dictated by two considerations: the physical shape of the source and the variations in intensity over the complete envelope. Subareas are normally selected to conform to components of the whole source such that the intensity over the subarea is reasonably constant. It is important that the subareas are contiguous and the measurement envelope totally encloses the source under investigation.

(B) Sweep or point measurement. Should one measure the intensity levels at discrete positions, with the probe stationary, or should the probe be swept over the subarea? This controversy has occupied much discussion time among practicing acousticians. For precision grade measurements, discrete points are used, but for lower grade work, sweeping is acceptable.

If discrete points are used then the number and distribution of the measurement points must be considered in relation to the field indicators.

In surroundings that are not highly reverberant and where extraneous noise levels are lower than the levels from the source under investigation, relatively few discrete points may be used, distributed uniformly over the surface. The distance from the source may be as great as 1 m.

As the extraneous noise levels increase and/or the environment becomes more reverberant, measurements must be made progressively closer to the source in order to maintain an acceptable level of uncertainty in the measurements. This also requires more measurement points to be used because of the increase in the spatial variation of the intensity distribution.

If sweeping is used then other factors must also be considered. The speed with which the probe is swept across the subarea must be uniform, at about 300 mm/sec, and the area should be covered by a whole number of sweeps with an equal separation between sweep lines. Care must be taken that excessive dwell time does not occur at the edges of the subarea when the probe’s direction of sweep is reversed. The operator must also be careful that his or her body does not influence the measurements by obscuring sound entering the measurement area as he or she sweeps.

(C) Distance between source and probe. Generally, the greater the extraneous noise and the more reverberant the environment then the closer should be the probe to the source. In extreme cases the probe may be only a few centimeters from the source surface in order to improve the signal-to-noise ratio. This is normally frowned upon when using conventional sound level meters because measurements

of sound pressure, taken close to a surface, may bear little relation to the pressures occurring further away from the surface. This discrepancy is not due simply to the attenuation with distance that normally occurs in acoustics.

The region very close to a surface is called the "near field." In this region the local variations in sound pressure may be very complex because some of the sound energy may circulate within this near field and not escape to the "far field." This recirculating energy is known as the *reactive* sound field. The sound energy that does propagate away from the surface is called the *active* sound field because this is the component that is responsible for the acoustic energy in the far field.

Since sound intensity meters can differentiate between the active and reactive sound fields, measurements of intensity taken close to noise sources can faithfully indicate the radiated sound energy. However, using a conventional sound level meter near to a noise source may indicate higher sound power levels than occur in the far field because these instruments cannot differentiate between active and reactive fields.

Some advantages and limitations in sound intensity measurements

Background noise. One of the main advantages of the sound intensity method of measurement is that accurate assessments of sound power can be made even in relatively high levels of background noise. But this is only true if the background noise is steady (i.e., not time varying). Using conventional sound pressure level methods the background noise level should be 10 dB below the signal level of interest.

Using sound intensity techniques the sound power of a source can be measured to an accuracy of 1 dB even when the background noise is 10 dB *higher* than the source noise of interest. Figure A-11*a* shows a noisy machine enclosed by a measurement surface. If the background noise is steady, and there is no sound absorption within the measurement surface, then the total sound power emitted by the machine will pass through the measurement surface, as shown.

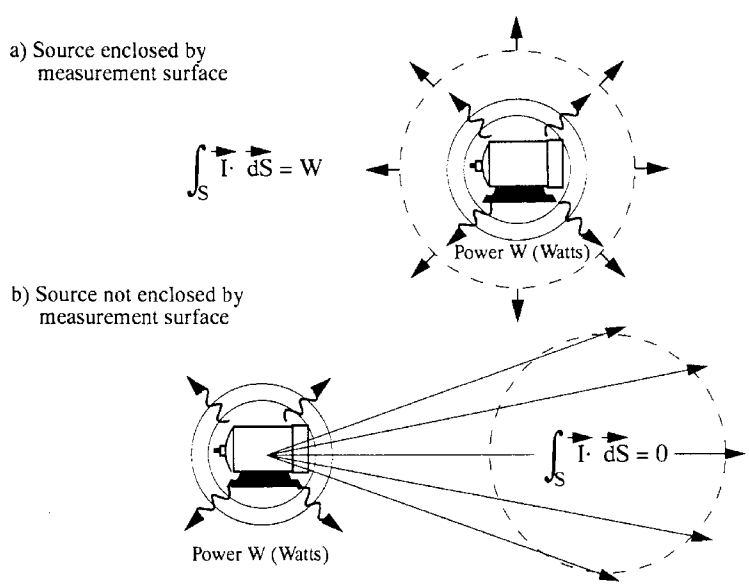


FIG. A-11 The effect of sound sources inside and outside the measurement surface. (Source: Altair Filters International Limited.)

If, however, the noisy machine is outside the measurement surface, as shown in Fig. A-11*b*, then the sound energy flowing into the surface on the left hand side will be emitted from the right hand side of the measurement surface. When the sound intensity is assessed over the whole measurement surface the net sound power radiated from the total surface will be zero.

Effects of the environment. When the sound power of a noise source is evaluated in the field using sound pressure level techniques, it is necessary to apply a correction to the measured levels to account for the effects of the environment. This environmental correction accounts for the influence of undesired sound reflections from room boundaries and nearby objects.

Since a sound intensity survey sums the energy over a closed measurement surface centered on the source of interest, the effects of the environment are cancelled out in the summation process in the same way that background noise is eliminated. This means that, within reasonable limits, sound power measurements can be made in the normal operating environment even when the machine under investigation is surrounded by similar machines that are also operating.

Sound source location. Since a sound intensity probe has strong directional characteristics there is a plane at 90° to the axis of the probe in which the probe is very insensitive. A sound source just forward of this plane will indicate positive intensity, whereas if it is just behind this plane the intensity will be negative (Fig. A-12).

This property of the probe can be used to identify noise sources in many practical situations. The normal procedure is to perform an initial survey of the noise source to determine its total sound power. The probe is pointed toward the source system to identify areas of high sound intensity. Then the probe is reoriented to lie parallel to the measurement surface and the scan is repeated. As the probe moves across a dominant source the intensity vector will flip to the opposite direction.

Testing of panels. The traditional procedure for measuring the transmission loss, or sound reduction index, of building components is described in the series of

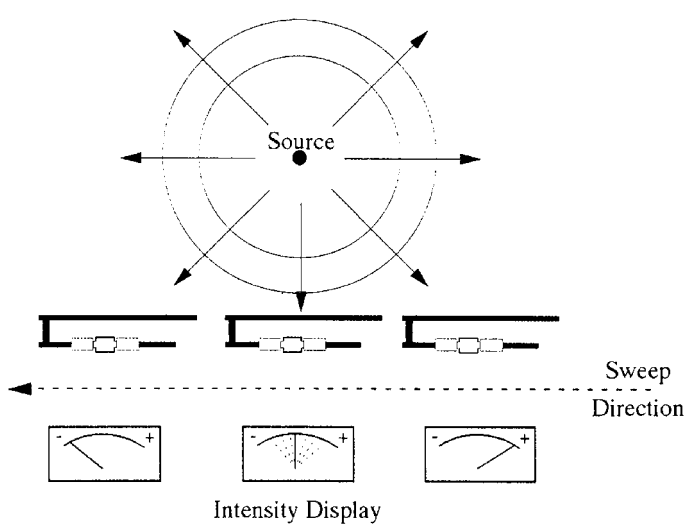


FIG. A-12 Sound source location using the intensity probe. (Source: Altair Filters International Limited.)

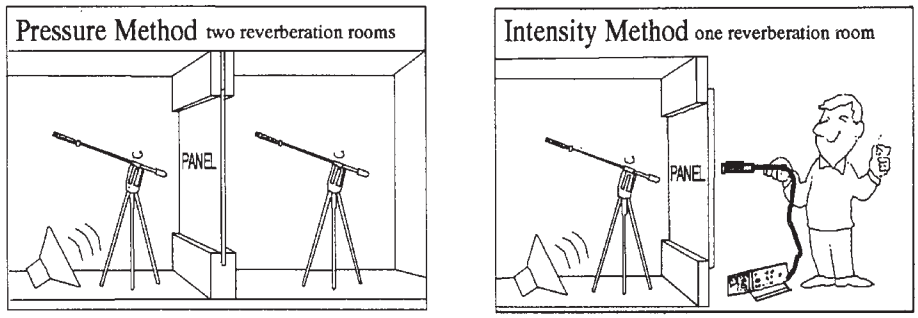


FIG. A-13 Comparison of the pressure and intensity methods for measuring the sound reduction index of panels. (Source: Altair Filters International Limited.)

standards ISO 140. The test method requires the panel under investigation to be placed in an opening between two independent, structurally isolated reverberation rooms, as shown in Fig. A-13.

Sound is generated in the left hand room and the sound pressure levels in the two rooms are measured. Assuming that the sound energy in the right hand room comes through the panel then the sound reduction index (SRI) of the panel is given by:

$$\text{SRI} = L_1 - L_2 + 10 * \log_{10}(S/A)\,\text{dB} \tag{5}$$

For this method to give accurate results, flanking transmission (sound bypassing the test panel) must be minimal and both rooms must be highly reverberant.

If the measurements in the right hand room are carried out using sound intensity, then it is necessary to reduce the amount of reverberation in this room. Since the probe can measure the sound intensity coming through the panel then flanking transmission is no longer a limitation.

For these reasons one can dispense with the second reverberation room altogether. The sound reduction index is then given by:

$$\text{SRI} = L_1 - LI - 6\,\text{dB} \tag{6}$$

If the panel contains a weak area, such as a window, the sound reduction index of the window can be assessed separately. But this will only work if the panel is a greater sound insulator than the window.

Measuring tonal noise sources. Measuring the sound power of tonal noise sources presents difficulties using traditional techniques (ISO 3740, 1980). Unfortunately, using sound intensity techniques on such sources is also fraught with problems. This is because the spatial distribution of the intensity is very sensitive to small alterations in source position and the presence of nearby sound reflective objects.

Case studies of the use of sound intensity

Gas turbine package witness testing. Gas turbine packages are normally assembled in large factory buildings or in the open air between factory buildings. In either case, the environment is totally unsuitable for reliable acoustic tests to be carried out using sound level meters alone. Sound intensity techniques are especially relevant in these situations because of the location in which the tests are to be carried out and because some components, such as the compressor test loop, may

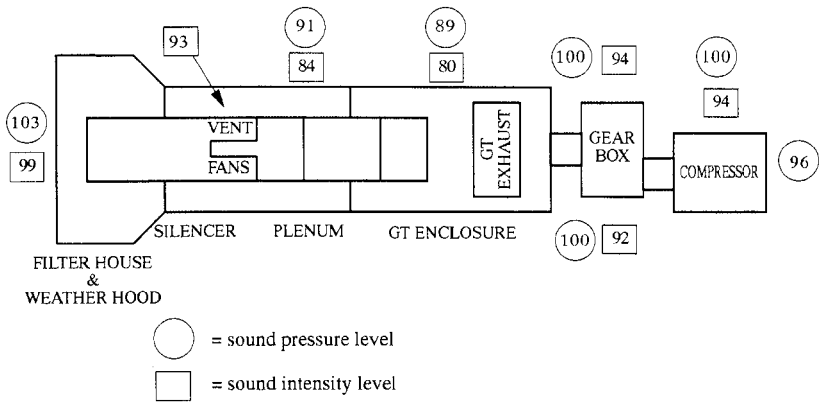


FIG. A-14 Sound pressure and sound intensity levels for a gas turbine package during witness testing. (Source: Altair Filters International Limited.)

TABLE A-3 Rank Ordering of Components in Terms of the A-Weighted Sound Power Levels

Description of Measured Item	Measured Sound Power Levels, dB(A)
Combustion air intake	110
Combustion air plenum and silencer	104
Turbine comp. vent. air breakout	104
Turbine enclosure	103
Compressor casing	103
Gearbox	103
Breakout from temporary exhaust	96

not be contract items. By surveying each component with a sound intensity meter the sound power for each component can be determined separately.

Figure A-14 shows a typical gas turbine driving a compressor. The figures on the drawing indicate the sound pressure and sound intensity levels that were measured during a particular witness test on an RB211 gas turbine package. The sound pressure levels in close proximity to the package were between 89 and 103 dB(A), with the higher levels dominating. Even so, reliable values of the intensity levels were obtained from which the sound power levels were determined. These values are given in Table A-3. Since the sound intensity level is numerically equal to the sound pressure level in free field, the average sound intensity over a given surface area of a gas turbine package provides a direct indication of the average sound pressure level from that surface in free field conditions.

Referring again to Fig. A-14, the sound intensity level measured by the casing of the ventilation fan was 94 dB(A). It would not normally be possible to measure the output from this fan accurately, using a sound level meter, in this situation because of the relatively high sound pressure level in this area due to other sources.

During the testing of another package, the sound power levels from the ventilation fan casing and the fan motor were measured separately. The fan motor was found to be noisier than the manufacturer's stated levels. Discussions with the motor manufacturer revealed that the wrong cooling fans had been fitted to the motors, which accounted for this increase in noise level. The correct cooling fans were subsequently fitted.

TABLE A-4 Comparison of the Sound Levels from Two Roller Mills

Description	Machine A, dB(A)	Machine B, dB(A)
Total sound power for machine	96	94
Sound pressure level by machine	92	89
Sound power level of the motor	93	85

This example clearly illustrates the benefits of sound intensity measurements to check compliance with noise specifications when the test items are very large and are sited in acoustically undesirable areas.

Comparison of two nominally identical production machines. This example is taken from an extensive survey of a production department that had many, relatively small, machines close together in a highly reverberant factory room. The two machines were nominally identical roller mills, as used in many production lines in the paint, flour, and confectionary industries. The drive motors were situated on the top of the machines. A routine sound intensity survey was carried out on each machine during normal production because it was not possible to run the machines in isolation. The two machines are identified as machine A and machine B.

Table A-4 gives the overall, A-weighted sound pressure levels and sound power levels for each machine, and the sound power levels of the motors. The total sound power levels of the machines agreed very well with the values obtained by the manufacturer using sound pressure measurements to derive the sound power levels (ISO 3740, 1980). This technique can only give the total sound power of a machine; it cannot obtain the sound power levels of parts of a machine.

At the time of the survey two machines were each in areas of high noise levels and their total sound power levels were 96 and 94 dB(A). But comparing the motor sound power levels revealed a difference of 8 dB(A) in their respective levels even though both motors were classified as the "low noise" type and cost more than the standard motors. This is just one example where a significant degree of noise control might be achievable by selecting the correct one of two nominally identical electric motors. However, using the traditional method of sound pressure level measurement would not reveal any difference between the motors.

The acoustical properties of flexible connectors. Heavy-duty flexible connectors are used to join separate components of a gas turbine package. When high-performance acoustic hardware is used it is imperative that these flexible connectors do not compromise the total acoustic performance of the package. This is particularly so in the gas turbine exhaust system where multilayered flexible connectors are exposed to high temperatures and severe buffeting from exhaust gases. In some exhaust systems overlapping metal plates are inserted inside the flexible connectors to reduce the buffeting of the flexible material. If additional sound attenuation is required, a heavy, fibrous mat, or "bolster," is inserted between the plates and the flexible connector.

For the acoustics engineer, these flexible connectors are a problem because there is very little data on their acoustical performance, and the designs do not lend themselves to simple theoretical prediction. A brief laboratory investigation was carried out to compare the performances of seven types of flexible connectors with and without plates and bolsters. The tests were carried out in Altair's acoustical laboratory, which was designed to test materials using sound intensity techniques. The samples were physically quite small so the low-frequency performances were

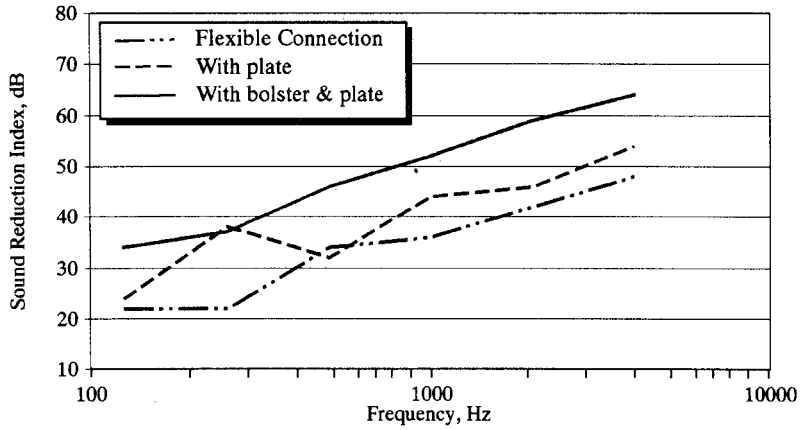


FIG. A-15 Sound reduction index of a flexible connection. (Source: Altair Filters International Limited.)

probably distorted by the small size of the samples. Nevertheless, the exercise yielded much valuable information and led to a simple engineering method of predicting a flexible connector's acoustic performance from a knowledge of its basic parameters.

Figure A-15 compares the sound reduction indices of a typical, multilayered flexible connector tested alone, with plates and with bolster and plates. All of the test results showed a sharp increase in performance at 250 Hz due to the plates. No satisfactory explanation can be offered at this stage for this effect, which may be related to the small size of the samples. But in the middle and high frequencies the results were generally as expected with the plate giving an additional attenuation of about 5 dB compared to the compensator alone. The combination of the plate and bolster gave an additional attenuation of between 10 and 15 dB compared to the flexible connector alone.

Sound source location. During two noise surveys the sound power levels from two different designs of lube oil console were measured using the sound intensity meter. The overall sound power levels were 93 dB(A) and 109 dB(A). The two consoles had electrically driven pumps and both emitted strong tonal noise. In the first case the tonal noise was centered on 8 kHz; using the sound intensity probe it was possible to "home in" on the pump suction pipe as a major noise source. This was confirmed by vibration velocity measurements.

In the second survey the pump outlet pipe gave the highest sound intensity reading. Since this was a relatively long pipe, which was rigidly attached to the frame of the console, it was a dominant source both in its own right and because it was "exciting" the framework. In these cases the sound intensity meter was a useful tool in identifying dominant sources among a large number of small, closely packed noise radiators.

This section has discussed the concepts of sound intensity, sound power, and sound pressure. It has shown how sound intensity meters have given the acoustics engineer a very powerful diagnostic tool. Noise specifications for large, complex machinery can now be checked without the need for special acoustics rooms.

The advantages, and limitations, of sound intensity have been discussed in some detail and several applications have been illustrated by case histories taken from surveys carried out in the process and gas turbine industries.

The superiority of sound intensity meters over sound level meters is clearly apparent. Certain types of laboratory studies can also be carried out more cost effectively using intensity techniques.

Although there are some limitations in the use of sound intensity instrumentation, when used intelligently, it can yield valuable information on dominant noise sources, which, in turn, should provide more cost-effective solutions to noise control in the oil and power industries.

Acoustic Design of Lightweight Gas Turbine Enclosures*

Nomenclature

a, b = panel dimensions, m
B, Bx, By, Bxy = bending and torsional stiffnesses
c = speed of sound in air, m/s
d = fiber diameter, μm
f = frequency, Hz
$f1$ = first panel resonance, Hz
f_c, f_{cx}, f_{cy} = coincidence frequency, Hz
l = thickness of absorptive layer, m
ln = natural logarithm
m = mass per unit area, kg/m^2
$R1$ = flow resistivity index or transmission loss, dB
R = sound reduction
S = stiffness
α = attenuation constant for the material, dB/m
η = damping
λ_m = wavelength of sound in the absorptive layer, m
υ = Poisson ratio
ρ_0 = density of air
ρ_m = density of absorptive layer
τ = transmission coefficient
ω = angular frequency

Gas turbines are used extensively in onshore and offshore environments for power generation, but their use introduces a number of potential hazards. To reduce the risks caused by fire and high noise levels, enclosures, with intake and exhaust silencers, are fitted around the turbines. These enclosures and silencers must be capable of withstanding large static loads produced by equipment sited on top of them and large dynamic loads due to wind.

Traditionally these enclosures are heavy and expensive, especially when stainless steel or aluminum is required for offshore use. This has led to a consideration of more cost-effective designs that still comply with the stringent demands of the oil and gas industry.

One approach that is proving successful is the use of a corrugated enclosure design, which employs a thinner steel wall than its flat panel counterpart, without compromising the structural and fire protection requirements. However, corrugated designs are intrinsically less effective as sound insulators than flat panels. These weaknesses must be understood so that multilayered panels based on the corrugated design can compensate for the deficiencies of unlined, corrugated panels.

This section presents the results of theoretical predictions and measurements on flat and corrugated panels, which were tested in the unlined condition and then with a sound absorbent lining. The effects of varying the profile of the corrugations is also considered.

---

* Source: Altair Filters International Limited, UK; also, this section is adapted from extracts from a paper published in *ASME Journal of Engineering for Gas Turbines and Power*, Vol. 113, October 1991.

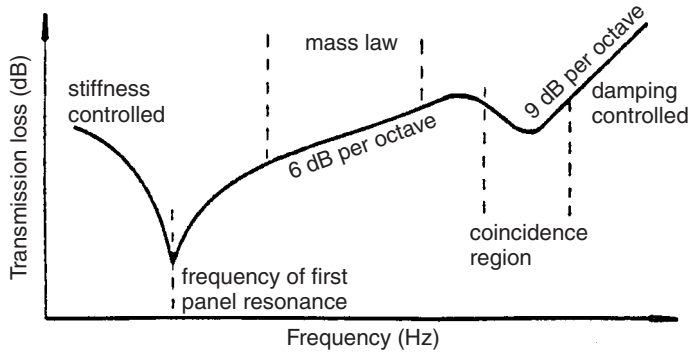


FIG. A-16 Sound reduction index for a typical flat, unlined panel. (Source: Altair Filters International Limited.)

This section considers first the behavior of flat, unlined panels, then describes the physical reasons why corrugated, unlined panels have a different acoustic response to flat panels. The effects of sound-absorbent linings on flat and corrugated panels are then considered.

Sound transmission through unlined flat panels

When a sound wave is incident on a wall or partition, some of the sound energy is transmitted through the wall. The fraction of incident energy that is transmitted is called the transmission coefficient. The accepted index of sound transmission is the sound reduction index, which is sometimes called the transmission loss. This is related to the transmission coefficient by the equation:

$$R = 10 \log_{10}(1/\tau), \text{dB} \tag{1}$$

The behavior of flat panels has been described extensively in the literature, so only the outline of their theoretical performance is given here. The general behavior of a single skin, isotropic panel is shown in Fig. A-16. This characteristic behavior is valid for a wide range of materials, including steel and aluminum.

The propagation of audio-frequency waves through panels and walls is primarily due to the excitation of bending waves, which are a combination of shear and compressional waves. When a panel is of finite extent then a number of resonances are set up in the panel that are dependent on the bending stiffness of the panel. The frequency of the first panel resonance, $f1$, is given by

$$f1 = (\pi/2)\sqrt{(B/m)(1/a^2 + 1/b^2)}, \text{Hz} \tag{2}$$

The second resonance occurs at the "coincidence" frequency, fc, when the projected wavelength of the incident sound coincides with the wavelength of the bending wave in the panel. The coincidence frequency is given by:

$$f_c = (c^2/2\pi)\sqrt{(m/B)}, \text{Hz} \tag{3}$$

The amount of sound transmitted by a panel is dependent on the surface weight, the damping of the panel, and the frequency of the sound. For a finite panel exposed to a random noise field, the acoustic behavior is specified mathematically as follows:

$$R = 20\log_{10} S - 20\log_{10} f - 20\log_{10}(4\pi\rho_0 c), \text{dB}, f < f1 \tag{4}$$

TABLE A-5 Comparison of Predicted and Measured Sound Reduction Indices of Flat, Unlined Panels

Material	Measured or Predicted	Sound Reduction Index, dB							
		63	125	250	500	1000	2000	4000	8000
3-mm steel	Predicted	16	22	28	34	40	43	35	44
	Measured	16	21	27	33	38	39	33	—
6-mm steel	Predicted	22	28	34	40	43	35	44	53
	Measured	22	27	35	39	44	37	42	—
6-mm aluminum	Predicted	13	19	25	31	34	26	35	44
	Measured	13	19	25	30	36	30	32	—

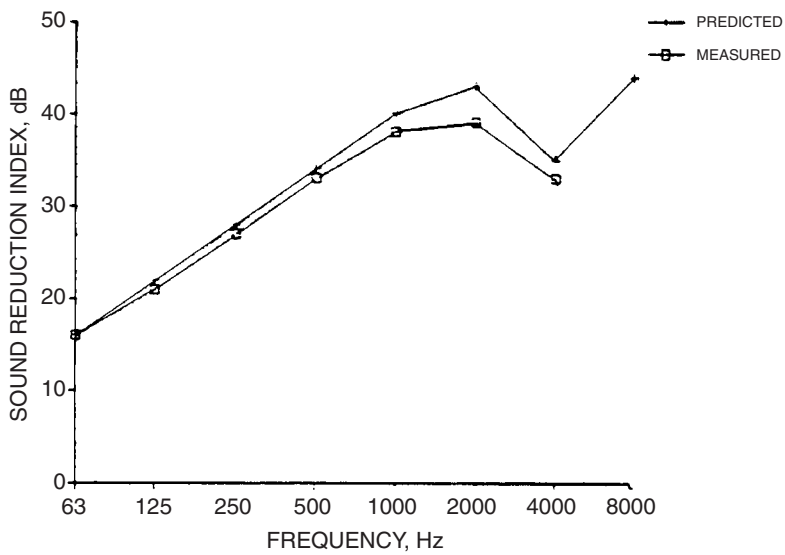


FIG. A-17 Predicted and measured sound reduction indices for 3-mm steel. (Source: Altair Filters International Limited.)

$$R = 20\log_{10}(mf) - 47, \text{dB}, f1 < f < f_c \tag{5}$$

$$R = 20\log_{10}(\pi f m/\rho_0 c) + 10\log_{10}(2\eta f/\pi f_c), \text{dB}, f > f_c \tag{6}$$

In most practical situations the lowest resonance frequency is below the audio range. Above this frequency a broad frequency range occurs in which the transmission loss is controlled by the surface weight and increases with frequency at the rate of 6 dB per octave. In the coincidence region the transmission loss is limited by the damping of the panel. Above the coincidence frequency the transmission loss increases by 9 dB per octave and is determined by the surface weight and the damping.

Clearly for a high value of sound reduction index over the majority of the audio-frequency range (the mass-controlled region) it is better to have a high surface weight, a low bending stiffness, and a high internal damping.

The acoustic performances of some materials have been predicted and are compared to measured values in Table A-5. The data for 3-mm steel and 6-mm aluminum are shown in Figs. A-17 and A-18. Each of the examples shown has the characteristic shape described above and the agreement between the measured and predicted values of transmission loss is good.

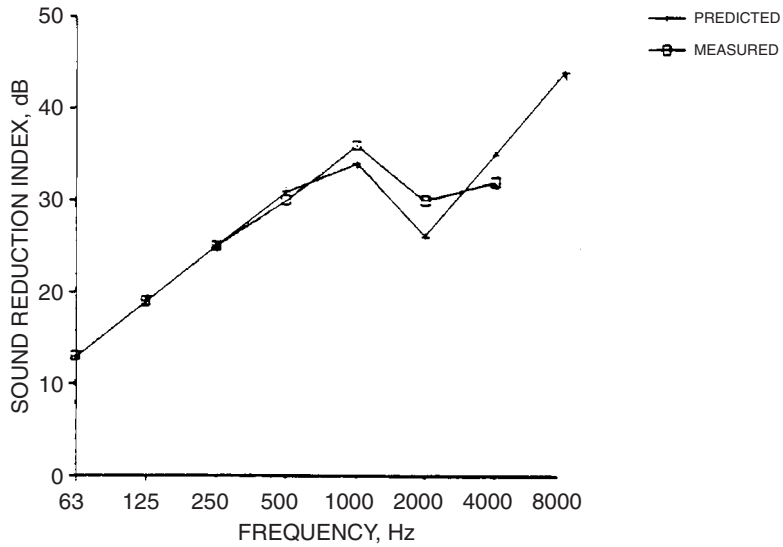


FIG. A-18 Predicted and measured sound reduction indices for 6-mm aluminum. (Source: Altair Filters International Limited.)

Sound transmission through unlined corrugated panels

In corrugated panels the characteristics of the panel are not the same in all directions. The moment of inertia across the corrugations differs from that parallel to the corrugations; thus the bending stiffness varies with direction. This affects both the first panel resonance and the coincidence frequency so that the sound transmission characteristics for these panels differs from flat panels with the same thickness.

The first panel resonance is now given by:

$$f1 = (\pi/2m^{0.5}) \times \sqrt{(Bx/a^4(1-v^2) + By/b^4(1-v^2) + Bxy/a^2b^2)}, \text{Hz} \quad (7)$$

where Bx and By are the bending stiffnesses in the two principal planes of the panel and Bxy accounts for the torsional rigidity of the plate.

For real panels, measuring several meters in length and height, the frequency of this first resonance may still be in the subaudio range but can be several octaves higher than the first resonance of a flat panel of the same dimensions.

Since the coincidence frequency is determined by the bending stiffness, the presence of two bending stiffnesses gives rise to two critical frequencies, f_{cx} and f_{cv}, where:

$$f_{cx} = (c^2/2\pi)\sqrt{(m/Bx)},\ f_{cv} = (c^2/2\pi)\sqrt{(m/By)}$$

If the ratio of the bending stiffnesses, Bx and By, is less than 1.4, then the effects on the transmission loss of the panel will be small, but in typical panels the ratio of the bending stiffnesses is usually much greater than 1.4. This gives rise to a plateau in the transmission loss curve, which is illustrated in Fig. A-19. The plateau may extend over several decades for common corrugated or ribbed panels.

The sound transmission through orthotropic (corrugated) panels has been investigated by Heckl, who derived the following relationships for the diffuse field sound transmission:

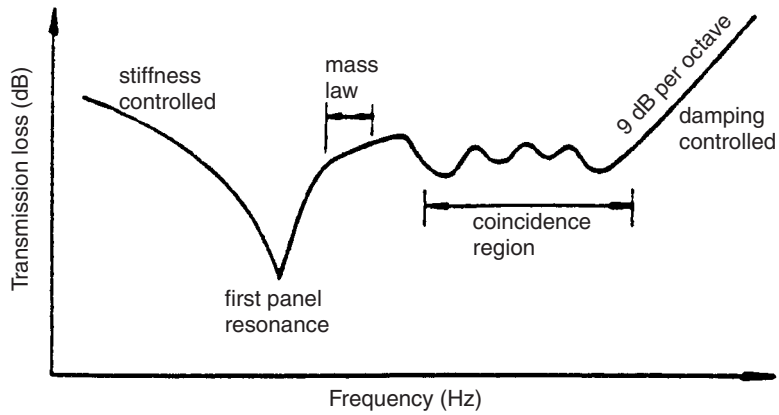


FIG. A-19 Sound reduction index for a typical corrugated, unlined panel. (Source: Altair Filters International Limited.)

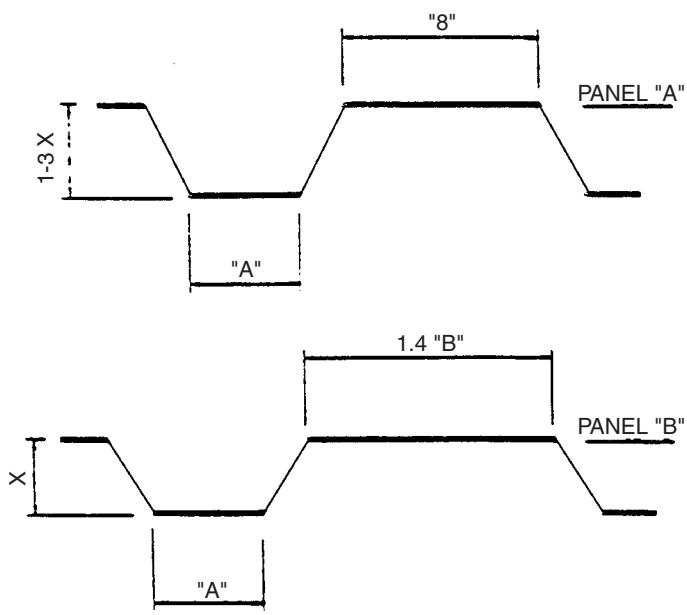


FIG. A-20 Two designs of corrugated panels. (Source: Altair Filters International Limited.)

$$\tau = (\rho_0 c f_{cx}/\pi\omega mf)(\ln(4f/f_{cx}))^2, f_{cx} < f < f_{cy} \quad (8)$$

$$\tau = (\pi\rho_0 c/\omega mf)(f_{cx}f_{cy})^{0.5}, f > f_{cy} \quad (9)$$

where f_{cx} and f_{cy} are the two coincidence frequencies and where f_{cx} is the lower of the two values.

The performances of two designs of corrugated panels have been predicted for panels made of 2.5-mm-thick steel with the designs shown in Fig. A-20. The predicted transmission losses are given in Table A-6 and Fig. A-21.

The predicted panel bending stiffnesses and resonant frequencies for the two panels are tabulated in Table A-7 for a simply supported panel 4 m wide by 2.5 m high. The values for a flat panel of the same overall dimensions have also been

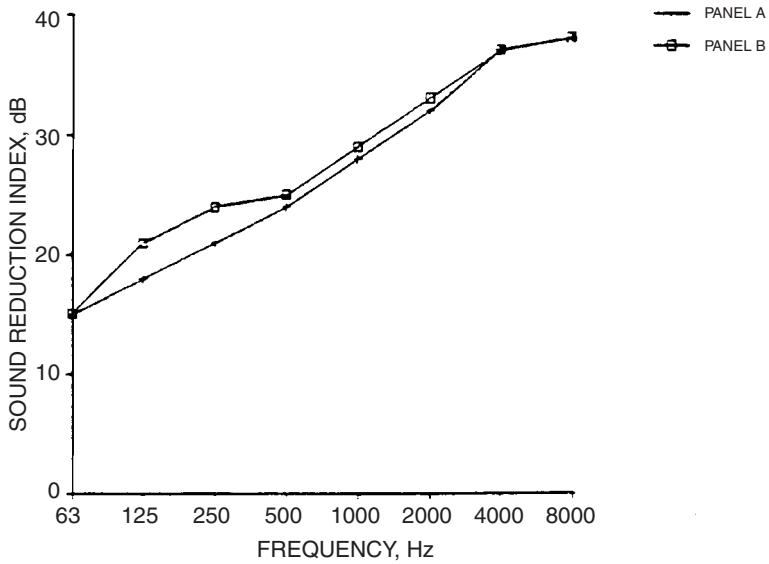


FIG. A-21 Predicted transmission loss for two designs of corrugated panels. (Source: Altair Filters International Limited.)

TABLE A-6 Comparison of Predicted Sound Reduction Indices of Two, Unlined Corrugated Panels

	Sound Reduction Index, dB							
	63	125	250	500	1000	2000	4000	8000
Panel A	15	18	21	24	28	32	37	38
Panel B	15	21	24	25	29	33	37	38

TABLE A-7 Predicted Parameters for Three Panels (the panels were made of steel measuring 4 m by 2.5 m by 2.5 mm thick)

	Bending Stiffness, N·m	First Panel Resonance, Hz	Coincidence Frequency, Hz
Panel A (corrugated)	240,000 : 241	30	163 : 5115
Panel B (corrugated)	90,000 : 263	18	264 : 4897
Flat panel	296	2.9	4825

included. The bending stiffness of panel B, in the direction parallel to the corrugations, is slightly less than that for panel A. This causes a shift in the lower critical frequency of about half an octave, which gives rise to a slightly higher transmission loss at lower frequencies.

Transmission loss measurements have been carried out on a partition with the design of panel A. The test was carried out in accordance with the standard ISO 140 (1978). Table A-8 and Fig. A-22 compare the predicted and measured performances. Both sets of curves show a plateau effect, which is more evident in the measured values. The predicted and measured values are within 4 dB of each

TABLE A-8 Predicted and Measured Sound Reduction Indices for Panel A

	Sound Reduction Index, dB							
	63	125	250	500	1000	2000	4000	8000
Predicted	15	18	21	24	28	32	37	38
Measured	20	23	25	23	23	28	29	29

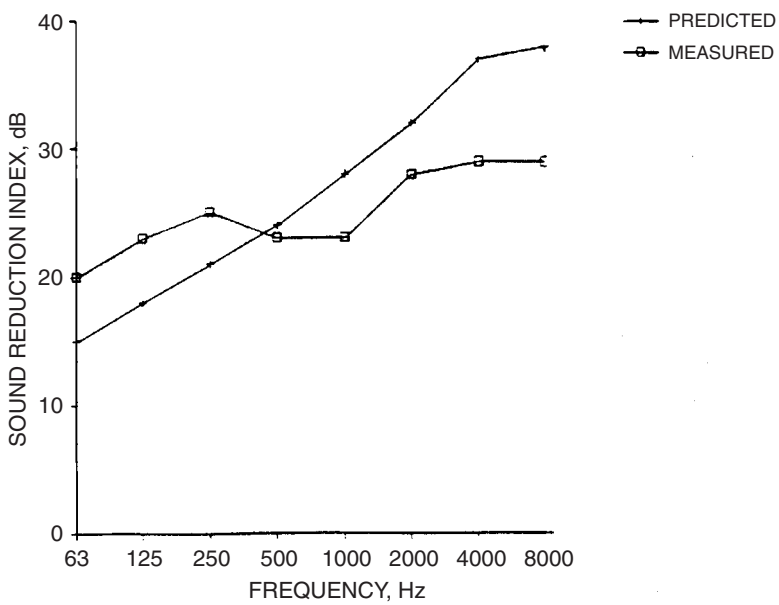


FIG. A-22 Predicted and measured sound reduction indices for panel A. (Source: Altair Filters International Limited.)

other up to 4 kHz, above which the curves differ by about 7 dB. Although the agreement is not as close as for flat panels, it is encouraging. Of more interest is the comparison between the transmission losses of unlined, flat, and corrugated panels with the same thickness. Figure A-23 and Table A-9 show that the corrugated panel is substantially less effective in reducing sound transmission in the mid-frequencies compared to the flat panel equivalent.

Clearly, if noise control is a primary consideration then unlined corrugated panels are not recommended, unless other engineering considerations dictate their use. Corrugated panels offer a considerable increase in bending stiffness, compared to flat panels, which reduces the amount of additional stiffening that would be required for a flat panel. But the volume of material required to produce a corrugated panel is significantly greater than that for a flat panel. For the two corrugated panel designs considered above, panel A contains 23 percent more material than the flat panel, and panel B contains 13 percent more material. Depending on the structural requirements, the additional stiffening required for a flat panel may result in a material savings when a corrugated design is chosen, especially if the flat panel is much thicker to compensate for its inherent low bending stiffness.

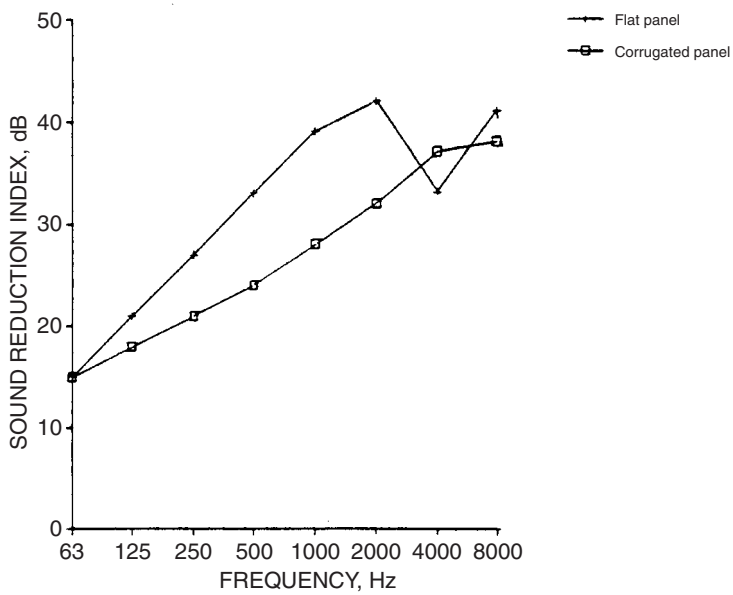


FIG. A-23 Predicted performance of flat and corrugated panels. (Source: Altair Filters International Limited.)

TABLE A-9 Predicted Sound Reduction Indices for Flat Corrugated Panels (Panel Thickness of 2.5 mm)

	Sound Reduction Index, dB							
	63	125	250	500	1000	2000	4000	8000
Flat panel	15	21	27	33	39	42	33	41
Corrugated panel	15	18	21	24	28	32	37	38

The attenuation of sound by absorptive linings

Sound-absorbent linings are frequently fitted in acoustic enclosures to reduce the buildup of reverberant noise inside the enclosure. Typical reductions in the reverberant noise level may be between 3 and 10 dB depending on the application. An additional benefit is the increase in transmission loss of the enclosure panel, which further reduces the noise level outside the enclosure.

The increase in panel transmission loss arises because of several mechanisms. First, if the absorbent lining is sufficiently heavy and the panel is relatively thin, then the added layer may give sufficient additional weight to affect the "mass-law" performance and increase the damping of the panel. At high frequencies the absorbent may be relatively thick in comparison to the wavelength of the sound. The high-frequency sound may be attenuated not only because of the impedance mismatch between the air and the absorbent, but as the sound wave passes through the added layer a significant amount of acoustic energy is converted into heat by viscous losses in the interstices. In practice, the amount of heat generated is minute.

It is possible to distinguish between three frequency regions in which different attenuating mechanisms are predominant. For convenience these are described as

TABLE A-10 Comparison of Predicted and Measured Sound Reduction Indices of Three Lined Panels

	Measured or Predicted	Sound Reduction Index, dB							
		63	125	250	500	1000	2000	4000	8000
Panel 1 (flat)	Predicted	14	22	31	39	48	47	65	63
	Measured	20	21	27	38	48	58	67	66
Panel 2 (flat)	Predicted	20	30	40	46	52	60	63	79
	Measured	31	34	35	44	54	63	62	68
Panel 3 (flat)	Predicted	16	19	22	27	36	44	52	56
(corrugated)	Measured	22	24	28	32	38	48	52	52

Panel 1: 1.6 mm flat steel lined with 100 mm thick glass fiber, 49 kg/cu.m density.
Panel 2: 5 mm flat steel lined with 100 mm thick glass fiber, 48 kg/cu.m density.
Panel 3: 2.5 mm corrugated steel lined with 50 mm thick mineral wool, 64 kg/cu.m density.

regions A, C, and B, where A is the low-frequency region, C is the high-frequency region, and B is the transition region. The boundaries between these three regions are defined by the physical characteristics of the absorptive material in terms of the flow resistivity and the material thickness.

The flow resistivity of fibrous absorptive materials is dependent upon the bulk density and fiber diameter by the approximate relationship:

$$R1 = (3.18 \times 10^3)(\rho_m^{1.53}/d^2)$$

The frequency limits of the three regions, A, B, and C, are defined by:

Region A: $101 \prec \lambda_m$
Region B: $101 \succ \lambda_m$, $\alpha l \prec 9$ dB
Region C: $\alpha l \succ 9$ dB

The values of λ_m, the wavelength of the sound inside the absorptive layer, and α, the attenuation constant for the material, can be measured or predicted for semi-rigid materials:

$$\alpha = (\omega/c)(0.189(\rho_0 f/R1)^{-0.595}) \tag{11}$$

$$\lambda_m = (c/f)(1 + 0.0978(\rho_0 f/R1)^{-0.7})^{-1} \tag{12}$$

For an absorptive layer of known thickness and flow resistivity, the attenuation predicted from the equations given above is additive to that produced by the unlined panel.

The predicted and measured acoustic performances of two flat panels and one corrugated panel, each with an absorptive lining, are shown in Table A-10 and Figs. A-24 to A-26.

The predicted performances of the two flat panels are in good agreement with the measured performances over the majority of the frequency range. The largest discrepancies occur at 63 Hz and 8 kHz.

The agreement between the theoretical and measured performances of the corrugated panel is not as good as for the flat panel. The largest discrepancies occur at the lower frequencies with better agreement occurring at high frequencies. This follows the low-frequency trend shown in Fig. A-23 where the corrugated panel was unlined and the predicted performance was less than the measured performance by 5 dB. Nevertheless, the agreement is sufficiently close to support the theoretical model.

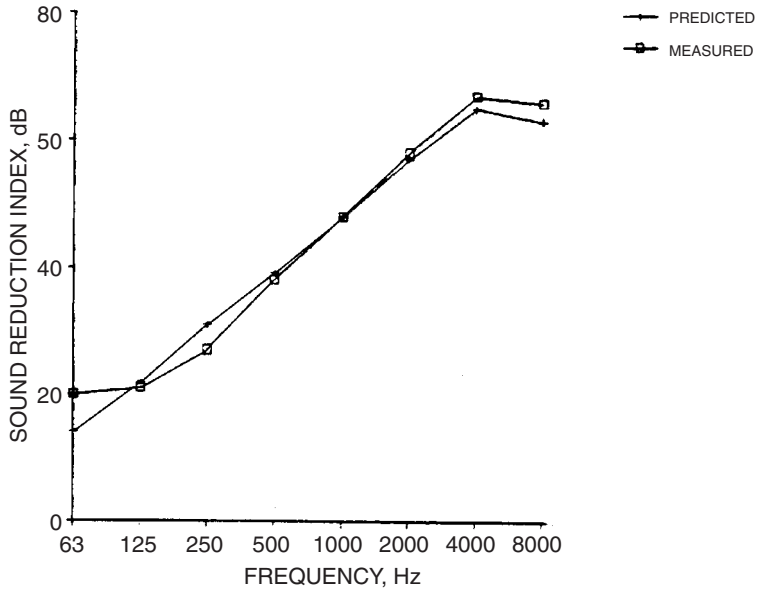


FIG. A-24 Predicted and measured sound reduction index of panel 1. (Source: Altair Filters International Limited.)

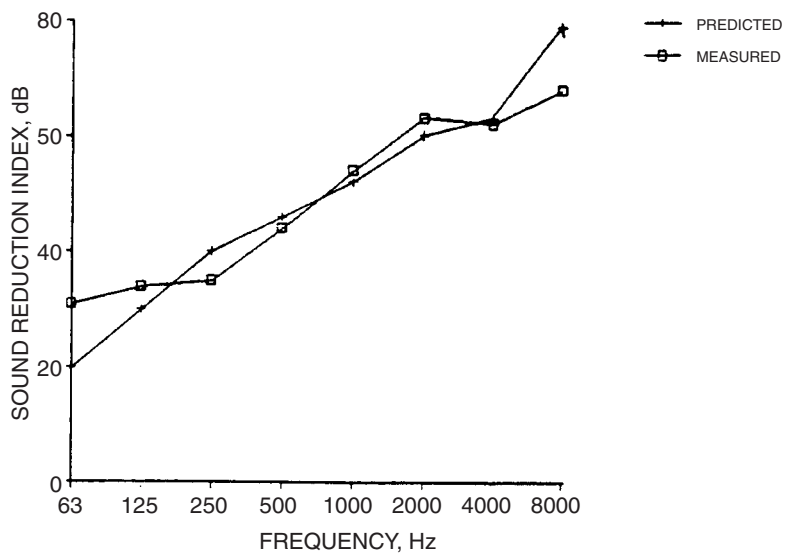


FIG. A-25 Predicted and measured sound reduction index of panel 2. (Source: Altair Filters International Limited.)

Experience thus far. It has been shown that the acoustic performance of lined and unlined panels can be predicted with reasonable accuracy for flat and corrugated panels. It has also been shown that, where noise control is important, unlined corrugated panels are not recommended unless other engineering considerations dictate their use, because corrugated panels are intrinsically less effective as sound insulators than flat panels of the same thickness.

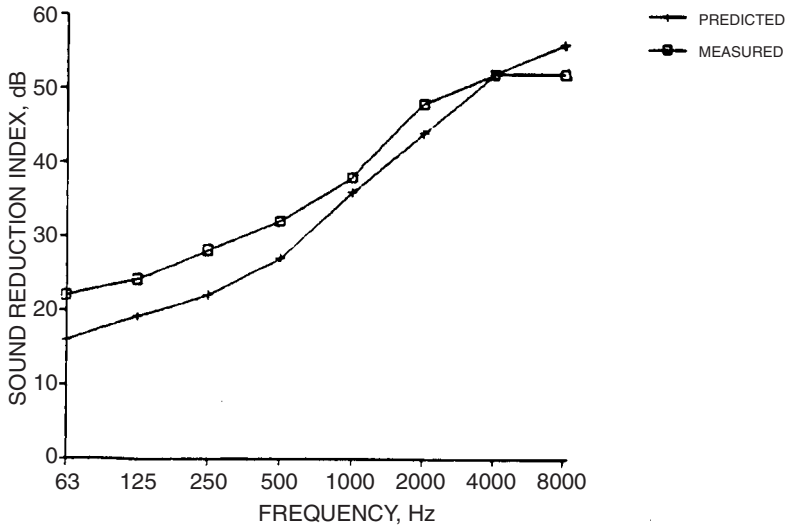


FIG. A-26 Predicted and measured sound reduction index of panel 3. (Source: Altair Filters International Limited.)

A lining of sound absorptive material can substantially increase the sound reduction index of panels and the additional attenuation depends on the density, fiber diameter, and thickness of the lining. By careful selection of these parameters, the acoustic disadvantages of corrugated panels can be considerably reduced so that corrugated panels can be used confidently in situations where noise control is a primary requirement. The additional bending stiffness of corrugated panels permits a thinner outer skin to be employed and reduces the amount of additional bracing required to provide the structural integrity necessary in the demanding environment offshore. This reduction in overall weight compensates for the additional material used in forming the corrugations.

By careful design of the panel, a corrugation profile can be selected, which provided the most cost-effective solution when structural integrity, weight cost, ease of manufacture, and acoustic performance are considered. When expensive materials, such as stainless steel and aluminum, are employed, the reduction in cost by using a thinner-walled corrugated panel can be considerable.

A further consideration is the fire rating of lined corrugated panels. The normal requirement for bulkheads and decks offshore is the "A-60" class division. Corrugated panel designs of the type described here have been submitted to, and approved by, the appropriate authorities.

In some situations where a particularly high acoustic performance is called for, the corrugated design lends itself well to a multilayer construction employing an additional inner layer of heavy impervious material. Cheaper materials are used for the additional septum rather than for the outer skin. The acoustic attenuation of these multilayer designs is comparable to the performance of flat panels employing outer skins of twice the thickness of the corrugated outer skin. Figure A-27 compares the measured performances of a traditional 5-mm-thick flat panel design with a 100-mm-thick absorptive lining and a multilayered panel based on a 2.5-mm-thick corrugated panel lined with a 50-mm absorptive layer.

The nominal surface weights of the two designs are 50 kg/m^2 and 40 kg/m^2 for the flat and corrugated panels, respectively. Except at 63 and 125 Hz, the performance of the two panels is very similar.

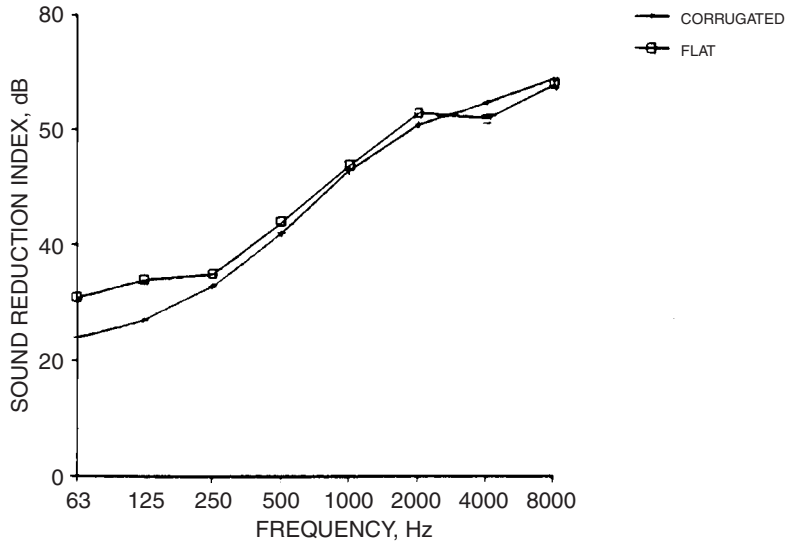


FIG. A-27 Predicted sound reduction indices of two high-performance panels. (Source: Altair Filters International Limited.)

In summary. The acoustic performance of corrugated and flat steel panels can be predicted. The acoustic behavior of corrugated panels is very different from that of flat panels. This means that if corrugated panels are required, careful consideration must be given to the design, since unlined corrugated panels are unsuitable on their own for noise control applications.

However, the greater bending stiffness of corrugated panels offers many financial and structural advantages in the demanding environment that exists offshore, especially for gas turbines.

By lining the interior of a corrugated panel with a material whose physical parameters have been carefully chosen, the inherent acoustic weaknesses can be overcome. Thus a more cost-effective approach to gas turbine enclosure design can be adopted, which considers the structural integrity, weight, cost, ease of manufacture, and acoustic performance. The resultant designs employ less bracing and thinner outer skins to achieve the same acoustic performance as flat-walled constructions weighing typically 25 percent more than the equivalent corrugated design.

Actuators

Actuators, Electrohydraulic

Electrohydraulic actuators are among the more common varieties of actuators in the process plant market and are also more accurate in terms of position control. These components have very specific (to a particular manufacturer) design components. Therefore, terminology in the detailed descriptions that follow is specific to the information source, J.M. Voith GmbH in this case. In the case of requesting competitive bids, the end user should consider requesting similar or alternate features.

Areas of Applications Benefits*

- Reliable and highly accurate conversion of electrical control signals into specific process values.
- Combination of electronics, sensor technology, and mechanics resulting in reduction of interfaces and high degree of reliability.
- High regulated magnetic forces (use of Hall effect) make it possible to apply robust magnetic drives.
- No need for external regulating equipment since complete regulating system is integrated in the chassis of the control unit for the magnetic drive (high degree of EMV resistance).
- Parameters for controller output range can be set from outside.
- High degree of reliability.
- Infinitely variable conversion of input signal i_E into output modes power, pressure, or stroke with high dynamic force.
- Inversion range $E \Leftarrow 0.05\%$.
- Conversion time for 50% regulating value 25 msec.
- Integrated sensor technology and control electronics with function monitoring and actual value remote display output in robust housing or in pressure-resistant casing.
- Electrohydraulic alternative to the retrofitting and modernization of mechanical/hydraulic control and regulatory systems.

Figures A-28 to A-37 and their descriptions outline a typical comprehensive range of electrohydraulic actuators. Different designs may be designated with a specific trademark. This is indicated where relevant.

Aerfoils; Airfoils (*see* Metallurgy; Turbines)

Agitators

Broadly speaking, agitators can be used to produce the following:

1. Uniformity between different components, solid or liquid, miscible, or otherwise. This produces liquid blends or solid suspensions (*see* Gravity Blending in the section on Tanks).
2. Heat or mass transfer between matter. Applications include extraction and leaching processes (*see* Oil Sands).
3. Phase changes in a mixture. Homogenizing, emulsification, and crystallization are among these processes (*see* Centrifuges).

Reference and Additional Reading

1. Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

---

* Source: J.M. Voith GmbH, Germany. Adapted with permission.



Power station applications:

Controlling the power output of gas and steam turbines.



Paper mills:

Controlling process demands steam pressure and grid power.



Chemical and petrochemical plants:

Controlling energy and mass flows.



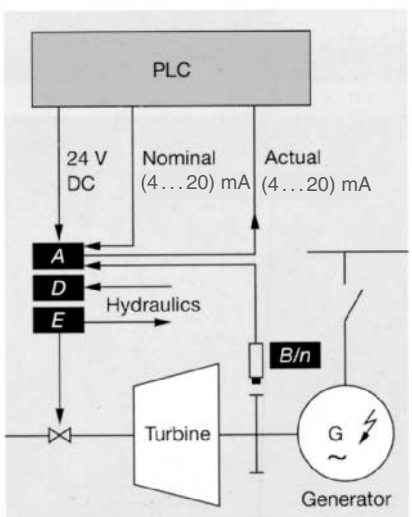
Food processing plants:

Controlling energy flows.

FIG. A-28 Applications of hydraulic actuators by industry and control function. (Source: J.M. Voith GmbH.)

Proportional and dynamic conversion of electrical control signals (0/4 . . . 20) mA into power, regulating pressure, regulating stroke and rpm is achieved with highly versatile modular component technology:

- **A** Control unit: magnetic drive with integrated control electronics in standard and Ex-protection housings.
- **B** Sensor technology: stroke **(B/s)** and rpm sensors **(B/n).**
- **C** Servo valve to control discharge quantities.
- **D** Pressure reduction valves.
- **E** Various operating cylinders.



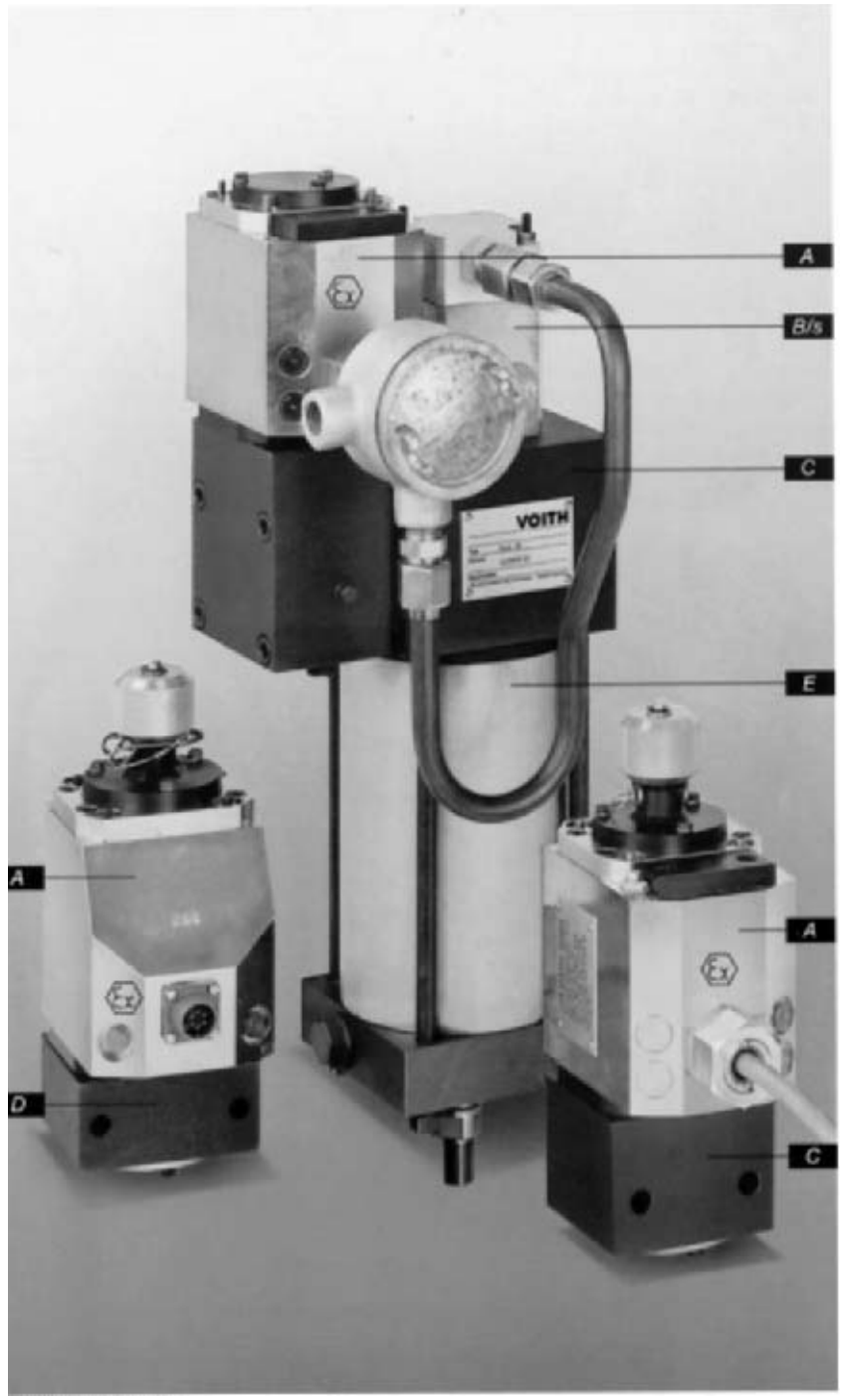


FIG. A-29 Modular component units used for conversion of electrical control signals. (Source: J.M. Voith GmbH.)

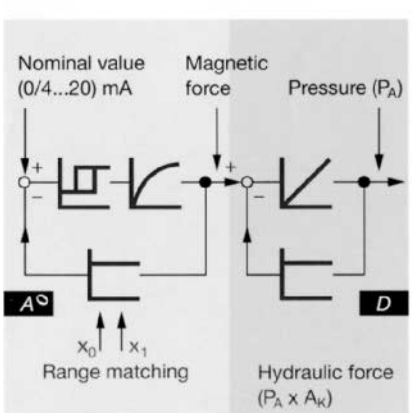
Basic item of the modular assembly is a power-regulated electromagnet consisting of the following components: magnet housing, coil, anchor with driving rod, and the control electronics.

A regulator is used to keep the degree of linear force applied to the anchor at a rating proportional to the input signal.

This ensures that the output (force depending on pressure output) is not affected by adverse influences such as hysteresis, air gap etc (DBP-2019345).

Forces are achieved that are applied directly via the driving rod, i.e., without any hydraulic amplification, to a control piston or a stroke valve.

The diagram below shows how the neutral position magnetic force on one side of the piston is checked by a force proportional to the reduced pressure P_A. The result is a direct and constant match between the input signal and the reduced pressure.



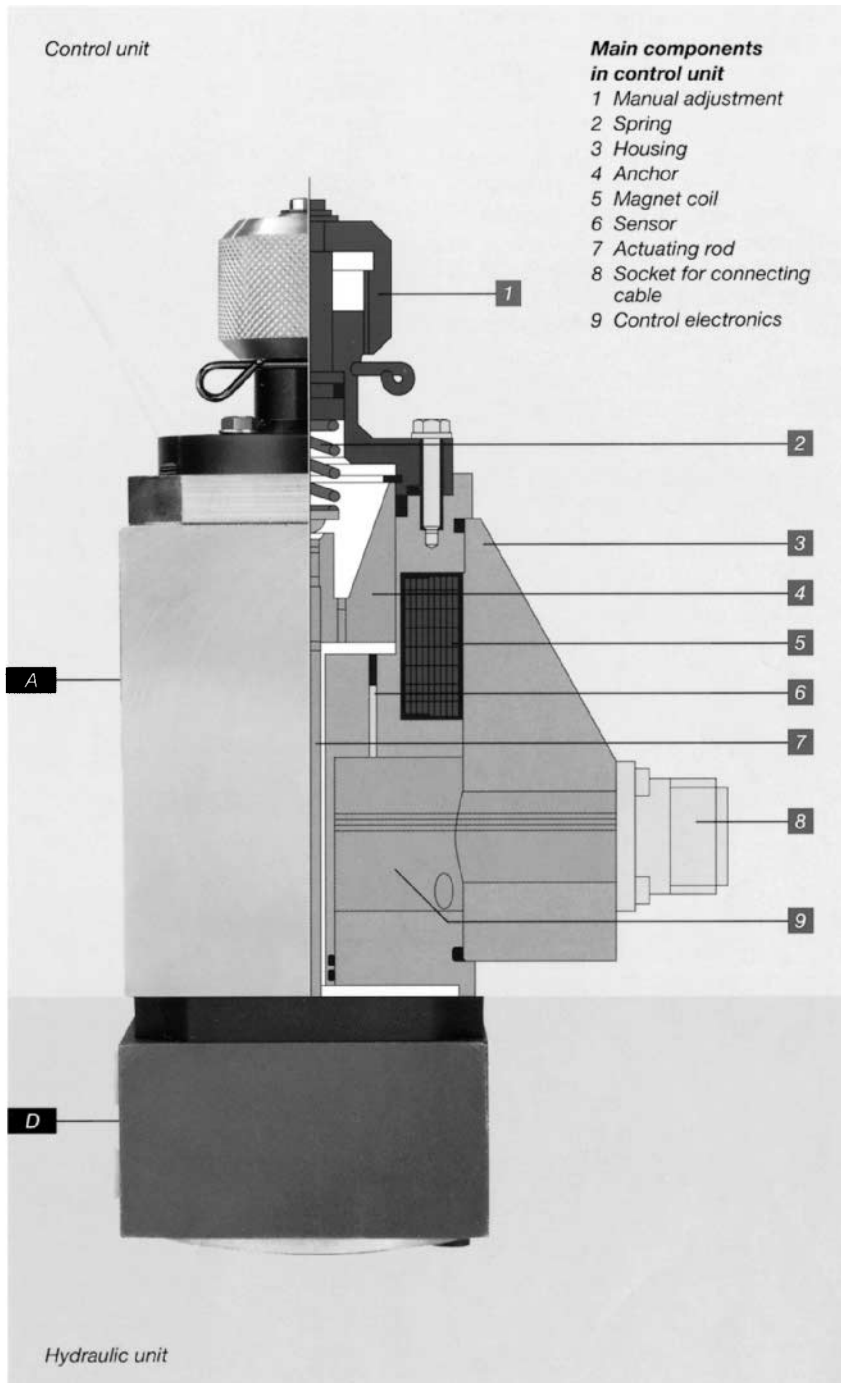


FIG. A-30 How actuators function: power-regulated electromagnet. (Source: J.M. Voith GmbH.)

The section diagram illustrates the main features of control regulator functioning:

- Drive and control pistons with failsafe spring return.
- Internal oil circulation as part of closedown process (rapid closedown ≤ 0.1 sec).
- Inductive stroke pick-up (7) with clamp magnet coupling (8).
- 400 N magnet drive (1) with integrated control electronics for control pistons (3) and position of piston rod (14)

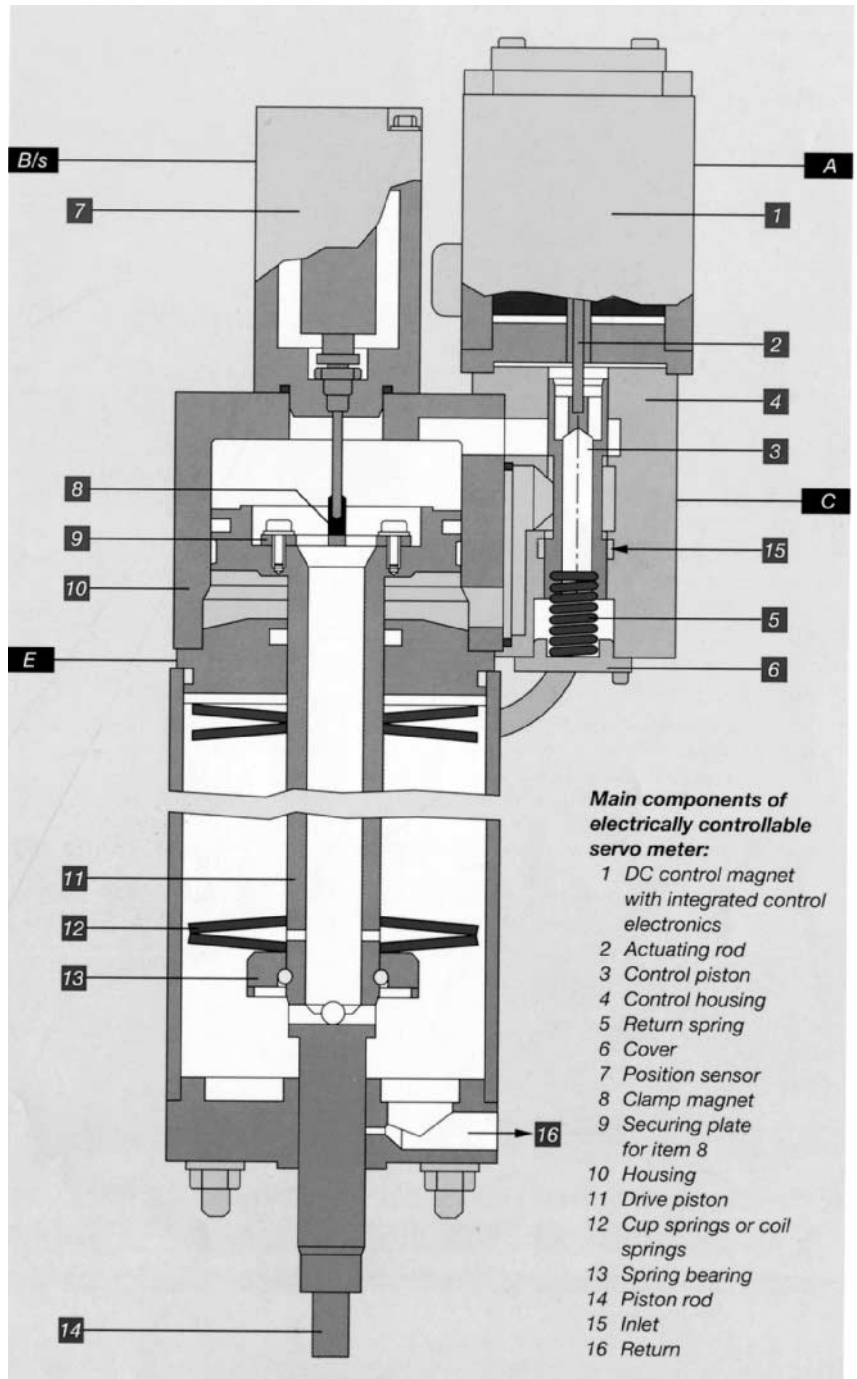


FIG. A-31 Control regulator functioning and main features. (Source: J.M. Voith GmbH.)

Electrically controllable pressure regulating valve (with or without PID controller)

consisting of **A** and **D**.
Converts (0/4…20) mA in (0/7); (0/16); (0/60) bar ($\Delta Q \sim 50 \frac{ltr.}{min.}$)

A directly applied controlled magnetic force F is brought into exact counterbalance with a proportional hydraulic force. The appropriate output pressure in relation to the input signal i_E is controlled by X_0 and X_1. Conversionis effected with a loss of < = 0.1%.

In the version incorporating a PID controller you can compensate for pipeline pressure losses or you can control valve positions and turbine rpm.

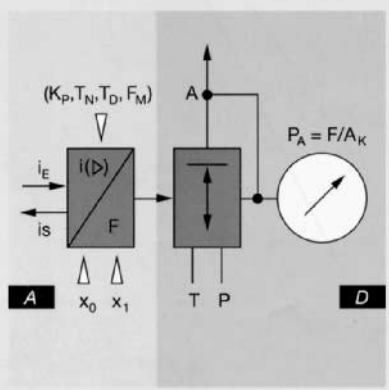


Parameters without integrated PID controller

X_0 = *pressure p_A at $i_E = 0$ or 4 mA*
X_1 = *pressure p_A at $i_E = 20$ mA*

Parameters with integrated PID controller

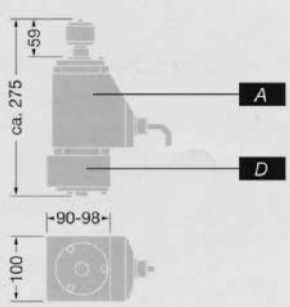
(e.g. for position regulation)
K_P = *proportional amplification*
T_N = *integral action time*
T_D = *deviractive action*
F_M = *gate valve centre*
x_0 = *position of motor at i_E min*
x_1 = *position of motor at i_E max*
is = *remote display (position)*

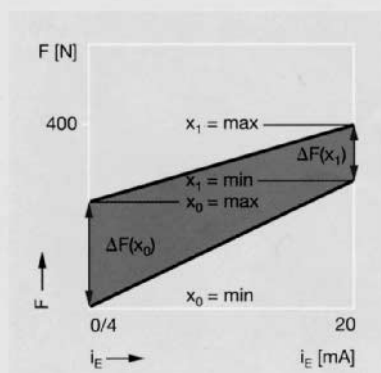


Directly controlled gate valve
(with or without PID controller)

consisting of **A** and **C**.
Controls passage cross-sections (flow control):
(12-20) mA ≙ (0-200) mm² inlet
(12- 0) mA ≙ (0-700) mm² return

With gate valves a controlled magnetic force F is brought into counterbalance with an elastic force, i.e., a dependent force. The input signal i_E is allocated the appropriate cross-section for the valve with X_0 and X_1. The decisive feature is hysteresis-free control in the area around the dydraulic middle position. Symmetrical or asymmetrical controlled cross-sections A can be controlled directly up to 700 mm². The Turcon® CTo version with protection against over-speed rpm is available as a specially adapted gate valve.





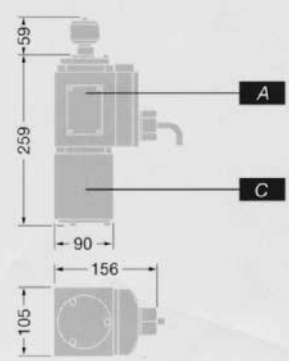


FIG. A-32 Control regulator valves. (Source: J.M. Voith GmbH.)



Electrically controlled regulator

consisting of A, B, C, E.

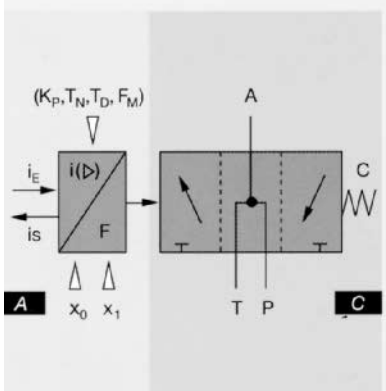
In principle the way that regulators are controlled is via a gate valve for which the magnetic drive has both a magnetic force controller and a superimposed position regulator. The set size of the position regulator acts as the reference value for force control. The input signal i_E—in this example the reference value for regulation—is allocated via X_0 and X_1 the appropriate stroke from the drive piston which is displayed by a (4...20) mA signal. If the control deviation is excessive this is displayedvia a potential-free optocoupler output. In order to linearize flow lines on flaps and valves the control electronics can be enlarged by the addition of a 10-stage function indicator.

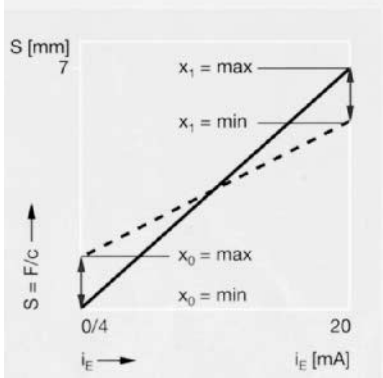


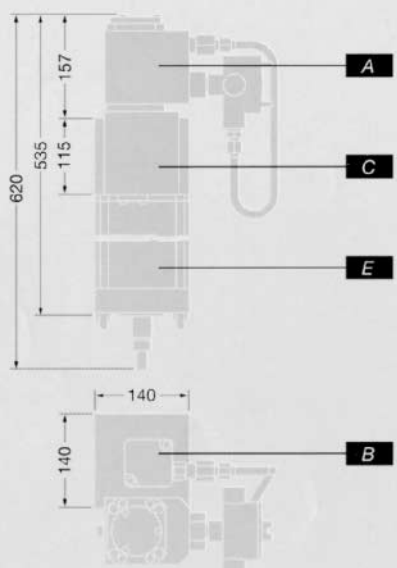
Special technical data

Operating pressure:	6…100 [bar]
Working stroke:	5…60 [mm]
Controlled forces:	500…14000 [N]
Control response times (min):	100 [m/sec]
Positioning accuracy:	15 [µm]

Special models made to order.







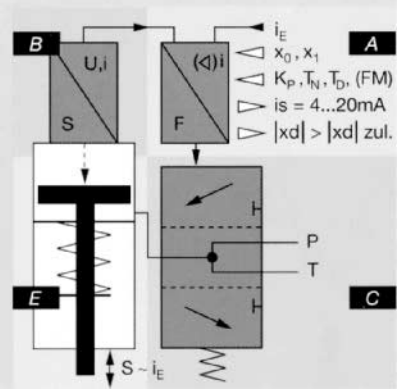


FIG. A-33 Electrically controlled regulator. (Source: J.M. Voith GmbH.)

A

Magnetic drives with integrated control electronics

Input: Signal current (0/4 . . . 20) mA

Output: Magnetic force: Initial value of X_0 adjustable from (0 to 250) N. Final value of X_1 adjustable from initial value up to 400 N.

Hysteresis: ≤ 0.05%

Dynamics: Corresponds to a PT2 component with τ = 15 msec and D = 0.8

Versions available: with and without manual adjustment, with and without Ex-protection, and with and without integrated PID controller for additional, dynamically demanding control and regulation tasks (e.g., position or rpm regulation.)

B/s

Sensor with integrated evaluation/read-out electronics

For strokes of (0 to 40) and (0 to 60) mm in pressure-resistant enclosed casing as per EEx d with short cable lead and standard version supplied with 3-pole plug.

A + D

Magnetic drive + pressure reducing valve = pressure regulating valve

This combination converts (0/4 . . . 20) mA into (0 to 7), (0 to 16) or (0 to 60) bar. Respective control piston diameter readings are: 26, 18 and 10 mm.

A + C

Magnetic drive + control valve (3/3 or 4/3) = gate valve

The control valve is at the hydraulic centre (adjustable using magnetic force) at about 12 mA. An increase to 20 mA moves the valve against a spring force to the inlet fully open position (approx. 200 mm^2). A decrease to (0/4) mA brings it to the return fully open position (approx. 700 mm^2).

A + B + C + E

Magnetic drive with PID controller + position sensor + control valve + regulating cylinder with inbuilt return spring = regulator

For converting (0/4 . . . 20) mA into controlled strokes of (0 to 30), (0 to 60) mm. Flow forces in open directionca. 15000 N. Spring forces in closedirection ca. 9000 N. Time taken from open to close ≤ 0.10 sec.

FIG. A-34a Drive/control valve options. (Source: J.M. Voith GmbH.)

Technical data for all electronic component assemblies

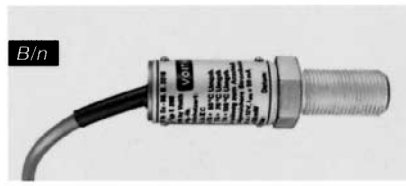
Operating voltage U_B	(20 ≤ U_B ≤ 30) V
Operating current i_B	approx. 0.5 A (up to 2.4 A when switching-on)
Surface temperature T	(-40 ≤ T ≤ 85)°C
Storage temperature T_L	(-60 ≤ T_L ≤ 150)°C
Type of protection	DIN 40050 IP 65
Protection against explosion as per PTB-No. EX-90.C.1065	EEx d IIC T4 (T5)*
EX-connection	Short cable lead or EExe distributor
Standard connection	Mil-plug (6 pole)
Impact resistance	18 impulses of 50 g over 0.005 secs.
Vibration	18 to 500 Hz at 4.2g
Humidity	100% condensing

* as per US standard currently being prepared.

FIG. A-34b Technical data for electronic component assemblies. (Source: J.M. Voith GmbH.)

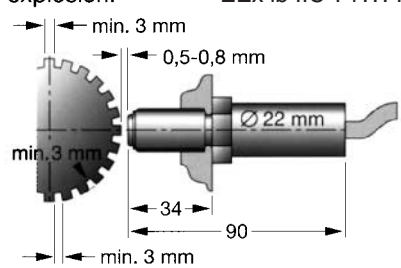
Sensors that are encased in robust EEx d housings operate more reliably when used for constant measurement of the rpm and valve positions in compressors and gas and steam turbines. This type of housing protects them from adverse environmental factors (EMV, temperature changes, humidity, and oscillations).

RPM sensors



Technical data Type E 288
Draw-No. 43.9423. xx

Measurement cog module:	≥ 2.5
Cog shape:	any – square profile preferred
Distance between sensor and cog:	0.5 . . . 0.8 mm
Measurement frequency:	25 Hz . . . 15 KHz
Operating temperature:	–20 . . . +125°C
Type of protection as per DIN 40050:	IP 65
Protection against explosion:	EEx ib IIC T4 . . . T6



Technical data of sensors 90.1265.xx and 90.3131.xx

As above, but without the impulse ammplifier and without Ex-protection*.These sensors have a coil resistanceof ca. 1.1 kΩ at 25°C and areauthorized to operate at temperaturesof up to 150°C.

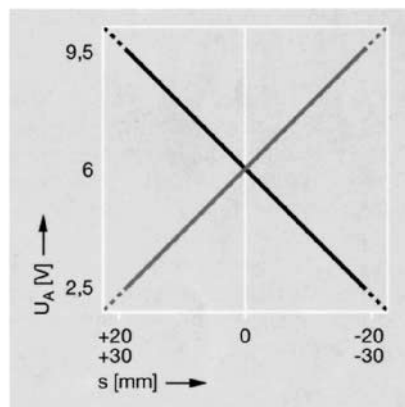


90.1265.xx 90.3131.xx

*) *EEx d IIC T5 – version available shortly.*

Position sensors





Technical data

Supply voltage:	18 . . . 30 V/DC
Measurement range:	Type 40 ± 20 mm Type 60 ± 30 mm
Output voltage for rising or falling trace:	see diagram
Power consumption max.:	50 mA
Pressure resistance (static):	350 bar
Type of protection as per DIN 40050:	IP 65
Protection against explosion:	EEx d IIC T5

FIG. A-35 How sensor technology works. (Source: J.M. Voith GmbH.)

Turcon® CTo is an electrohydraulic rapid trigger shut-off device being a special version of the gate valve. Turcon® CTo is used mainly as a monitoring system resp. as an rpm protection device for high-rpm steam turbines. It offers major advantages compared with the mechanical-hydraulic systems available hitherto:

- Very high degree of reliability as a result of constant trigger speeds independent of oil temperature, electromagnetic direct drive with failsafe return/reset. The rapid trigger device can be tested without interrupting operations by means of 10 msec test impulse using trigger frequency generator (integrated in magnetic system).
- Easy to install and service: the magnetic valve and electronics are all one unit. Alignment takes place at the factory, removing the need for labor-intensive adjustments during commissioning.
- High reliability as a result of robust construction and low pressure hydraulics.
- Contact-free speed readings, avoiding strain on main bearings from load displacement in case of overspeed rpm. This makes the device suitable for turbines with extremely high rpm ratings.
- discontinuity in wire and/or short circuit at speed sensor.
- Housing temperature $\geq 85°C$.
- Supply voltage ... $\geq$ (30 ... 33) V/DC.
- Irregular loss of speed -tripping speed (n_{trip}) $\geq (1.03 \cdot n_{trip})$ (10 msec-Test) $\geq (n_{trip} \pm 0.5\%)$.
- External switchoff (activation of standby power circuit).

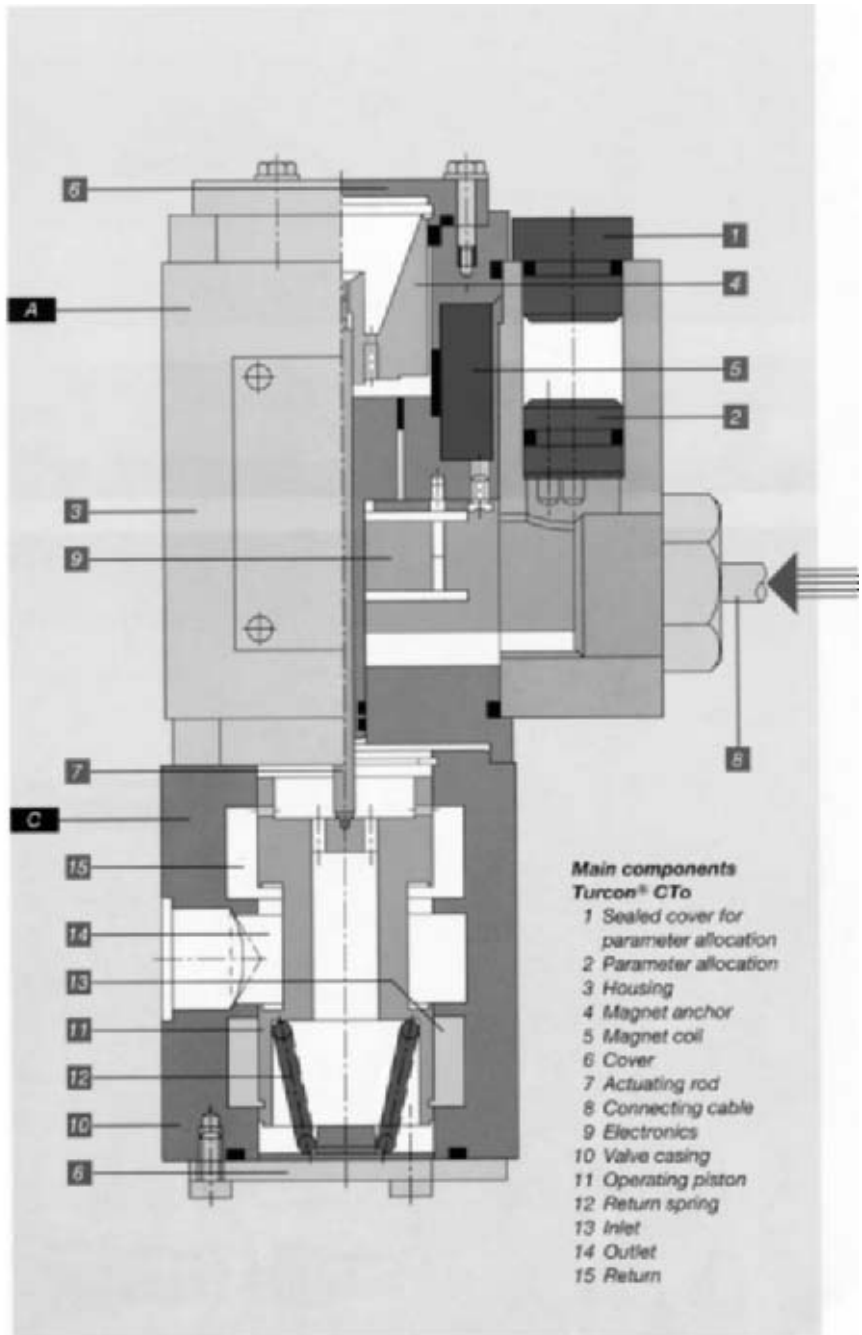


FIG. A-36 Speed protection device operation. (Source: J.M. Voith GmbH.)

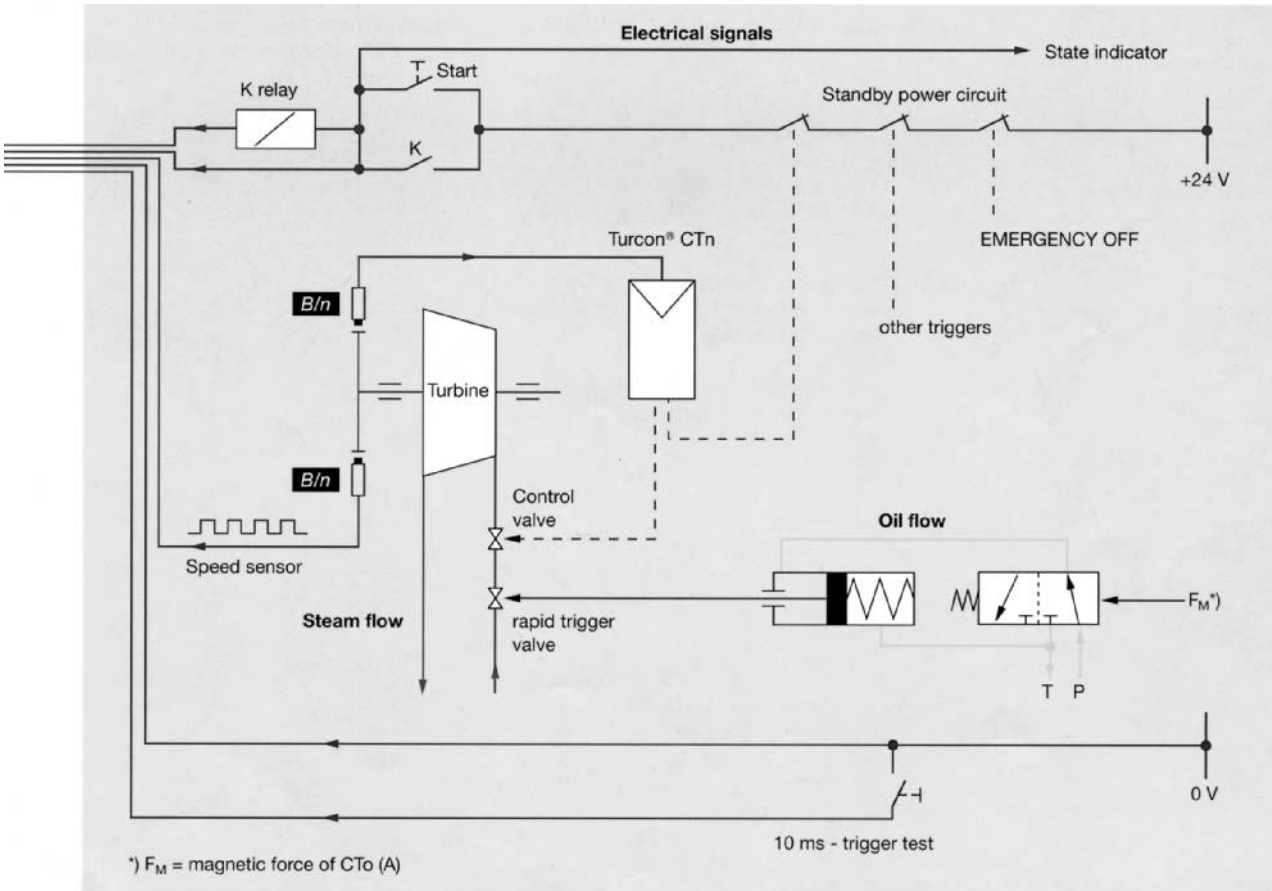


FIG. A-37 Trigger criteria for protection against overspeed. (Source: J.M. Voith GmbH.)

Agriculture; Agricultural Product Processing

(*See also* Ecological Parks; Environmental Accountability; Forest Products; *other related topics*.)

Agriculture is too wide a field to be dealt with comprehensively in this book. However, many generic types of equipment used in this field are discussed, including centrifuges, conveyors, pumps, motors, chillers, and so forth. Options, such as specific material selections, for instance, plastic gears and lobes (instead of metal ones) in pumps handling food, may alter overall designs. Agricultural product machinery is often custom designed or has customized options for this reason. Certain machinery types most commonly used in, if not unique to, the agricultural industry have been essentially left out of this book. These types include: pelletizers (such as might be used for making food pellets), briquette makers, and homogenizers (for milk for instance).

Although agricultural machinery might be simpler than, say, machinery used in a modern plastics plant, there is a growing sophistication with all forms of the process industry, such as coolers in agriculture. *See* Cooling.

Agricultural products that support the industry are similarly frequently custom designed or specified, such as the insecticides made for agricultural crops. There is no uniformity in quantities used for application either. For instance, 80% of all the agricultural insecticides used on crops in the United States are used on the cotton plant.

One indicator targeted for optimization, for the process engineer handling agricultural products, is reduced chemical pollutants that originate from a process. The potential for decreasing chemical pollutant levels in product handling increases with technical developments. Methods for reducing these levels are frequently provided by biological engineering means, including:

- Bioremediation of polluted soil
- Use of naturally occurring pesticides instead of chemical pesticides
- Breeding plants/crops with characteristics that enhance production without further chemical use

Examples of such technology include the ability to develop grazing grass and crops with an aerated root system that will resist drought, floods, and also potentially neutralize toxic mineral compounds by oxidizing them.

Firms in the agricultural industry are excellent candidates for joining ecological industrial parks. They must have the highest standards of cleanliness and have a great deal to offer a group of industries in terms of experience in this area. If a metal workshop and industrial furnace can coexist in the proximity of a milk homogenizing facility, health conditions for all will improve and pollutants, overall, will drop. If agricultural firms can thus convey the environmental practices they must abide by, the industry as a whole, and the conditions under which they must work, will automatically improve.

References and Additional Reading

1. Soares, C. M. *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.
2. Comis, D., "Miracle Plants Withstand Flood and Drought," *The World and I*, February 1998.

Air Filtration; Air Inlet Filtration for Gas Turbines

One of the most common applications of air filtration in a process engineer's world is filters at the air intake of a gas turbine. These filters take a toll on the gas turbine's thermal efficiency and therefore increase the turbine's fuel consumption, so their designers make every attempt to minimize pressure drop across the filter elements.

Industries and applications where these filters are used include refineries and chemical plants, the food industry, compressors, power stations, electrical generators, warehouse and building air-conditioning systems, as well as computers and electrical cabinets.

Purposes for installing gas turbine air-inlet filtration include

- Prevention or protection against icing
- Reduction/elimination of ingestion of insects, sand, oil fumes, and other atmospheric pollutants

Potential ice-ingestion problems can be avoided with a pulse-jet–type filter, commonly called a *huff and puff* design. Ice builds up on individual filter elements

that are part of the overall filter. When a predetermined pressure drop is reached across the elements, a charge of air is directed through the filter elements and against the gas turbine intake flow direction. The ice (or dust "cake") then falls off and starts to build up once more.

Ice ingestion has caused disastrous failures on gas turbines. A few companies also make instrumentation that will detect incipient ice formation by measurement of physical parameters at the turbine air inlet. Sometimes, if the pulse-type filter is retrofitted, the anti-icing detection instrumentation may already be there. The pulse filter, however, provides a preventive "cure" that will work, regardless of whether the icing-detection instrumentation is accurate, as the cleaning pulse is triggered by a signal that depends on differential pressure drop across the filter elements. If a pulse filter is used, icing-detection instrumentation, which is normally necessary in a system that directs hot compressor air (bleed air) into the inlet airstream, is not required.

Many filtration applications are examples of retrofit engineering or reengineering because the original application may have been designed and commissioned without filters or the original choice of filters/filter elements was inappropriate. For tropical applications, filter media that swells or degrades (also rain must not be allowed to enter the filter system) cannot be used because of the intense humidity. This will exclude cellulose media. Tropical installations present among the most severe applications.

Inlet Air Filters for the Tropical Environment*

The factors that determine design include the following:

Rainfall

The tropics extend for 23°28′ either side of the equator stretching from the tropic of Cancer in the north to the tropic of Capricorn in the south and represent the tilt of the earth's axis relative to the path around the sun. The sun will pass overhead twice in a year, passing the equator on June 21 on its travel north and September 23 on its travel south. The sun's rays will pass perpendicular to the earth's atmosphere and so will have the least amount of filtration, giving high levels of ultraviolet rays.

The area has little seasonal variation; however, the main characteristic of the area is the pronounced periods of rainfall. Typhoons and cyclones are common to certain parts of this area.

It is not surprising that the records for the highest rainfall ever recorded are all within the tropics. Intense rainfall is difficult to measure since its maximum intensity only lasts for a few minutes. Rainfall can be expressed in many ways, either as the precipitation that has fallen within 1 hour (in millimeters per) or over shorter or longer periods but all relating back to that same unit of measurement. Since gas turbines experience problems due to rainfall within a few minutes, it is important to take account of the values of "instantaneous rainfall" that can occur.

The most intense rainfall ever recorded was in Barst, Guadeloupe (latitude 16°N), on November 26, 1970, when 38.1 mm fell in just 1 min.

Another important feature regarding rainfall is the effect of wind speed. "Horizontal rain" is often described, but in practice is unlikely to occur. However, wind speeds can give rain droplets significant horizontal components. The impact

---

* Source: Altair Filters International Limited, UK. Adapted with permission.

of this can be very important, particularly with small droplet sizes. Even droplets 5 mm in diameter will cause a vertical surface to be almost 4 times wetter in wind speeds of 37 m/s than on the horizontal surface (to which the rainfall rates relate).

The effect of rainfall in tropical environments on the operation of gas turbines has been very much underrated.

The humid environment also ensures that relative humidities are generally high, with the lowest humidities being experienced during the hottest part of the day and the highest occurring at night. During the rainy season, the humidity tends to remain constant throughout the day. The effect of humidity is important where airborne salt is concerned since salt can become dry if the humidity is below 70%.

Dust

Dust levels in tropical environments in southeast Asia are generally low.

There are, of course, always specific exceptions to this, for example, near new construction sites or by unpaved roads. But, in general, dust is not a significant problem. Mother nature ensures this by casting her seeds on the fertile soil and quickly turning any unused open space into a mass of overgrown vegetation very quickly, thereby suppressing the dust in the most natural way.

Insects and moths

In the tropics the hot, humid environment is a natural encouragement to growth of all kinds. It is often said that if a walking stick is stuck into the rich fertile soil of the area and left for 3 months, it will sprout leaves and grow. Certainly the insect population reflects this both in size and quantity. Large moths are common to the area and tend to occur in quantity during specific breeding periods. These can quickly cover intake grills, obstructing airflow and even causing large gas turbines to trip. Some of the largest moths are found in the tropics. A common moth in India and southeast Asia is the Swift moth (Hepralidae), which can have a wing span of some 15 cm and is said to lay up to 1200 eggs in one night. Another moth is the Homoprera shown in Fig. A-38, which has a similar wing span. Moths are attracted by the lights that often surround the turbine installations, as well as the airflow, which acts as a great vacuum cleaner.

On one installation in Sumatra, large gas turbines have been known to trip out after only 8 hours of operation due to blockage of the air filters with moths.

Fortunately, moths tend to confine themselves to within a few miles of land and so offshore installations do not tend to suffer these problems.

Problems Experienced

Many feel that standardization is the key to reducing costs and boosting profits. It is not surprising, therefore, that gas turbine air-filter systems were designed with this in mind.

Dust was important to system designers, and so filter systems may be chosen to be able to deal with prodigious amounts of it, whereas, in practice, dust is only normally a problem next to unpaved roads or construction sites.

Despite this, most gas turbines were fitted with elaborate and expensive solutions to overcome a problem that hardly existed, or at least only in a relatively small percentage of installations. Many of these systems employ bleed fans that need additional electrical energy and a constant maintenance requirement. A typical system is shown in Fig. A-39. This system employs spin tubes that swirl dust to the outside of the tube where a bleed slot extracts the dust while allowing the

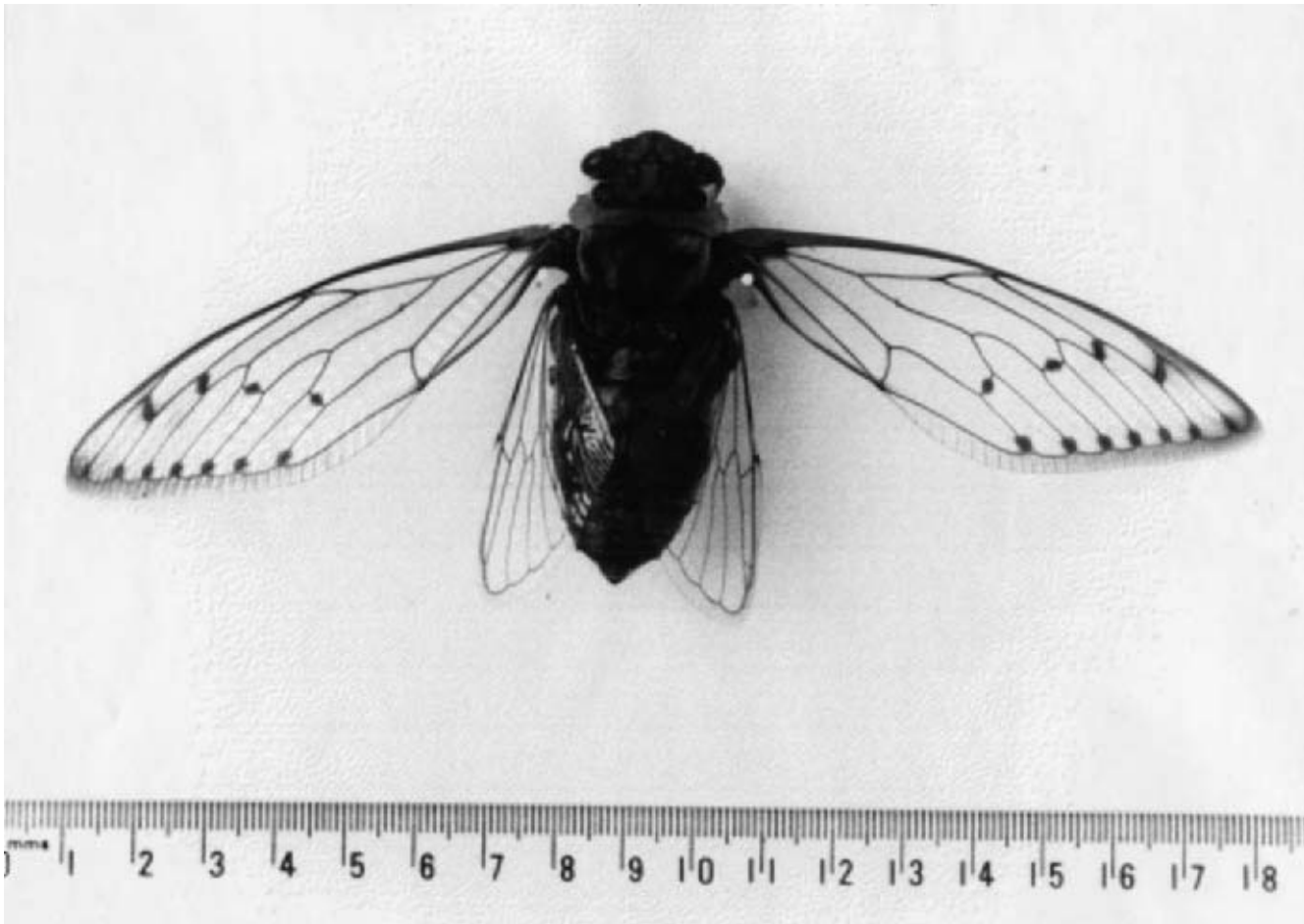


FIG. A-38 Homoprera (insect type). (Source: Altair Filters International Limited.)

cleaner air to pass through the main core of the tube. Since the air is rotated, the peripheral speeds need to be high, which, in turn, results in a relatively high pressure-loss coefficient. The efficiency of the system is very reliant on the bleed air, which is provided by auxiliary fans.

Another bleed extract system uses a series of convergent vanes that funnel the air toward a central slot through which bleed air is extracted. The heavier dust particles are guided toward the bleed extract, while the main air passes between the vanes at almost 180° to the general direction of airflow. Again, the efficiency of the system is reliant on the provision of bleed air.

Protection against rain was elementary. On many systems that had dust-extract systems, no further provision was made. On others a coarse weather louvre, often of plastic, was provided, as shown in Fig. A-40. Sometimes a partial weatherhood was provided, sometimes not.

The emphasis on the designer was to provide a "three-stage" filter system, without worrying too much about the suitability of those stages.

There was recognition of the high humidities that exist, and so most systems incorporated a coalescer, whose function was to coalesce small aerosol droplets into larger ones, which could then be drained away. These coalescer panels varied

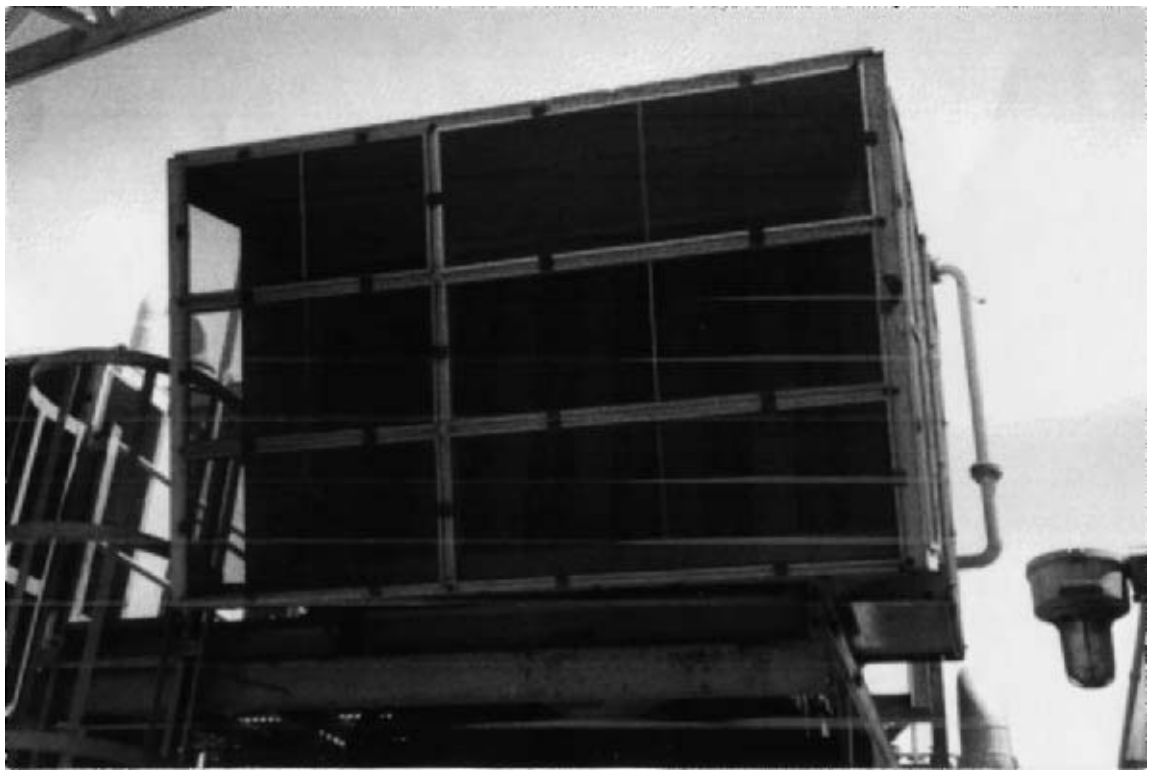


FIG. A-39 A typical spin-tube inertia filter. (Source: Altair Filters International Limited.)

between pieces of knitted mesh wound between bars within the filter housing to separate panels with their own framing. Loose glass fiber pads were used as an inexpensive solution and also served as a prefilter pad.

The final stage in almost all of these systems was the high-efficiency filter element, either as a cartridge or as a bag. The high-efficiency cartridge was typically a deep-pleated glass-fiber paper sealed in its own frame and with a seal on its rear face. A typical example is shown in Fig. A-41. Other high-efficiency filters employed glass-fiber pockets that fitted into permanent wire baskets within the filter house.

Almost all of the filter systems were enclosed in housings constructed in carbon steel, finished with a variety of paint finishes. Other materials, such as stainless steel, were not common since their initial cost was thought to be excessive.

Protection against complete filter blockage was often provided by means of a bypass door. This normally was a counterbalanced door in which a weight held the door closed. When the pressure drop across the filter was high, this overcame the force exerted by the balance weight and the door opened, thereby bypassing the filter system with unfiltered air. A typical system is shown on the top of the filter housing shown in Fig. A-40.

Operational Experience

The ultimate test of the filter system is whether it is providing the required protection to the gas turbine. Unfortunately, there are mainly instances where problems have been found. The following figures illustrate problems that have arisen with this information source's designs, as installed in the field:

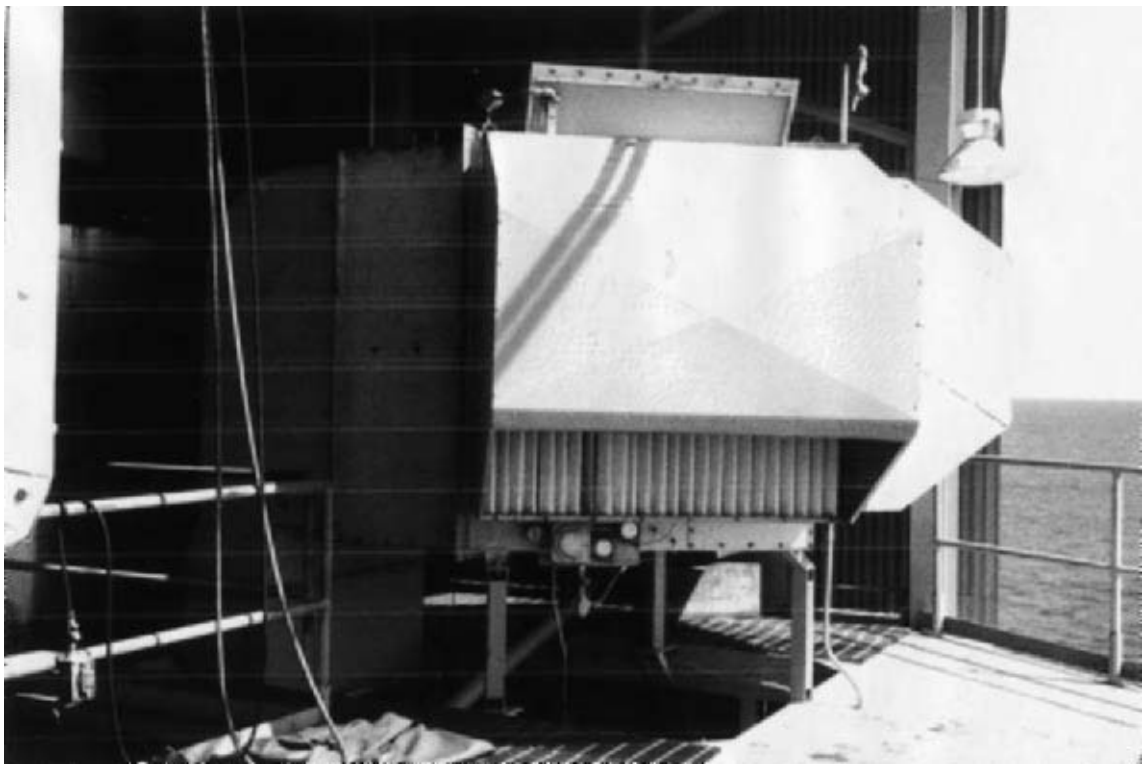


FIG. A-40 A filter housing with weather louvres and a bypass door. (Source: Altair Filters International Limited.)

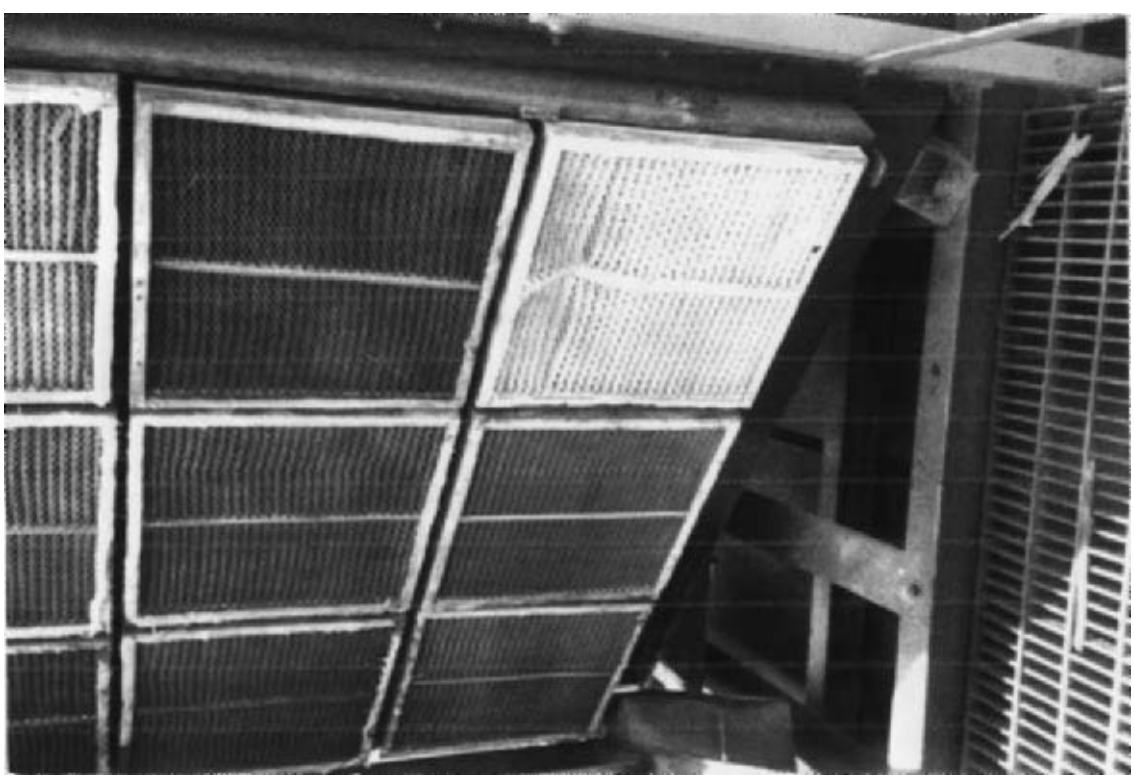


FIG. A-41 Typical high-efficiency cartridges. (Source: Altair Filters International Limited.)



FIG. A-42 A fouled compressor. (Source: Altair Filters International Limited.)

Fig. A-42 A fouled compressor rotor from an engine in Brunei.

Fig. A-43 Turbine corrosion from an engine in India.

Fig. A-44 Compressor fouling and corrosion from an engine in Indonesia.

Fig. A-45 Turbine blade failure from an engine in Indonesia.

Fig. A-46 Debris in an engine compressor in Brunei.

In addition to these visual indications, there are many instances where engine overhaul cost has soared because the installed filter system was ineffective. The main problems can be categorized as follows:

- Design suitability
- Material suitability
- Maintenance

Design suitability

The main problem with most of the systems was that they were incapable of dealing with the rainstorms that frequent the area. As previously shown, these storms are much more severe than in the more temperate parts of the world where the systems are normally designed.

In general, it was found that even where weatherhoods were fitted, they did not provide adequate protection. In Fig. A-40 the air immediately contracts and turns through 90° within a very short distance of the filter section. In addition, the weatherhoods induce an upward inlet airflow, with the result that the majority of

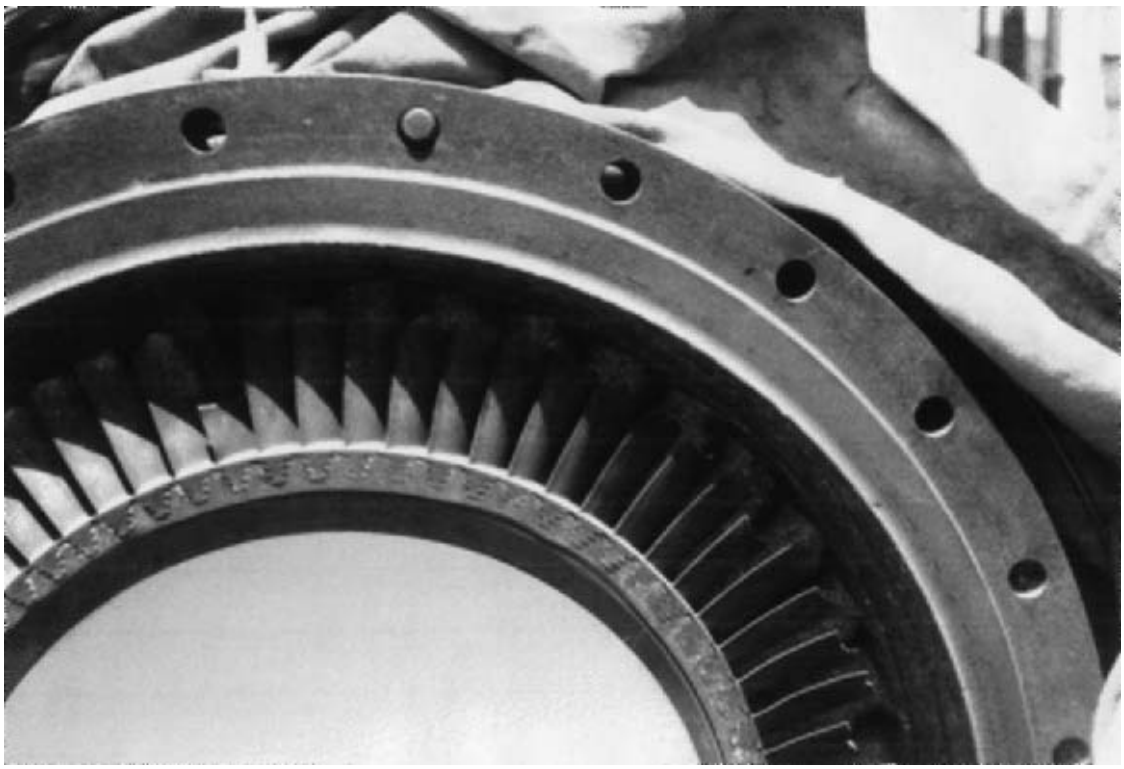


FIG. A-43 Turbine corrosion. (Source: Altair Filters International Limited.)

the airflow is concentrated at the lower half of the filter face, increasing the local velocity. Any water that is caught by the weather louvres drains vertically downward right into the area where the high velocities exist, with the result that the water is reentrained off the vanes into the filters downstream. This situation is further worsened as no drains were provided anywhere in the housing, including the weather louvre section. So even if rain had been caught by the weather louvres, it would have inevitably passed downstream.

The weather louvres that were mostly fitted were a low-efficiency type, with a large louvre pitch. Most were of a plastic construction that tends to embrittle and fracture with time. Their effectiveness against rain is so low that many operators were forced to shield them in some way. Figure A-47 shows a typical modification while Fig. A-48 shows a more elaborate protection. The latter solution is a very unwelcome compromise since the enclosed canopy is a potential gas trap.

On some of the more recent installations of this kind an attempt was made to provide drainage. Synthetic rubber dump valves were used, which rely on the weight of a column of water opening up a slit in the bottom of the valve. Unfortunately, the atmospheric pollution and the high-humidity climate tends to glue the valve openings. It is only when they are manually opened that the valve will discharge the water (Fig. A-49); at other times water will be reentrained into the filters.

Although great attention has been paid to designing systems that can remove large quantities of dust, in practice only a small percentage of sites in this climate can be regarded as dusty. The inclusion of inertial-type systems is unnecessary and too expensive. In addition, it actually worsens the already poor rain protection.



FIG. A-44 Fouling and corrosion. (Source: Altair Filters International Limited.)

Spin tubes, that employ high tangential air velocities, will shatter any incoming rain droplets into a fine mist, which then penetrates the remainder of the filter system. Similarly, the other inertial dust systems will do the same, although to a lesser extent. Any water that passes into the high-efficiency stage of the filters is potentially very serious, since it can wash contaminants through the filter and into the engine. If the installation is coastal or offshore, there is a real risk that sea salt may pass straight into the engine. Under these conditions the filter is a potential "salt bomb," waiting to explode as soon as the weather changes.

It has now been recognized that, in offshore or coastal environments, special attention is needed to prevent salt ingress. It is not sufficient just to provide a high-efficiency filter system without understanding the behavior of salt and making allowance for it. Salt, more commonly sodium chloride, will absorb water from the atmosphere under certain conditions of relative humidity. A seawater droplet will dry out at relative humidities of 40% and lower, but will absorb water from the air at high humidity. Therefore, if salt is caught on a filter and the humidity rises, the salt may absorb water and droplets will form that can then reentrain into the airstream and into the engine with disastrous results.

It was a combination of poor rain protection and salt carryover that caused disastrous problems to two Avon engines operating in offshore Brunei. Water and

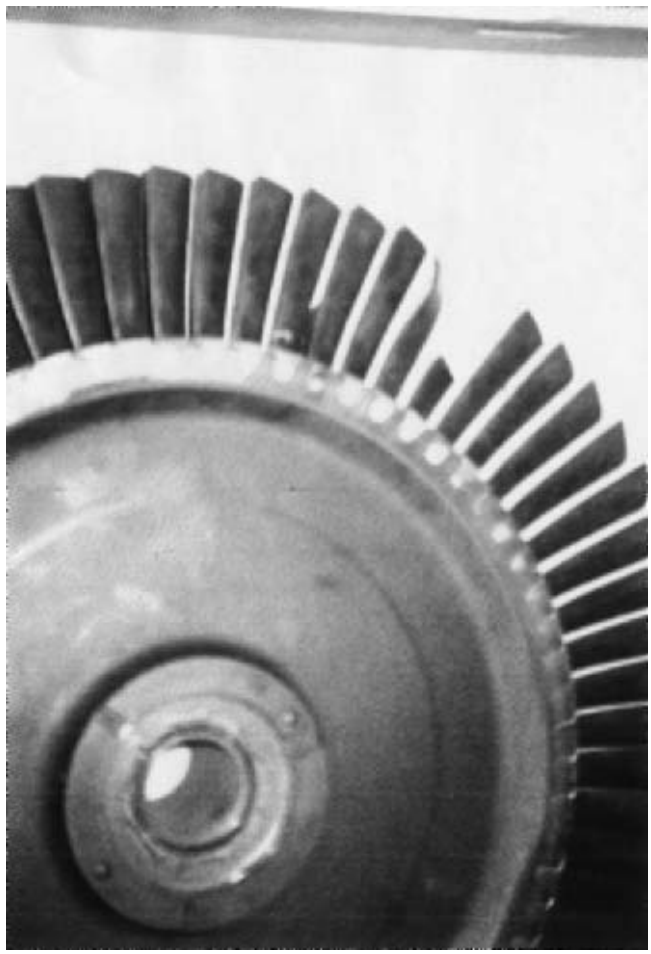


FIG. A-45 Turbine damage. (Source: Altair Filters International Limited.)

salt leached through the final-stage, high-efficiency cartridge filters (Figs. A-50 and A-51) and were carried into the downstream ducting and onward into the engine, causing massive compressor and turbine corrosion.

This situation was further exacerbated by the choice of materials for the inlet system. While the filter housing was constructed of stainless steel, the downstream silencer and plenum used a stainless-steel perforated lining for acoustic attenuation, welded to a carbon steel base frame.

The entrained water soaked through the perforated steel into the acoustic lining where it acted as a perfect electrolyte to assist the galvanic corrosion which can take place between even relatively similar metals such as carbon steel and stainless steel.

Severe inlet corrosion was experienced (Figs. A-52 to A-54) to such a degree that in places the perforated sheet became detached from the base frame (Fig. A-55). This released considerable amounts of debris into the engine, causing even further damage.

The whole inlet systems are now being replaced, which will involve a two-week shutdown on each of these turbines, with the associated loss in production. In addition, the overall costs of one turbine alone was in the region of US $600,000. An expensive exercise indeed.



FIG. A-46 Debris in a compressor. (Source: Altair Filters International Limited.)

Corrosion in the tropical environment is widespread. The initial use of carbon steel gives a limited life, particularly offshore. Figure A-56 shows the remains of an inertial filter still in service and faithfully painted by the operators. Figure A-57 shows the bottom of a corroded weather louvre section that indicates lack of damage. Figures A-58 and A-59 show severe corrosion of ducting downstream of the air filters, in which the provision of air filters becomes meaningless with such large air bypass routes. Figure A-60 shows the corroded duct downstream of the air filters, with surface debris sitting awaiting ingestion into the turbine. Clearly, corrosion is a massive problem.

The provision of filter bypass doors is a mixed blessing. These doors are notoriously difficult to seal, and are a potential area where all the money spent on a good filter system can be wasted. They are designed, of course, to help maintain production and are, at times, hazardous to the health of the turbine. A typical

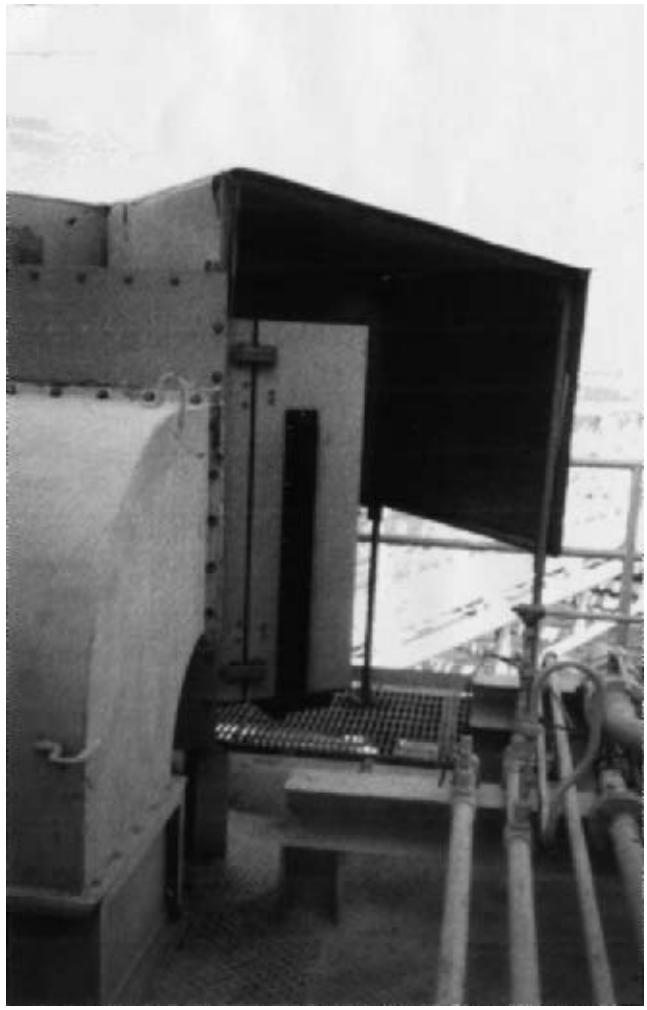


FIG. A-47 Modified weather protection. (Source: Altair Filters International Limited.)

example is when shot-blasting work is undertaken nearby. The filters quickly became blocked and the bypass doors opened, allowing the very erosive shot-blast material into the engine.

Modern engine-control systems now make bypass doors obsolete. A double-pressure-switch arrangement is used, one set to give an alarm and another set at a higher depression to initiate turbine shutdown. Occasionally, a third pressure switch is added (normally set for shutdown) but linked in so that two of the three pressure switches must activate before shutdown is initiated. This helps to prevent a spurious trip due to instrument error.

Painted carbon steel needs constant maintenance and, increasingly, sophisticated paint systems that ensure that the cost difference with stainless steel is reducing all the time. Figure A-61 shows a recently installed filter house where the initial paint has not been applied correctly, resulting in large bubbles on the metal surface. Figure A-62 shows part of a new carbon-steel filter housing awaiting installation, while Fig. A-63 shows that corrosion has already started.

It is not uncommon for filter housings to require repainting every two years to keep them in good condition. The painting process brings all the horrors of shot



FIG. A-48 Filters protected by a canopy. (Source: Altair Filters International Limited.)

blasting. The lifetime maintenance cost of a painted carbon-steel structure vastly outweighs the small extra premium that stainless steel affords for the initial installation.

Newer Filtration Designs

Clearly, the most worrisome deficiency of the old systems was the combination of poor water-handling ability either from rain or salt-related droplets and the materials of construction.

Gas turbines that operate on warships also experience intermittent heavy-water loadings. These are protected by vane separators, which are, in essence, a high-efficiency weather louvre. The main differences are the close vane pitching, which is 3 to 4 times closer than for weather louvres, the more intricate profile of the vanes, and the integral drain system. These vanes have a high capacity and efficiency and can easily cope with even the record rainfall rates mentioned previously without the need for a weatherhood, thereby saving weight and expense. Since droplets can develop on the filter stages due to humidity, a vane separator is also necessary as the final stage to prevent the problems shown in Fig. A-51.

In addition, technology has improved, so that new, tougher filter materials are now available that will allow filters to work at slightly higher velocities and so reduce air-filter housing sizes with the associated reductions in capital and shipping costs. These tough materials can withstand shot and grit blast without damage. The higher velocities allow the systems to be designed such that droplets will break off the filter stages but then be caught in the final-stage vane separator.

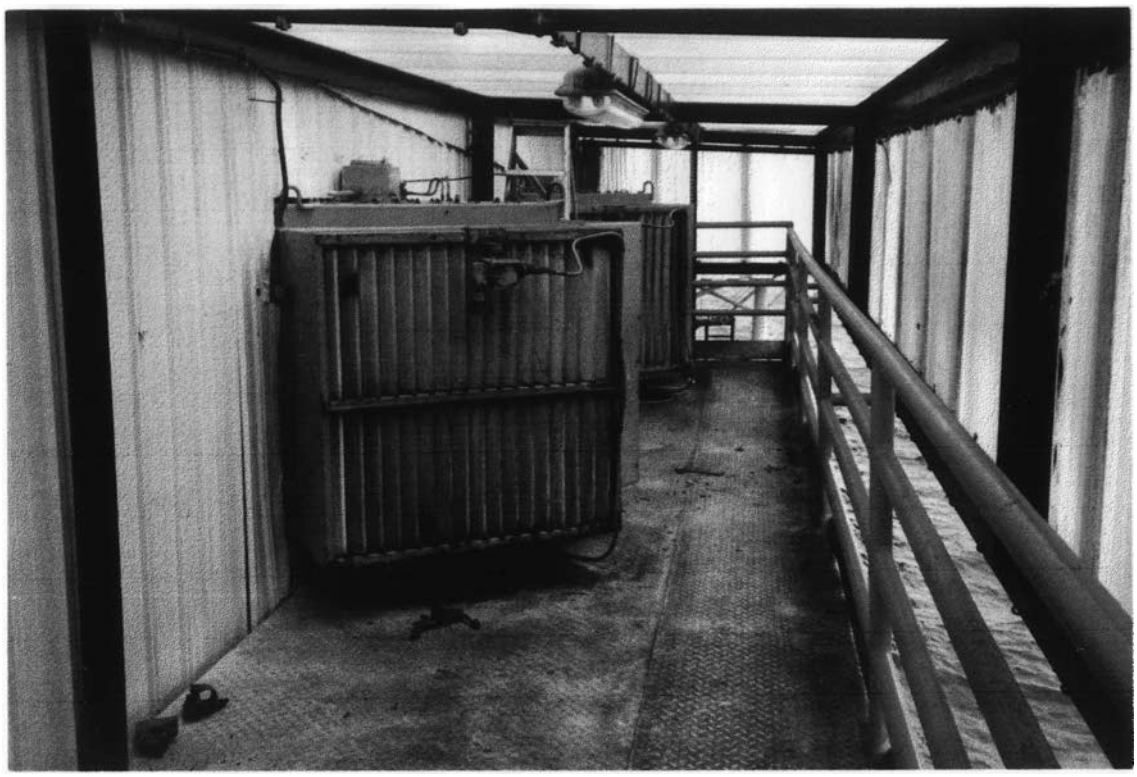


FIG. A-48 Filters protected by a canopy. (Source: Altair Filters International Limited.)

blasting. The lifetime maintenance cost of a painted carbon-steel structure vastly outweighs the small extra premium that stainless steel affords for the initial installation.

Newer Filtration Designs

Clearly, the most worrisome deficiency of the old systems was the combination of poor water-handling ability either from rain or salt-related droplets and the materials of construction.

Gas turbines that operate on warships also experience intermittent heavy-water loadings. These are protected by vane separators, which are, in essence, a high-efficiency weather louvre. The main differences are the close vane pitching, which is 3 to 4 times closer than for weather louvres, the more intricate profile of the vanes, and the integral drain system. These vanes have a high capacity and efficiency and can easily cope with even the record rainfall rates mentioned previously without the need for a weatherhood, thereby saving weight and expense. Since droplets can develop on the filter stages due to humidity, a vane separator is also necessary as the final stage to prevent the problems shown in Fig. A-51.

In addition, technology has improved, so that new, tougher filter materials are now available that will allow filters to work at slightly higher velocities and so reduce air-filter housing sizes with the associated reductions in capital and shipping costs. These tough materials can withstand shot and grit blast without damage. The higher velocities allow the systems to be designed such that droplets will break off the filter stages but then be caught in the final-stage vane separator.



FIG. A-49 Drain valve. (Source: Altair Filters International Limited.)

It is almost universally agreed that stainless steel is the most cost-effective long-term solution for construction of the air-inlet system. The premium between stainless steel over painted carbon steel can now be as low as 20%, whereas there are no further painting costs and the life is infinitely longer.

The grade of stainless steel is also important. It is recognized that the lower grades, such as American Iron and Steel Institute AISI 304 and AISI 321, do not have sufficient corrosion protection, particularly if the material is work hardened. AISI 316 is the most popular choice since it has up to 18.5% chromium, a metal whose presence helps to build up a passive protective film of oxide and prevents corrosion. Together with 10 to 14% nickel content, the steel has an austenitic structure that is very ductile and easily welded.

It also can have a low carbon content (below 0.03%) as well as a molybdenum content of between 2.0 and 3%, which increases its resistance against pitting. Indeed, one operator has paid a significant premium in both cost and delivery time by insisting that the molybdenum content be no lower than 2.5%.

Not only are the filter housings now constructed in AISI 316 but also almost all of the items such as vane separators, door locks, hinges, and instrumentation are supplied in this same material. These inlet systems will give a long life, and they look good as well. A typical system is shown in Figs. A-64 and A-65.



FIG. A-50 Salt penetration through filters. (Source: Altair Filters International Limited.)

The attention to detail is now evident. Figure A-65 clearly shows the elaborate drain systems that are now installed. In addition, the stainless-steel housings are carefully segregated in the manufacturing shop to prevent any cross-contamination from any other ferrous materials, which includes tooling.

Figure A-66 shows a Brunei 4 platform where five of the engines had been retrofitted with this system.

In summary the main requirements of a filtration system in a tropical environment are

1. Protection against tropical rainstorms by vane separators
2. The inclusion of an integrated drain system
3. The selection of AISI 316 stainless steel as the material of construction
4. Protection against droplet carryover by a final stage vane separator



FIG. A-51 Salt penetration through filters. (Source: Altair Filters International Limited.)



FIG. A-52 Corrosion at silencer outlet. (Source: Altair Filters International Limited.)

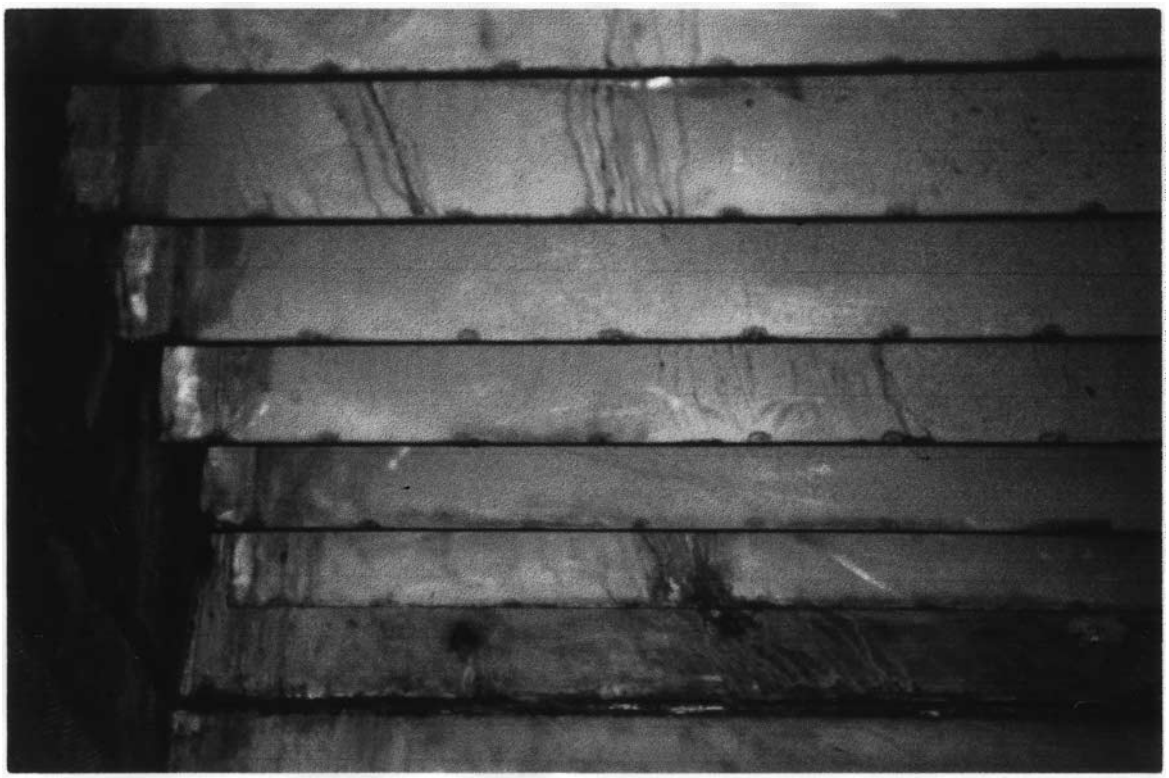


FIG. A-53 Water penetration through the inlet silencer. (Source: Altair Filters International Limited.)



FIG. A-54 Corrosion in plenum. (Source: Altair Filters International Limited.)

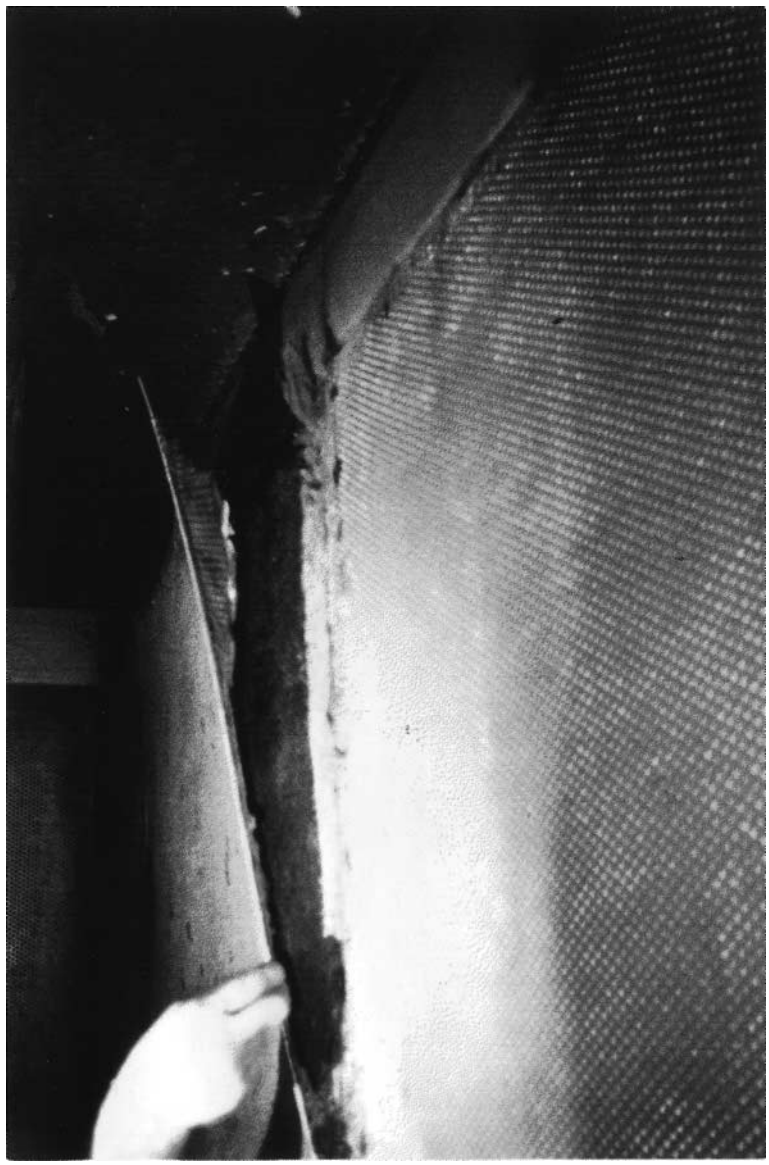


FIG. A-55 A detached plenum lining. (Source: Altair Filters International Limited.)

5. Protection against insects with an insect screen
6. The use of dust extract systems only where essential

The Offshore Environment*

In Europe in the late 1960s, the only data generally available on the marine environment was generated from that found on ships. Since at that time there was considerable interest in using gas turbines as warship propulsion systems, several attempts were made to define the environment at sea, with particular respect to warships.

---

*Source: Altair Filters International Limited, UK. Adapted with permission.



FIG. A-56 A corroded inertia filter. (Source: Altair Filters International Limited.)

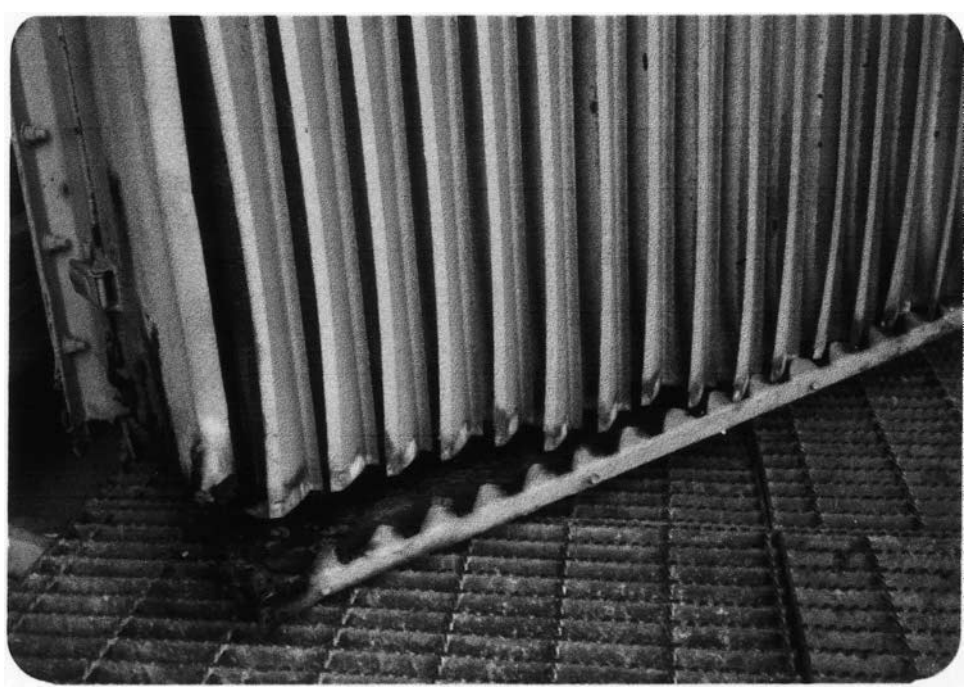


FIG. A-57 A corroded weather louvre. (Source: Altair Filters International Limited.)

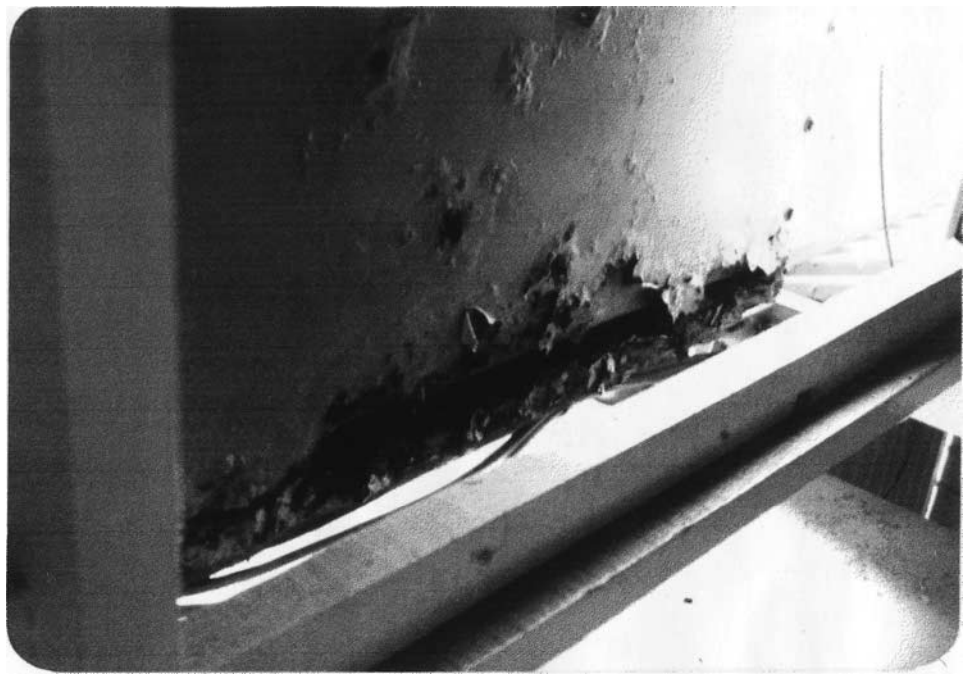


FIG. A-58 Corrosion downstream of filters. (Source: Altair Filters International Limited.)

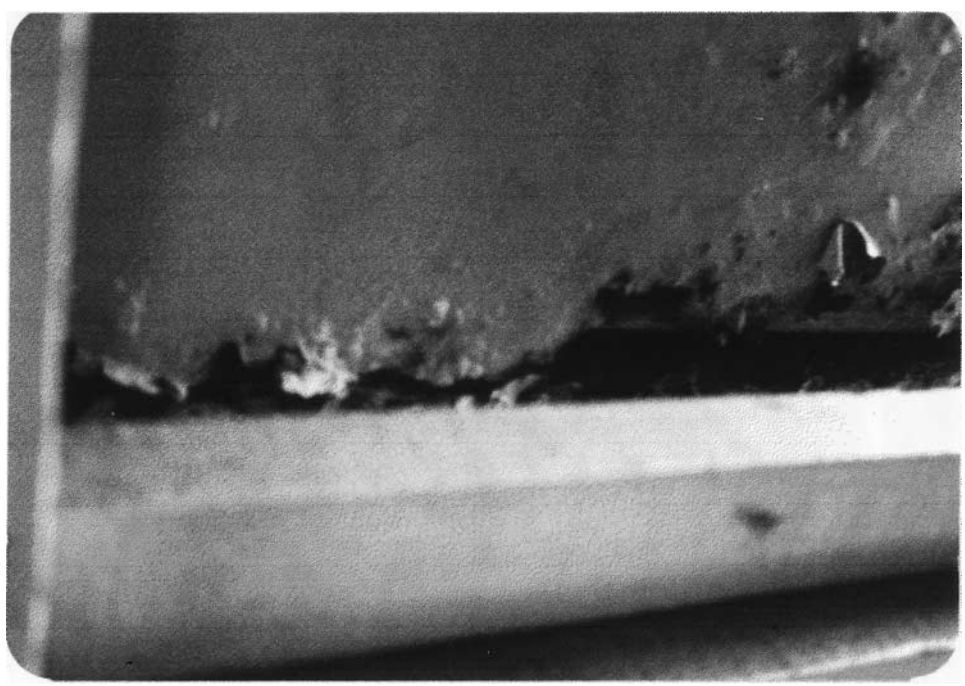


FIG. A-59 Corrosion downstream of filters. (Source: Altair Filters International Limited.)

Not only was it found difficult to produce consistent data, but other factors such as ship speed, hull design, and height above water level had major effects. It became apparent that predicting salt in air levels was as difficult as predicting weather itself.

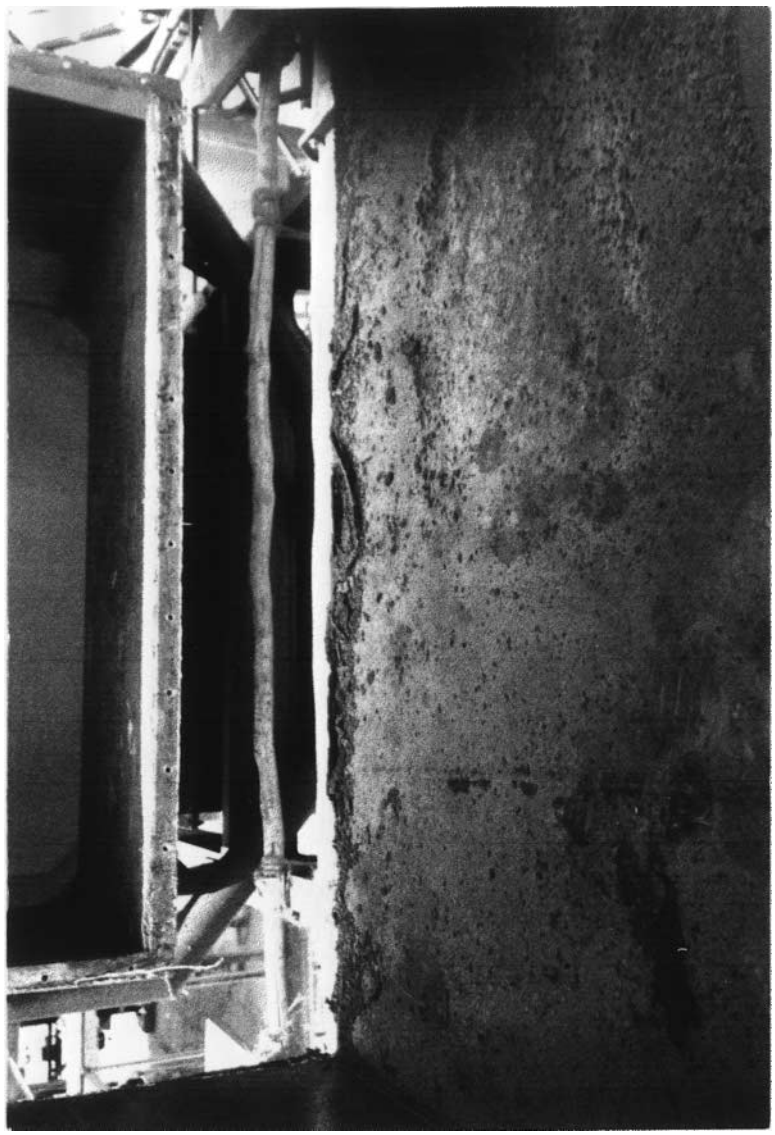


FIG. A-60 Corrosion debris in inlet duct. (Source: Altair Filters International Limited.)

Since the gas turbine manufacturers had defined a total limit of the amount of the contaminants that the turbines could tolerate, some definition of the environment was essential to design filter systems that could meet these limits.

Many papers and conferences were held with little agreement, as can be seen in Fig. A-67. However since the gas turbine industry is a conservative one, it adopted the most pessimistic values as its standard, namely the National Gas Turbine Establishment (NGTE) 30-knot aerosol (Table A-11). It was treated more as a test standard rather than what its name implied. In the absence of any other data, this was used to define the environment on offshore platforms, despite the fact that they were much higher out of the water, and did not move around at 40 knots!

This then defined the salt in air concentration, but did not address any other particulates. In hindsight, it now seems naive that the offshore environments were originally considered to be clean with no other significant problems than salt. Many



FIG. A-61 A paint blister. (Source: Altair Filters International Limited.)



FIG. A-62 A new filter housing awaiting installation. (Source: Altair Filters International Limited.)

equipment specifications were written at that time saying the environment was "dust free."

In the early 1970s there was also a lively debate as to whether the salt in the air was wet or dry. One argument was put forward that if the salt was wet it would



FIG. A-63 Corrosion on the new housing shown in FIG. A-62. (Source: Altair Filters International Limited.)

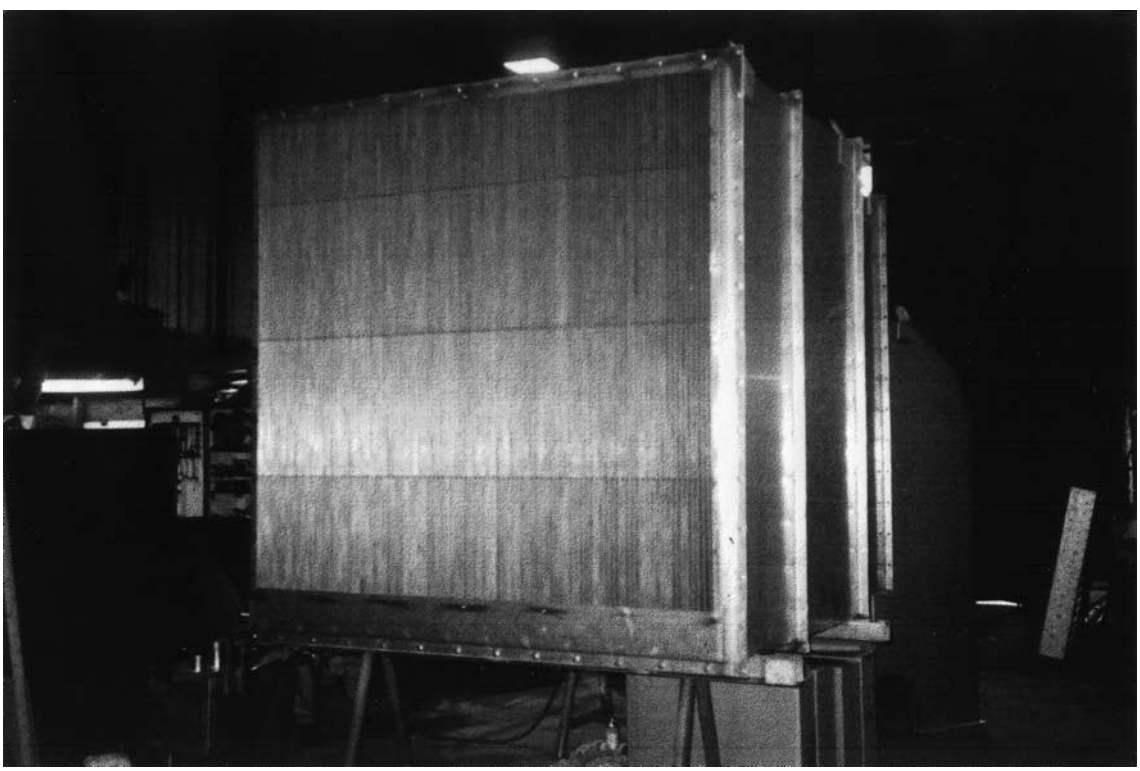


FIG. A-64 A stainless-steel filter housing. (Source: Altair Filters International Limited.)

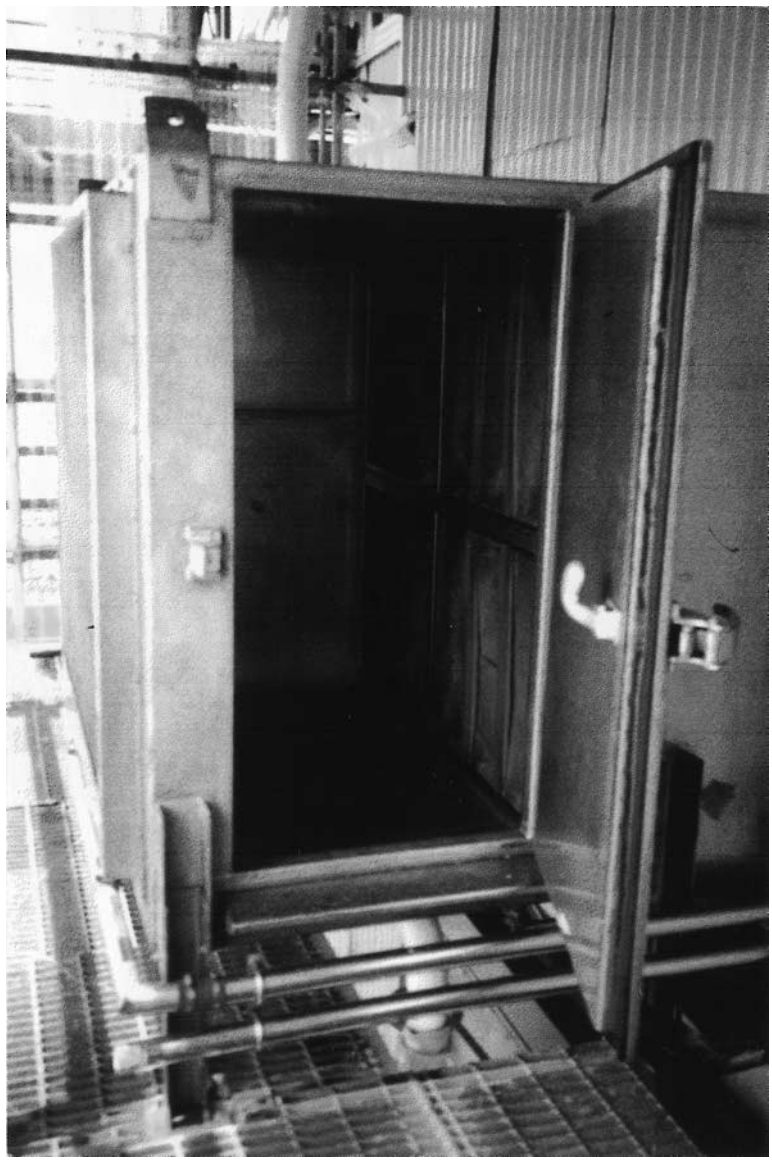


FIG. A-65 A stainless-steel filter housing. (Source: Altair Filters International Limited.)

TABLE A-11 NGTE 30-knot Aerosol

Microns	Salt Content, ppm
<2	0.0038
2–4	0.0212
4–6	0.1404
6–8	0.3060
8–10	0.4320
10–13	0.6480
>13	2.0486
Total	3.6000



FIG. A-66 Brunei shell petroleum Fairley 4 platform, showing five new filter housings. (Source: Altair Filters International Limited.)

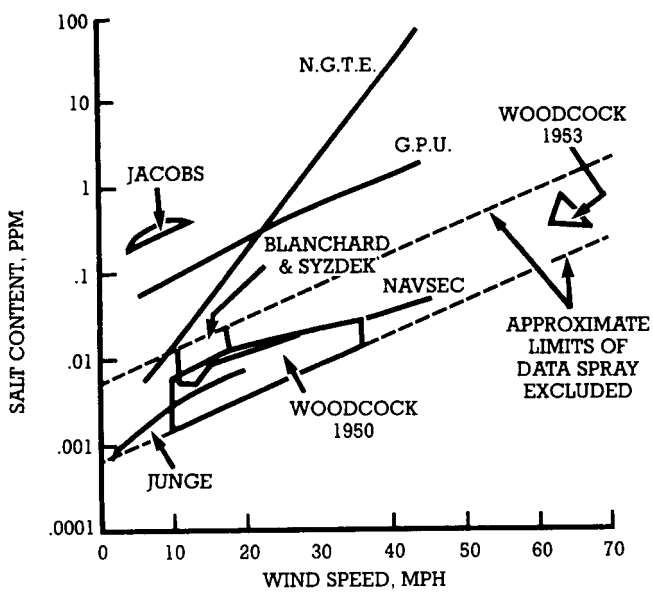


FIG. A-67 Airborne salt comparisons. (Source: Altair Filters International Limited.)

require a further stage of vane separators as the final stage to prevent droplet reentrainment from the filters. The opposing argument maintained that vane separators were unnecessary and that a lower humidity resulted in evaporation of the droplet, giving a smaller salt particle that required a higher degree of filtration. Snow and insect swarms were largely ignored as a problem.

The Initial Filter Designs

The types of systems that were used on the first phase of the developments in the North Sea fell into two categories: high-velocity systems where the design face velocity was normally around 6 m/s, and lower velocity systems that operated at 2.5 m/s.

The low-velocity system was a similar system used on land-based installations, and usually comprised a weather louvre, followed by a prefilter and a high-efficiency bag or cartridge filter. Sometimes a demister stage was added and occasionally, bleed extract inertials were supplied as a first stage.

The high-velocity system was attractive to packagers since it was lighter and occupied a much smaller space. It was the system derived from shipboard use and comprised a vane, coalescer, and vane system.

In general, both systems were housed in mild steel housings with a variety of paint finishes. The weather louvres and vanes were normally constructed from a marine grade aluminum alloy. The filter elements often had stainless steel or galvanized frames.

Bypass doors were used to protect the engine against filter blockage.

The emphasis by package designers was to include a provision for a "3 or 4 stage system" often without regard for what those stages should comprise.

The Actual Offshore Environment

The actual offshore environment is in many ways different from that originally envisaged.

Salt in air is present, although it is only a problem when the filtration system leaks or is poorly designed. Horizontal rain can be a severe problem although sea spray does not generally reach the deck levels even in severe storms.

Flare carbon and mud burning can be a significant problem if the flare stack is badly positioned or if the wind changes direction. (See Fig. A-68.) Not only do the filters block more quickly, but greasy deposits can cover the entire filter system, making the washing of cleanable filters more difficult.

The relative humidity offshore was found to be almost always high enough to ensure that salt was in its wet form. Some splendid work by Tatge, Gordon, and Conkey concluded that salt would stay as supersaturated droplets unless the relative humidity dropped below 45%. Further analysis of offshore humidities in the North Sea showed that this is unlikely to happen (Table A-12).

Initially it was thought that the platforms were dust free, but this is far from the case.

Drilling cement, barytes, and many other dusts are blown around the rig as they are used or moved. But the main problem has resulted from grit blasting. As the platforms got older, repainting was found to be an accelerating requirement with grit blasting a necessary prerequisite.

TABLE A-12 Monthly Average Relative Humidity, North Sea

January	91	July	85
February	86	August	83
March	87	September	83
April	84	October	80
May	86	November	81
June	84	December	80

Source: ASME Report 80-GT-174.

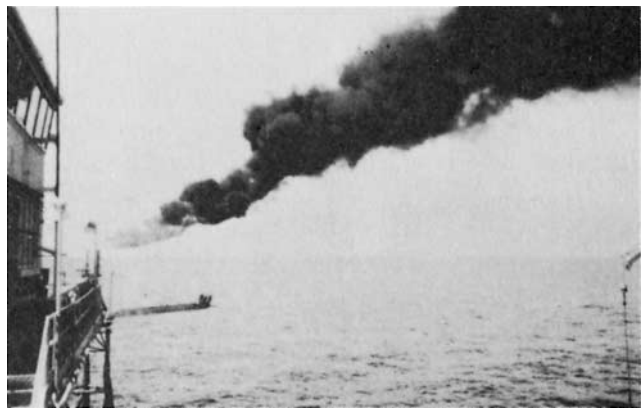


FIG. A-68 Flare carbon can cause problems. (Source: Altair Filters International Limited.)

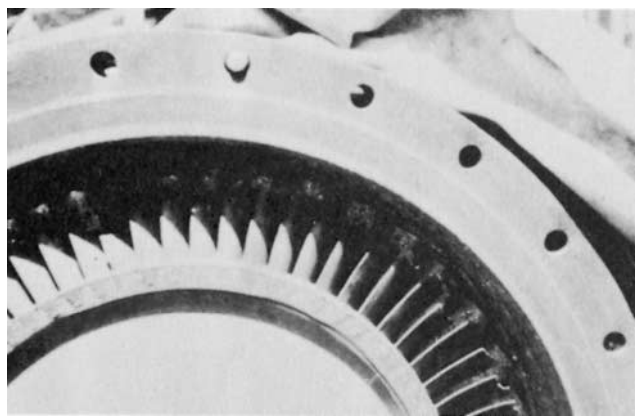


FIG. A-69 Typical turbine damage. (Source: Altair Filters International Limited.)

In order to be effective, grit is sharp and abrasive by design and can be devastating if ingested into a gas turbine. (See Fig. A-69.) The quantities used can seem enormous. On one platform it was found that over a 12-month period, 700 tons of grit blast had been used!

The Problems Encountered

In general, problems were slow to appear, typically taking three to five years after start-up, but since a lot of equipment had been installed at about the same time, the problems manifested themselves like an epidemic.

These problems could be categorized as follows:

1. Erosion of compressor blading
2. Short intervals between compressor cleaning
3. Frequent filter change-out
4. Turbine corrosion
5. Corrosion of the filters and housing

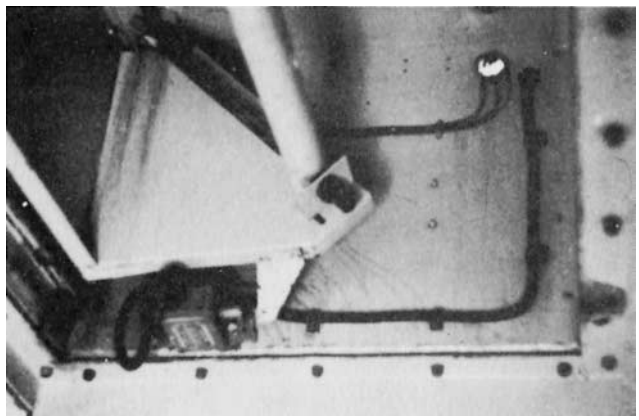


FIG. A-70 Typical leak caused by a missing cable gland. (Source: Altair Filters International Limited.)

By far the most serious of these problems was the erosion of compressor blading that was experienced almost simultaneously on many platforms. This occurred about three to five years after start-up, as this was the time that repainting programs were initiated. Grit blast found its way into the turbine intakes either through leaking intakes, bypass doors, or through the media itself. (See Fig. A-70.) Since the airborne levels were high, the air filters quickly blocked up, allowing the bypass doors to open. As filter maintenance is not a high priority on production platforms, considerable periods were spent with grit passing straight into the turbine through open bypass doors. Even where maintenance standards were more attentive, there were usually enough leaks in the intake housing and ducting to ensure delivery of the grit to the turbine.

It often seemed contradictory that the system designers would spend a lot of time specifying the filter system, but would pay little attention to ensuring the airtightness of the ducting downstream.

Since the grit was sharp, it sometimes damaged the filter media itself, reducing the system efficiency dramatically.

Bypass doors were a major problem. Early designs failed to take account of the environment or the movement in the large structures of the filter housings. Very few of those initial designs were airtight when shut, and it was not uncommon for them to be blown open by the wind.

Turbine corrosion could almost always be traced to leaky ducting or operation with open bypass doors. Very few systems gave turbine corrosion problems if the ducting was airtight. The few installations that did give problems were usually the result of low-velocity systems operating with poor aerodynamics, so that local high velocities reentrained salt water droplets into the airstream and onward to the engine. (See Fig. A-71.)

Rapid compressor fouling was usually the prelude to more serious problems later, since it was usually caused by the combined problems of filter bypass.

Compressor cleaning almost once a week was fairly standard for systems with those problems.

As time progressed the marine environment took its toll on the carbon steel and severe corrosion was experienced on the intake housing and ducting. (See Fig. A-72.) In some cases, corrosion debris was ingested into the turbine causing turbine failure. This again was accelerated by poor design, which allowed dissimilar metals to be put into contact, leading to galvanic corrosion.

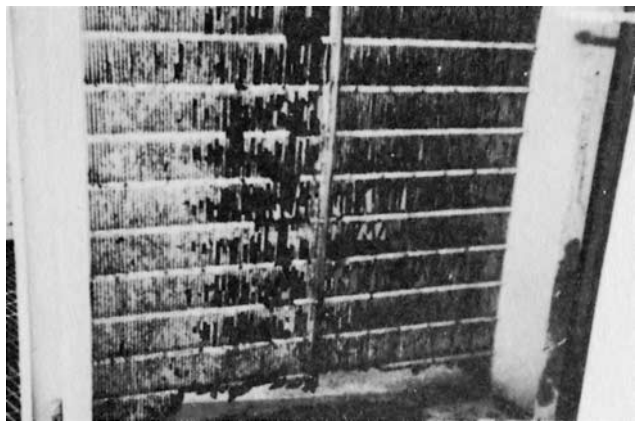


FIG. A-71 Inertial filters showing severe corrosion. (Source: Altair Filters International Limited.)

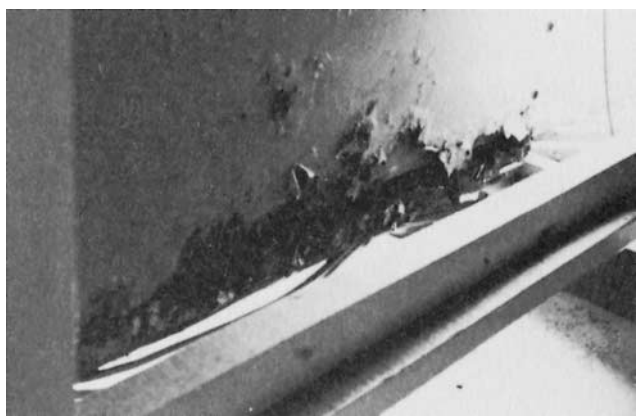


FIG. A-72 Severe duct corrosion. (Source: Altair Filters International Limited.)

Specification of a Typical Filter Used in the Offshore Environment*

Gas turbines are an ideal power source for driving compressors, pumps, and generators. Since they are relatively small compared to their power output, they can be used easily in remote locations such as jungles, deserts, and offshore platforms. They are, however, very complex pieces of machinery, comprised of many high-tolerance rotating parts.

The engineering is further complicated by the engine manufacturers' need to increase the turbine efficiency by increasing operating temperatures. In order to overcome the material stresses associated with these higher temperatures, internal cooling passages have been introduced into the engine. Typically, turbine blades are now of hollow construction with cooling air blown through them, exiting through tiny holes in the blade surface. These holes can be very small and are very susceptible to blockage. The requirement for filtration of the gas turbine air is, therefore, even more stringent than in the past. The need to filter the air to the gas turbine is fourfold:

* Source: Altair Filters International Limited, UK. Adapted with permission.

- To prevent erosion
- To protect against fouling
- To prevent particle fusion
- To protect against corrosion

Erosion

Erosion is caused by particles impacting and wearing away the metal surfaces. The high speeds of the rotating blades collide with the airborne particles and produce a quite large change in energy, which results in fragments of metal being blasted out of the blade surface. Even particles as small as 10 μm in diameter can cause severe erosion. The composition and shape of the particle can also significantly affect the erosion rate. Blade profiles are so carefully designed that even minor abrasions can alter the profiles to such an extent that engine performance is affected. Erosion is an expensive problem since it causes permanent damage and the affected parts will require replacing. It is proportional to concentration and in severe duties, such as gas turbine–powered hovercraft operated in desert conditions, engine life has been reduced to as little as 6 hours.

Fouling

Engine fouling, by comparison, is normally only a temporary problem and is caused by a buildup of contamination that adheres to the internal surfaces.

Again, deposition on blade surfaces can change profiles, with the resultant loss in engine power and an increase in fuel consumption. Particles of 2 μm and less are generally the major cause of fouling. Smoke, oil mist, and sea salt are common examples.

The particles are attracted to the metal surfaces by a variety of forces, including impaction, electrostatic, and capillary action. The composition of the particle, again, is important in determining the rate of fouling. In marine environments, dry dust particles are often coated in a layer of sea salt, which is viscous by nature and adds to the fouling action. While fouling is basically a temporary problem, it can be removed by various cleaning techniques. It is an irritant to the operator, as many of these cleaning processes have to be conducted at reduced powers or with the engine stopped. In the past, engines were cleaned by injecting a mild abrasive into the engine to clean off the contamination when the engine was running. While the most common material was a mixture of ground coconut shells, rice has been used on some tropical engines. The practice of using online cleaning has now mostly been abandoned since it tended to transfer large particles of debris into other areas of the turbine, causing even more problems. There was also a view put forward that it accelerated hot end corrosion. Modern cleaning methods use a detergent sprayed into the engine on a cold cycle, leaving it to soak and then washing it off with clean water.

Particle fusion

Dry particles, which range in size from 2 to 10 μm, could, on old engine designs, pass through the engine causing little or no damage. However, on the new generation of hotter engines, these particles can cause problems if their fusion temperature is lower than the turbine operating temperature, since they will melt and stick to the hot-metal surfaces. This can cause severe problems since this molten mass can block cooling passages and cause thermal fatigue. The affected surface is permanently disfigured and will need replacement.

Corrosion

The high temperatures of the gas turbine can also cause rapid acceleration of the corrosion process. Even though the hot-metal surfaces are made of some of the most sophisticated materials, corrosion can still be extremely rapid. Blade failures in as little as 100 operating hours have been known, and failures within 2000 operating hours are relatively common. Corrosion, however, can be completely prevented by modern techniques, and yet it still occurs.

Normally, corrosion is produced by a salt, such as sodium or potassium, but lead and vanadium are also common contributors. Since many gas turbines are based either offshore or close to the sea, sea salt (sodium chloride) is the main offender. In the cold parts of the engine it is the sodium chloride that does the damage, whereas in the hot parts of the engine it is sodium sulfate (or sulfidization) that creates most of the corrosion. Sodium sulfate is produced from the combination of sulfur in the fuel and sodium chloride in the air.

It is important to recognize that the corrosion process is self-propagating, and, once started, will continue even though the source problem has been cured.

The modern gas turbine therefore is a sensitive machine and needs to be protected to provide an acceptable life cycle. For this reason, there are limits that are recommended by the manufacturers in order to achieve this. There is not one universal limit that is adopted by all manufacturers. Each has its own, which is expressed in many different forms, either as an absolute limit or one that is time dependent. However, all seem to work from the same premise.

Previously, it was often thought that providing a gas-turbine air-filter system was well chosen; it could be used in almost any environment with equal effect. This has proved to be a fallacy, as many operators have found at their cost.

The Problems Solved

Since engines appeared to be eroding at a fairly rapid rate, irrespective of which type of filter system was fitted, an equally rapid response was required.

Phillips decided, after removing a GT22 engine from their Ekofisk Bravo platform, that they would not operate the repaired engine until the filter system was changed, even though the source of the problem had not been identified at that time.

An evaluation of all the likely contaminants was quickly undertaken, with large quantities of various suspicious substances shipped back to the laboratory for analysis and trial against the installed filter system in the wind tunnel. Grit blast was confirmed as the source of erosion, and the installed filter system gave only a 28% protection.

Trials against other conventional filter medias proved negative, since the grit was sharp and eventually cut its way through the media.

A new type of filter was required with a very strong media and a large capacity to absorb the huge quantities of grit without blocking too quickly. The ability to work at high velocity would also be an advantage.

This was achieved by using a bonded polyester fiber, which proved almost impossible to tear. The strength of the polyester allowed the media to be operated at higher velocities without fear of fiber loss, which can be a limitation with the brittle glass fibers used in conventional filtration.

The old filter makers' methods of packing in more media with the hope that the increased area would reduce the media velocity has distinct limitations since the extra area is not effectively used by the poor aerodynamics created. Instead a careful aerodynamic design ensured a more even distribution throughout the filter,

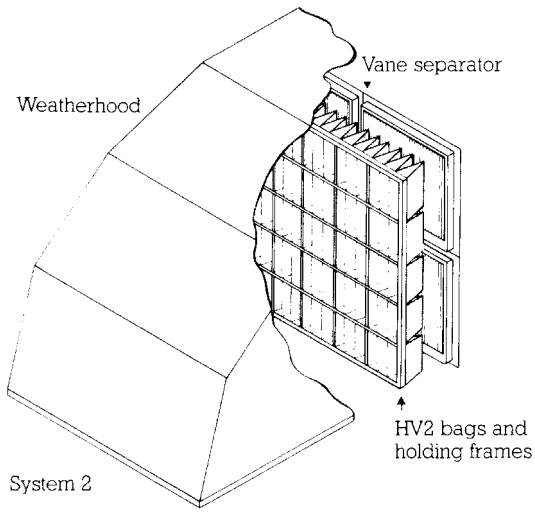


FIG. A-73 Typical air filter on a platform. (Source: Altair Filters International Limited.)

giving a relatively wide pocketed bag filter capable of those higher velocities and giving protection against the problem contaminants. By having a relatively high loft to the media, protection against filter blinding was ensured, with a resulting longer life. Testing showed the new HV2 filter to be over 99% efficient against the damaging grit blast.

A cleanable prefilter bag of similar but less dense media was also quickly designed. This PB1 prefilter bag was unique in that it was designed to be tucked inside the final filter bag and so only took a further 25-mm installation depth.

The new filters were installed into a new stainless steel housing protected front and back by vane separators and delivered on Christmas Eve 1983, just three weeks after the initial problems were investigated; a record for which all those involved have a right to be proud of. (See Fig. A-73.)

Shell was the next platform operator to experience similar problems, firstly on the Avon gas turbines on its Brent Delta platform. In these installations it was possible to fit the HV2 filters in an access space between the existing filter stages. Shell undertook a bold and very correct decision to weld up the troublesome bypass doors, having first revised the alarm and trip levels for the intake depression pressure switches. This system was carefully monitored for a period of nine months before the remaining 23 Avons on the Brent platforms were similarly converted. Comparison of two adjacent engines, one with the original system and one with a retrofitted system showed that over a three-month period the performance of the retrofitted engine was unchanged, while the other engine showed a steady increase in exhaust temperature for a given power output amounting to 30°C at the end of the period. Also, the requirement to change out the filters was reduced from 350 hours to, in some cases, over two years.

On Shell Leman BK platform, a similar comparison of two Avons was investigated, with an air sampling program constantly monitoring the quality of the inlet air over a period of one month. This showed the modified installation to be 10–14 times more effective, in terms of particle penetration.

It is not surprising that Shell has now retrofitted 78 installations worldwide. In total this design has been selected for 212 new and retrofitted gas turbine engines.

Filter systems that were an operational irritation every 15 days or so are now forgotten to such a degree that on some platforms the filters have operated, without replacement, for up to three years.

Compressor cleaning is operated on a planned maintenance approximately every 2000 hours.

The Systems of the Future

With so much experience gained with these retrofitted installations, new installations are now designed taking account of the lessons learned.

Typically a new offshore system will comprise the following features:

1. A housing made entirely of 316 stainless steel
2. A weather hood or high-efficiency weather louvre constructed in 316 stainless steel
3. A prefilter and filter stage capable of high efficiency against grit blast and other contaminants
4. A final vane separator to protect against droplet reentrainment
5. No bypass doors
6. Pressure switches for alarm and shutdown
7. An integrated drain system
8. All materials capable of withstanding a marine environment, with an exclusion of dissimilar metals, cardboard, and the like
9. A leak-free intake system

The argument for the high-velocity (6 m/s) system is now proven, with over 200 installations worldwide. The advantage of smaller size and lower weight will become more important in the future, and may push the current designs even further.

The key components of the system* are:

High-efficiency filtration. (Hercules™)

Dynamic water eliminator. High-efficiency separator ensures that salt carryover problems are eliminated (Hydra™)

Hercules and Hydra combine to form System Aquila (Fig. A-74), providing the following features:

High volume flow. Leading to a filter house with a 65% smaller face area than traditional systems. This means a customer saves space and weight, which also saves cost. (See Fig. A-75.)

High efficiency. High dust arrestance and salt removal efficiencies provide excellent protection from erosion, corrosion, and fouling of turbine blades. (See Fig A-76.)

Low pressure loss. A typical clean system pressure loss of only 45-mm H_2O means lower fuel consumption, higher output, and longer filter life for operators.

High-efficiency filtration

This feature has been aerodynamically designed to ensure that maximum particulate efficiency is achieved with the minimum resistance to airflow. The

* Note: Trademarks are specific to the source for this section. Each manufacturer will have its own equivalent terms and trademarks.

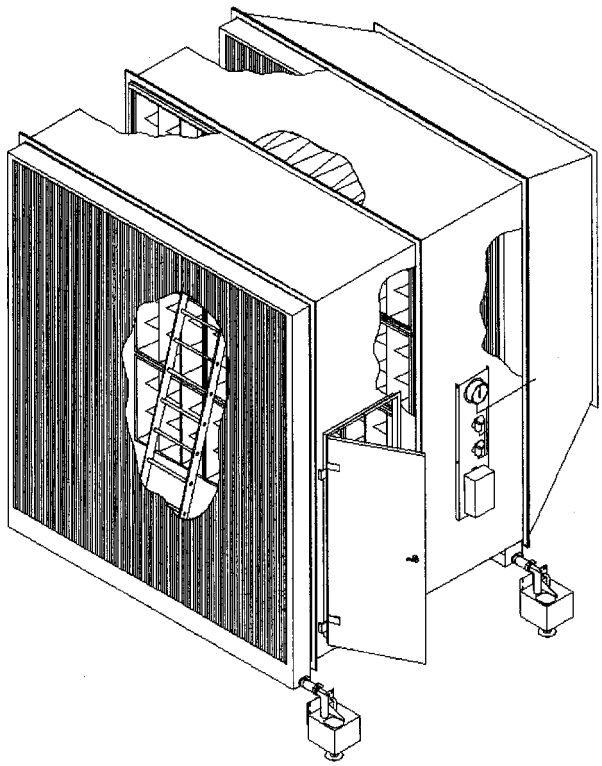


FIG. A-74 Filter with water eliminator. (Source: Altair Filters International Limited.)

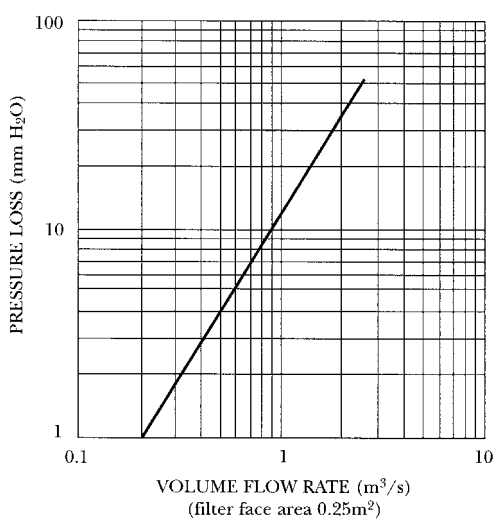


FIG. A-75 Pressure loss versus volume flow rate filter characteristic. (Source: Altair Filters International Limited.)

semirigid construction, together with the fact that each pocket is divided into smaller segments by means of a semipermeable "shelving" system, ensures the best possible profile throughout all operating conditions. This produces an extremely uniform flow distribution, leading to improved dust-holding capacity and eliminating the likelihood of localized dust breakthrough.

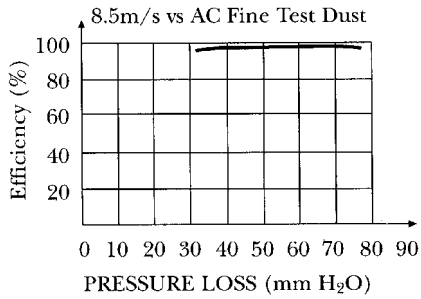


FIG. A-76 Efficiency versus pressure loss filter characteristic. (Source: Altair Filters International Limited.)

Dynamic water eliminator

This feature conducts water and salt removal. The vanes, which are constructed from corrosion-resistant marine grade aluminum (other materials are available), are produced with a profile that allows the maximum removal of salt and water, yet produces an extremely low pressure loss. This optimal profile has been achieved by the very latest design methods, and in particular by utilizing a Computational Fluid Dynamics (CFD) flow modeling system. Hydra also incorporates a unique and novel method of separating water droplets from the air stream, and this has led to improvements in bulk water removal compared with conventional methods.

Reference and Additional Reading

1. Tatge, R. B., Gordon, C. R., and Conkey, R. S., "Gas Turbine Inlet Filtration in Marine Environments," ASME Report 80-GT-174.

Typical Specifications for Range of Air Filters

This range includes panels and bags as well as high-efficiency, high-velocity systems and air/water separators.

Filter holding frames are constructed in mild or stainless steel, designed to provide quick and easy removal from upstream, downstream, or sides of ducting, without the use of springs or clips of any kind. Filter housings, ducting, louvres, dampers, and silencers can also be designed and fabricated, providing a total system capability.

Air Pollution Control*

The main methods of combating and controlling air pollution include:

- Electrostatic precipitators (for particulates)
- Fabric filters (for dust and particulates)
- Flue gas desulfurization (for SO_x removal)
- SCR $DeNO_x$ (for NO_x removal)
- Absorbers (for environmental toxins)
- End-product–handling systems (for solid and liquid wastes)
- Combined unit systems (for some or all of the previous items)

*Source: Alstom. Adapted with permission.

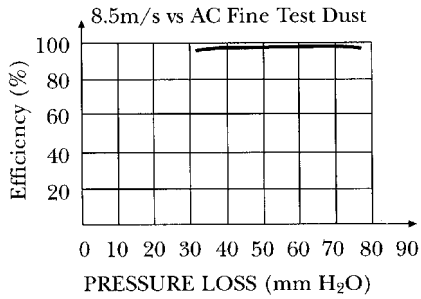


FIG. A-76 Efficiency versus pressure loss filter characteristic. (Source: Altair Filters International Limited.)

Dynamic water eliminator

This feature conducts water and salt removal. The vanes, which are constructed from corrosion-resistant marine grade aluminum (other materials are available), are produced with a profile that allows the maximum removal of salt and water, yet produces an extremely low pressure loss. This optimal profile has been achieved by the very latest design methods, and in particular by utilizing a Computational Fluid Dynamics (CFD) flow modeling system. Hydra also incorporates a unique and novel method of separating water droplets from the air stream, and this has led to improvements in bulk water removal compared with conventional methods.

Reference and Additional Reading

1. Tatge, R. B., Gordon, C. R., and Conkey, R. S., "Gas Turbine Inlet Filtration in Marine Environments," ASME Report 80-GT-174.

Typical Specifications for Range of Air Filters

This range includes panels and bags as well as high-efficiency, high-velocity systems and air/water separators.

Filter holding frames are constructed in mild or stainless steel, designed to provide quick and easy removal from upstream, downstream, or sides of ducting, without the use of springs or clips of any kind. Filter housings, ducting, louvres, dampers, and silencers can also be designed and fabricated, providing a total system capability.

Air Pollution Control*

The main methods of combating and controlling air pollution include:

- Electrostatic precipitators (for particulates)
- Fabric filters (for dust and particulates)
- Flue gas desulfurization (for SO_x removal)
- SCR $DeNO_x$ (for NO_x removal)
- Absorbers (for environmental toxins)
- End-product–handling systems (for solid and liquid wastes)
- Combined unit systems (for some or all of the previous items)

*Source: Alstom. Adapted with permission.

Electrostatic Precipitators

In combustion processes, the largest quantities of heavy metals and dioxins are found in the fly ash, or can be contained there by technical means. It is therefore essential to increase even further the very high precipitation efficiencies that are already being achieved. There are two types of ESPs, wet and dry, for collecting particles. (See Fig. A-77.)

New and retrofit systems are used. Retrofitting with new spiral electrodes, a rapping system, and pulsed energization pay immediate dividends in the form of improved abatement efficiency and lower power consumption.

Semipulse® and Multipulse® for enhanced separation and energy efficiency

The rather uncomplicated process of charging dust by means of a high-voltage DC system, which makes dust stick against collector plates, has undergone high-tech refinement. Several of the improvements have been implemented to minimize energy consumption. Originally, it took about 1 MW of power to operate an ESP in a large coal-fired power station. Pulsed energization is a means to cut energy consumption substantially while simultaneously improving separation efficiency. Two systems for this purpose have been developed: the Semipulse Concept (SPC) with millisecond pulses, and the Multipulse Concept (MPC) with microsecond pulses. (See Fig. A-78.)

Since their introduction in 1983, more than 3500 SPC and MPC units have been supplied. SPC can be easily installed in existing plants, while MPC involves a higher investment and is generally considered for retrofits and new plants.

The savings for high-resistivity dust can be substantial. Energy consumption after installation of SPC or MPC is typically between 10 and 20% of the original. At the same time, dust emissions are reduced to 25–50%.

Upgrading or retrofitting with pulsed energization is often the solution when a utility wants to switch to low-sulfur coals, which often produce dust of higher



FIG. A-77 A typical electrostatic precipitator. (Source: Alstom.)

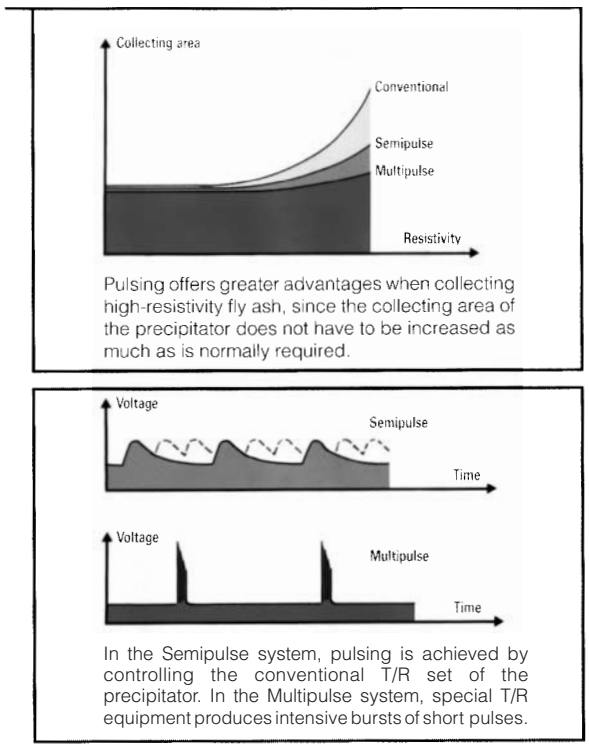


FIG. A-78 Pulse systems in precipitators. (Source: Alstom.)

resistivity. It also gives a utility a wider choice of coals that can easily be valued in money terms.

Semipulse and Multipulse offer an inexpensive route not only by improving energy and separation efficiency, but also by requiring a minimum of supervision and maintenance.

Fabric Filters

Fabric filters are used for cleaning large flows of flue gases from coal-fired power plants and municipal waste incinerators. (See Fig. A-79.)

The fabric filter has gained a wide market, due to its versatility for a large number of dust and process types and its ability to capture all particles, not only those that can be charged electrically (as in ESPs). (See Fig. A-80.)

Another reason for the recent success of fabric filters is that they operate by passing the dust-laden gas through a dust cake that is constantly being built up with the support of the fabric. This enables the removal of a large portion of the finest particles, a feature that is becoming increasingly important as more stringent emission controls are required. (See Fig. A-81.)

With the fine particles, several heavy metals can be trapped in the dust cake, together with sulfur dioxide, if lime is introduced in the flue gas.

This manufacturer/information source supplies two different kinds of fabric filters: the high-ratio and low-ratio type, denominated by the air-to-cloth ratio.

A major difference between the two filter types is the cleaning system. The high-ratio fabric filter is cleaned by the Optipulse® cleaning system (*see following text*).

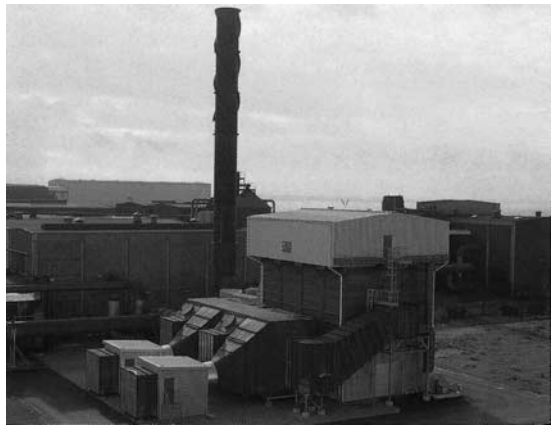


FIG. A-79 Fabric filter installation in a metallurgical plant, Höganäs, Sweden (left). (Source: Alstom.)

In the low-ratio filter, gas enters the filter bags from the inside, then outward. The filter bags are cleaned "off-line" using either the reverse gas flow, reverse gas with high-energy sound horns, or a cleaning system of the deflate or shake mechanism type. (See Fig. A-82.)

The Optipulse® cleaning concept

(Optipulse is a trademark for a proprietary design of this information source.)

Pulse-jet fabric filters operate with dust-laden gas approaching the filter elements from the outside, depositing the particles on the fibers of a depth-filtering medium.

The clean gas leaves the open end of the filter element, which is typically of tubular design with a diameter of 120–150 mm (5–6 in). An internal wire cage supports the filter element against the pressure caused by the gas flow. (See Fig. A-83.)

Periodically, the dust cake is cleaned off by expanding the filter element with a rapid pulse of air. The removed dust cake is transported by gravity toward the dust hopper.

The effectiveness of the cleaning depends on the character of the pressure pulse. Optipulse produces a forceful pulse by an optimized geometry of the pneumatic system delivering the pulse (see Fig. A-84):

- The pulse air is injected in the filter element, without dissipation of energy in a large volume of flue gas, through injection nozzles optimally selected in relation to filter element size.
- The area of all nozzles on the header serving one row of filter elements is matched to the area of a large, pilot-operated, fast-opening supply valve.

Flue Gas Desulfurization (FGD)

Today, flue gas desulfurization is a well-established method to fight global environmental impairment such as acid rain. Most industrial countries have set standards for SO_2 emissions and committed themselves to large reductions of national emissions in international agreements.

There are several different types of FGD technologies for a wide range of applications.

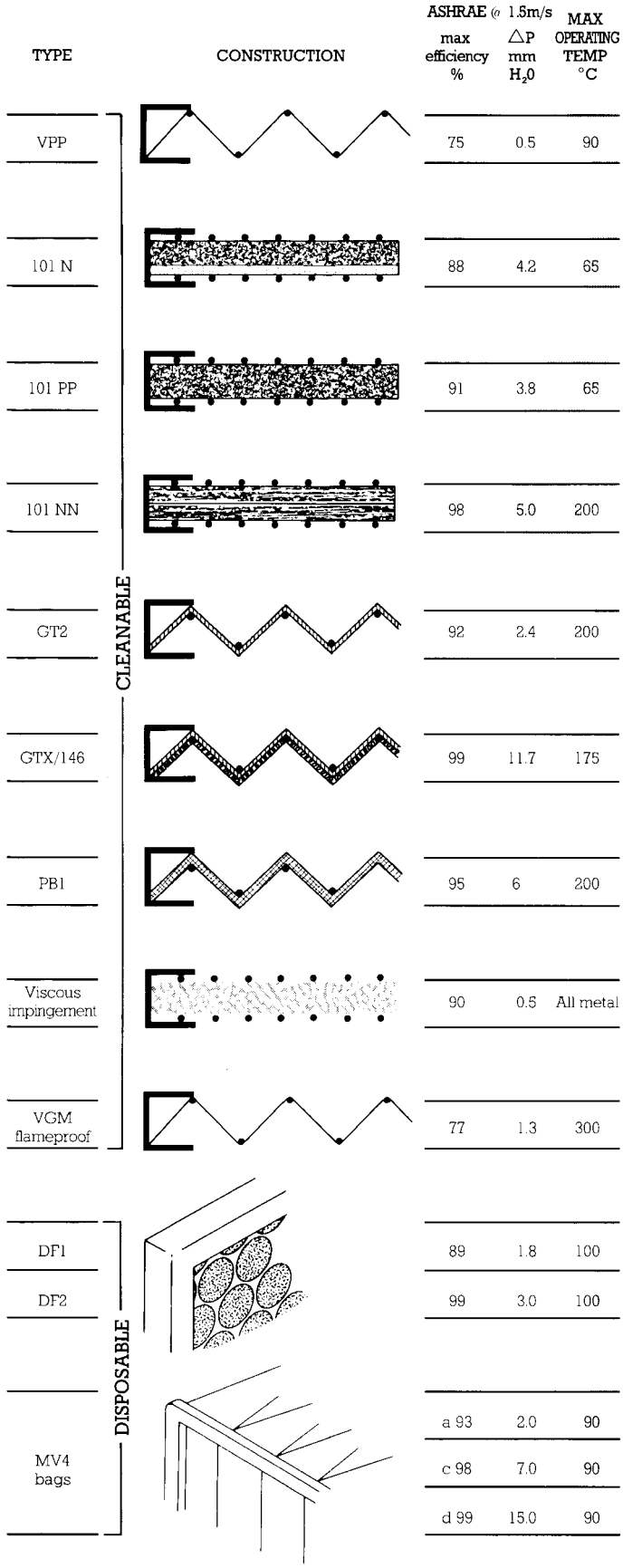


FIG. A-80 Typical filter product range. (Source: Altair Filters International Limited.)

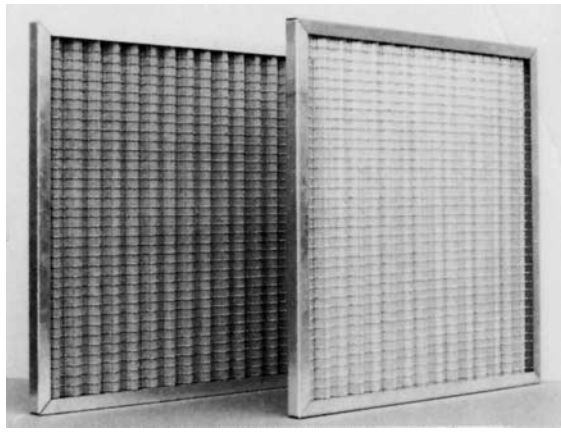


FIG. A-81 Air filter elements. (Source: Alstom.)

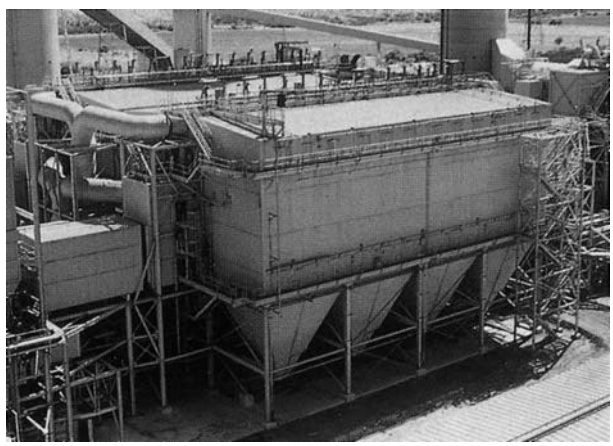


FIG. A-82 Low-ratio fabric filter installation at Nevada Power, United States. (Source: Alstom.)



FIG. A-83 Installation of filter elements in an Optipulse fabric filter. (Source: Alstom.)

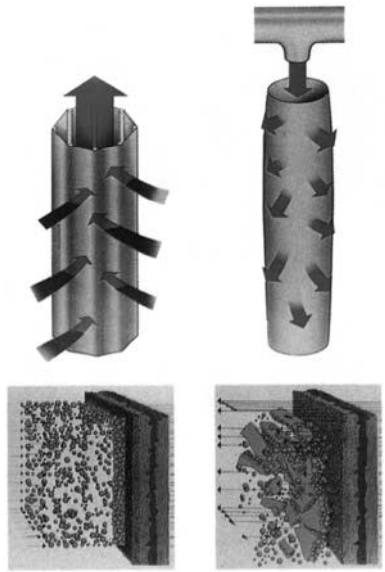


FIG. A-84 Operating principles of the Optipulse pulse cleaning system. (Source: Alstom.)

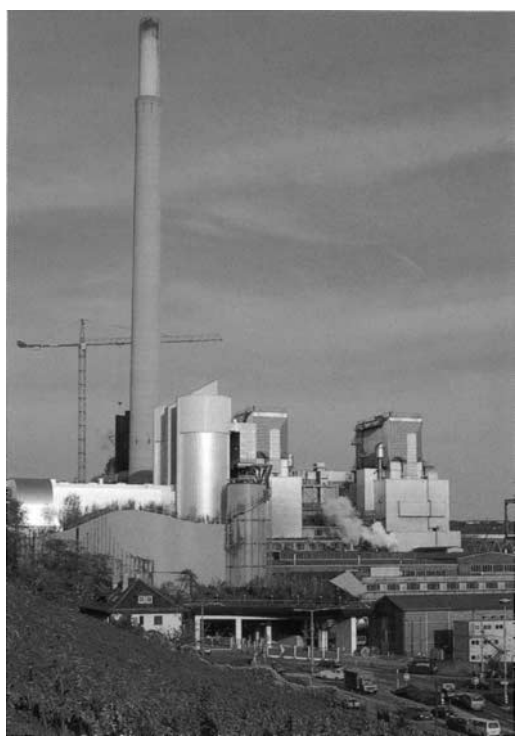


FIG. A-85 Wet/dry flue gas FGD plant, including fabric filters, after two coal-fired boilers at TWS Dampfkraftwerk, Stuttgart, Germany. (Source: Alstom.)

Three common technologies for FGD

1. The wet/dry lime spray drying process offers low capital costs and an easily disposable/reusable end product for small and medium-sized plants. (See Figs. A-85 and A-86.)

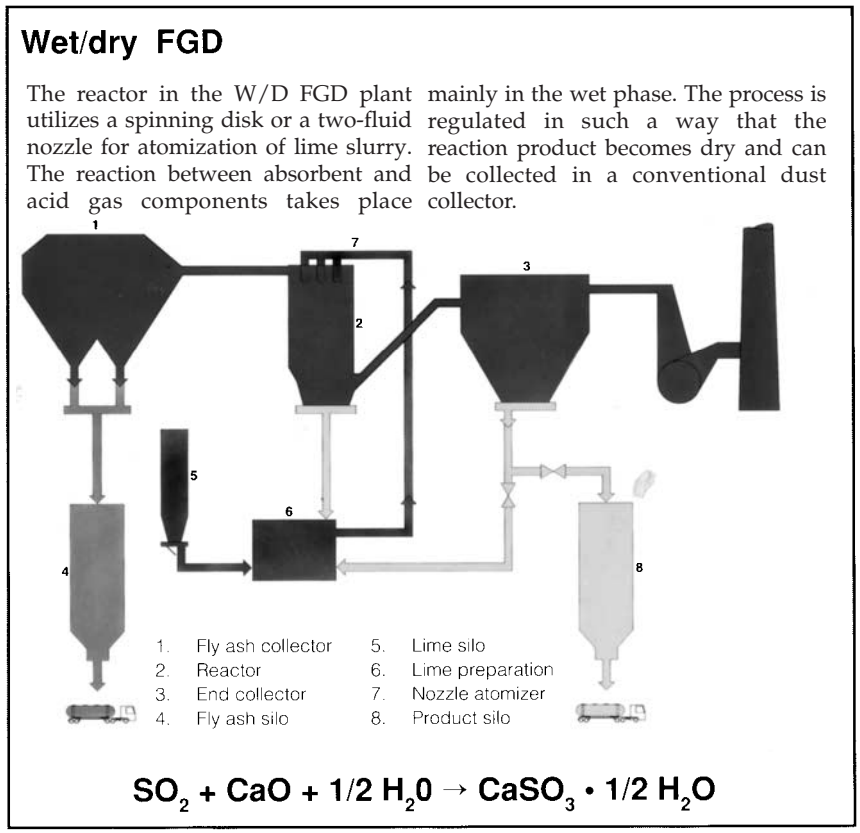


FIG. A-86 Basic principles of wet/dry FGD installation. (Source: Alstom.)

2. The open spray tower lime/limestone wet FGD process offers low operating costs and proven production of commercial grade gypsum. (See Figs. A-87 and A-88.)
3. The seawater process offers low operating costs and fully eliminates disposal problems of end products at plants with access to suitable and sufficient amounts of seawater. (See Figs. A-89 and A-90.)

SCR DeNO$_x$ Technology

Catalysts solve pollution problems

The SCR (selective catalytic reduction) technique was transferred to Europe from Japan, where it was first developed.

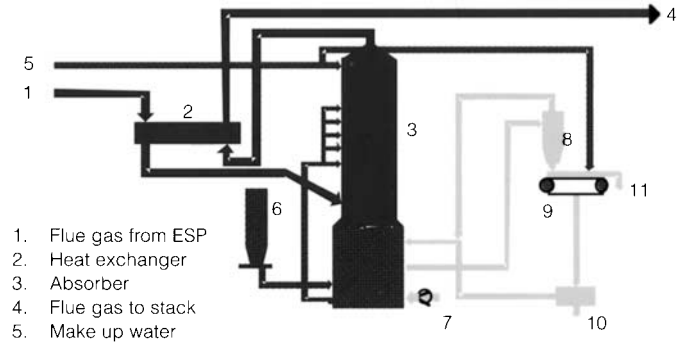
The reduction of nitrogen oxides with ammonia, which occurs spontaneously at high temperature [about 950°C (1750°F)], can be achieved at a manageable temperature after the boiler with the aid of a catalyst. (See Fig. A-91.)

For NO_x reduction, the catalyst is usually an active phase of vanadium pentoxide and tungsten trioxide on a carrier of titanium. Other types of catalysts are available, however.

The ideal catalytic reactor is made up of catalyst elements that are assembled in modules usually $1 \times 1 \times 2$ meters in size. A reactor normally has three to four layers of catalyst modules.

Three positions of the reactor are possible in the treatment chain (see Fig. A-92):

$$SO_2+CaCO_3+2H_2O+1/2\ O_2 \rightarrow CaSO_4 \bullet 2H_2O+CO_2$$



1. Flue gas from ESP
2. Heat exchanger
3. Absorber
4. Flue gas to stack
5. Make up water
6. Limestone silo
7. Oxidation air
8. Hydrocyclone
9. Vacuum filter
10. Waste water
11. Gypsum

Wet FGD

In a wet FGD plant, line or linestone slurry is sprayed through nozzles into the gas flow. The mixture of slurry and reaction products is gathered at the bottom of the absorber-tower and recycled through the spray-nozzles. An important element in the wet FGD process is the mist eliminator above the spray nozzlebanks.

Secondary oxidation is normally achieved through introduction of oxygen at the bottom of the slurry tank.

With oxidation, the reaction product, after dewatering, will be gypsum.

FIG. A-87 Wet FGD plant. (Source: Alstom.)



FIG. A-88 Wet FGD plant with one single absorber installed after a 700-MW coal-fired boiler at Asnæsværket utility in Kalundborg, Denmark. The plant produces commercial quality gypsum. (Joint venture Alstom and Deutsche Babcock Anlagen.) (Source: Alstom.)

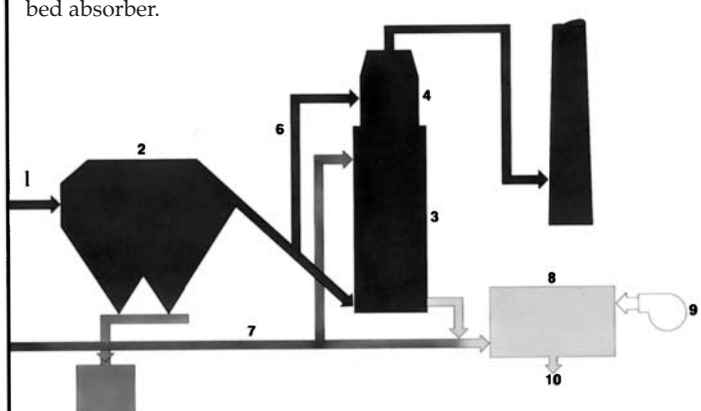


FIG. A-89 Seawater FGD plant at Tata Industries, India. (Source: Alstom.)

Seawater FGD

The wet FGD process can utilize the alkalinity of seawater to absorb SO2 in the flue gas. Absorption takes place in a once through packed bed absorber.

The effluent is aerated in a seawater treatment plant and mixed with cooling water from the condensers before disposal at sea.



1. Flue gas from boiler
2. Fly ash collector
3. Scrubber
4. Reheat unit
5. Fly ash silo
6. Flue gas for reheat
7. Seawater from condensers
8. Water treatment plant
9. Air blower
10. Effluent seawater

$$SO_2 + H_2O + 1/2\ O_2 \rightarrow H_2SO_4$$

FIG. A-90 Basic operation of seawater FGD. (Source: Alstom.)

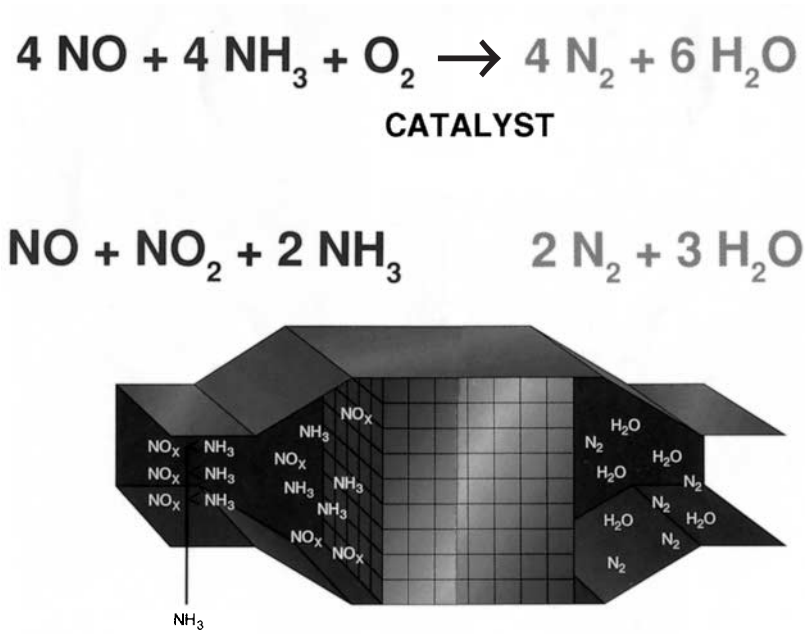
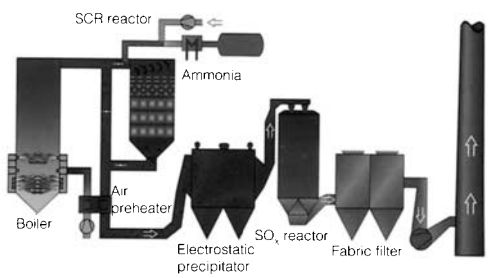
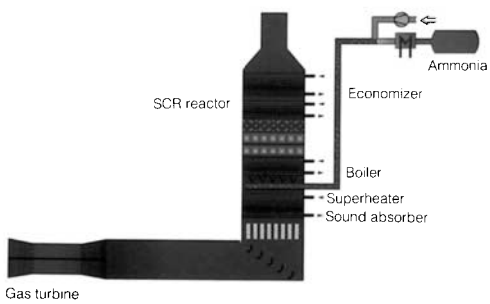


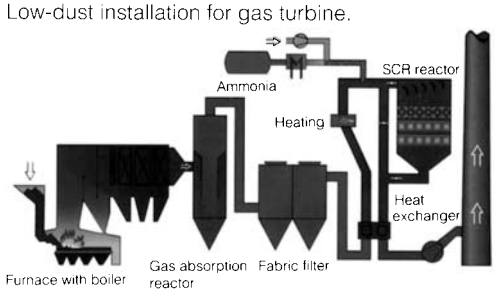
FIG. A-91 The reduction of nitrogen oxides with ammonia is achieved at a manageable temperature by the use of a catalyst. (Source: Alstom.)



High-dust installation in boiler plant.



Low-dust installation for gas turbine.



Tail end installation for waste incineration.

FIG. A-92 Flow diagrams for different DeNO$_x$ process systems. (Source: Alstom.)

1. **High dust system.** The reactor is placed before the air preheater, and operates directly in the dust-laden and acidic gas that leaves the boiler. This system dominates the fossil fuel boiler market today. (See Fig. A-93.)
2. **Low dust system.** The reactor is placed after the hot electrostatic precipitator but before the air preheater, which is possible, for example, in the case of waste incineration or, in the case of a gas turbine, in the heat recovery boiler.
3. **Tail end system.** The reactor is placed after particulate control and after sulfur dioxide and/or hydrochloric acid removal in the flue gas cleaning train. This allows for the use of a much more compact catalyst reactor. The tail end solution is used when the particulates or gases are harmful to the catalyst. (See Fig. A-94.)

SCR reactor design

Although design and operation of an SCR reactor is fairly straightforward and simple, there are a few issues that require special attention. One such issue is gas distribution at the inlet of the reactor. In the case of high dust it is of the utmost importance that the gas is properly distributed to avoid catalyst erosion problems. The reactor is equipped with guide vanes and distributor plates to ensure even gas distribution under all operating conditions. (See Fig. A-95.)

Another important issue is ammonia slip, which must be kept to a minimum for several reasons.

If the gas still contains sulfuric gases, i.e., in the high dust case, ammonia slip will react with sulfur trioxide to form ammonia bisulfate when cooled in the air

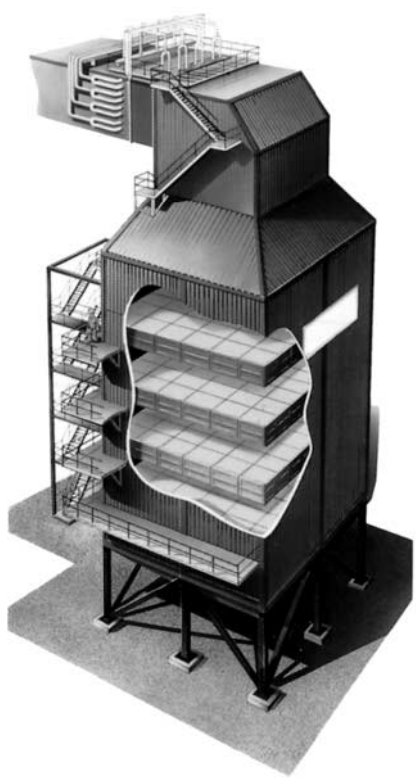


FIG. A-93 SCR reactor of the high-dust type. (Source: Alstom.)



FIG. A-94 This coal-fired 550-MWe/900-MWth combined heat and power plant is located centrally in the town of Västerås, Sweden. Alstom has gradually extended its flue gas treatment system, which today comprises ESP, FGD, and SCR units for full emission control. (Source: Alstom.)

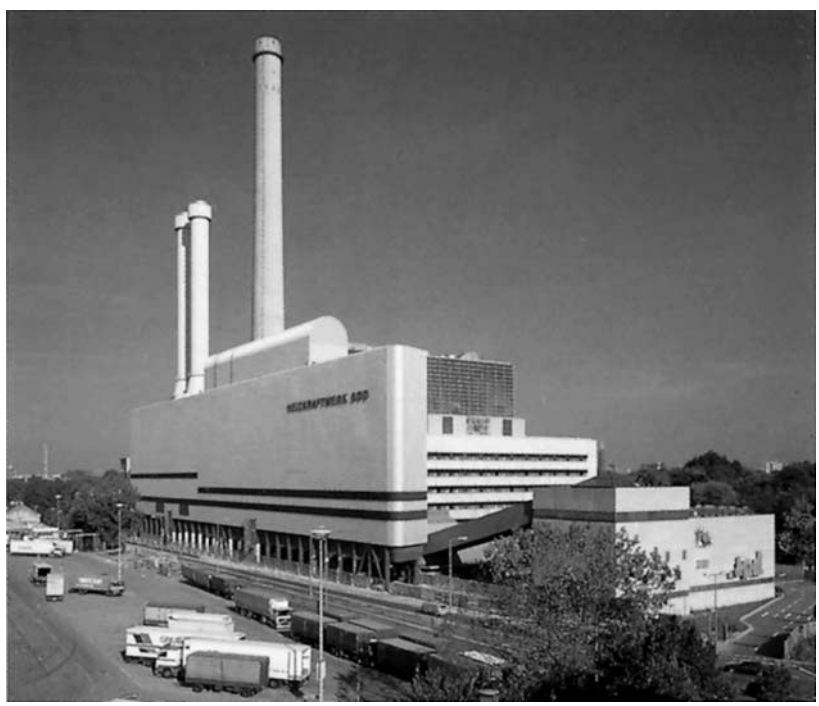


FIG. A-95 Stadtwerke München Süd, Germany, has installed a CDAS (Conditioned Dry Absorption System), as well as a tail-end-type SCR unit, at its 300,000-tons-a-year waste-to-energy plant. (Source: Alstom.)

preheater. This formation will take place on the dust particles and make the fly ash unsuitable for direct use in concrete manufacturing.

The ammonia will also end up in the effluent from a downstream FGD plant, requiring effluent treatment.

The SCR plant is therefore equipped with a control system of the "feed forward-trim back" type. In this system, ammonia is injected before the catalyst in relation to both the measured NO content after the reactor and the amount of NO present in the gas fed into the reactor.

Catalyst activity will inevitably decrease with time. The mechanisms that control the rate of deactivation are mainly: (i) *sintering* of the microsurface due to elevated temperatures, (ii) *poisoning* of the active metal atoms or molecules through a permanent bond or reaction with, for example, alkali metals, and (iii) *blocking* of pores by, for example, ammonia bisulfite or dust.

The reactor is equipped with a spare layer that can be charged when the efficiency of the catalyst has dropped below a certain level. When the activity drops further, the catalyst has to be replaced.

The obvious advantages with the tail end system, with favorable conditions regarding all three deactivation parameters, are offset only by the cost of bringing the gas temperature back to the elevated operation temperature of the catalyst. This manufacturer is also conducting research to find catalysts with lower operation temperatures for various applications.

Absorbers for Environmental Toxins

The dioxin and heavy metal problem

This refers to the entrapment of dioxins in dry scrubbers. The experiments in this field were first conducted in gas cleaning systems for waste-to-energy plants, where dioxin emissions are a major problem. The emission control system described combines the dual effect of chemically enhanced adsorption/absorption and filtration.

The fabric filter is also very effective for controlling heavy metals, due to its capacity for filtering submicron particles. The combination of dioxin and heavy metal abatement has been especially important for the environmental acceptance of waste-to-energy plants. This manufacturer has developed the TCR (Total Cleaning and Recycling) concept for complete control of flue gases from waste-to-energy installations.

More than 50 Filsorption plants have been installed in Europe and the United States. These plants repeatedly measure dioxin emissions below 0.1 ng/Nm^3. (See Fig. A-96.)

The latest development is the Filsorption® II system, which introduces a mixture of lime and coke in a safe blend to enhance abatement of organic emissions, primarily dioxins, heavy metals, and acidic gases.

Filtration and chemisorption (Filsorption® II)

Filsorption is short for filtration and chemisorption, indicating the dual duty of the system. The Filsorption II system is primarily aimed at the control of organic emissions such as dioxins. The system also offers control of mercury, acidic gases, and particulate emissions. Filsorption is an "absolute filter" for securing very low emission levels.

The system includes a storage and injection system for the chemically active



FIG. A-96 Hazardous waste incineration plant with Alstom Filsorption system, Cleanaway Ltd., Ellesmere Port, UK. (Source: Alstom.)

sorbent. The sorbent is normally a mixture of coke and lime, mixed to a safe blend. This is a significant advantage of this system, which reduces risks for operating personnel.

The fabric filter collects the particulate matter that escapes the upstream particulate control units along with the injected sorbents and reaction products.

The fabric filter also acts as a chemical reactor for lime with SO_2, SO_3, HCl, and HF.

The ash collected in the filter is discharged for final handling or recirculation back to the combustion unit to destroy its organic contents. The contaminated reaction product requires a dust-free handling system. This information source provides an underpressure conveying system to prevent potentially hazardous ash from contaminating the working area.

The Filsorption system can be used for treatment of gases containing dioxins and heavy metals from other types of industrial processes besides waste-to-energy installations. (See Figs. A-97 to A-103.)

End-Products Handling Systems

The often substantial amounts of solid or liquid wastes resulting from emission control systems require careful and efficient handling systems within the plant itself, as well as environmentally sound methods of treatment, recycling or disposal.

Ash-handling technology needs various types of solids handling, including bottom ash submerged drag chain conveyors, wet impounded hoppers, economizer and pyrites systems, and pneumatic fly ash handling.

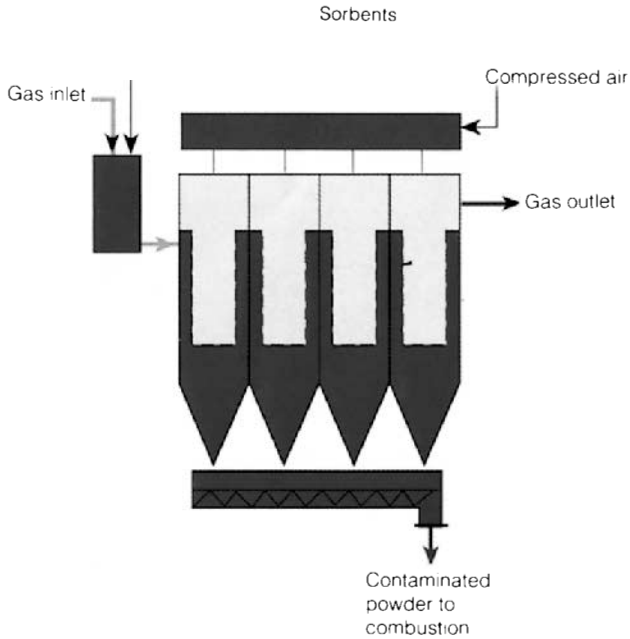


FIG. A-97 Schematic for filtration and chemisorption unit (Filsorption II). (Source: Alstom.)



FIG. A-98 Stabilized gypsum from a wet FGD plant. The end product is suitable for landfill use. (Source: Alstom.)

The DEPAC® system illustrated is a pneumatic conveying system based on dense phase technology.

Cost-efficient methods are developed to utilize the solid waste products. A number of commercial operations have already been established such as commercial grade gypsum for wallboard manufacturing and high-strength fill materials.

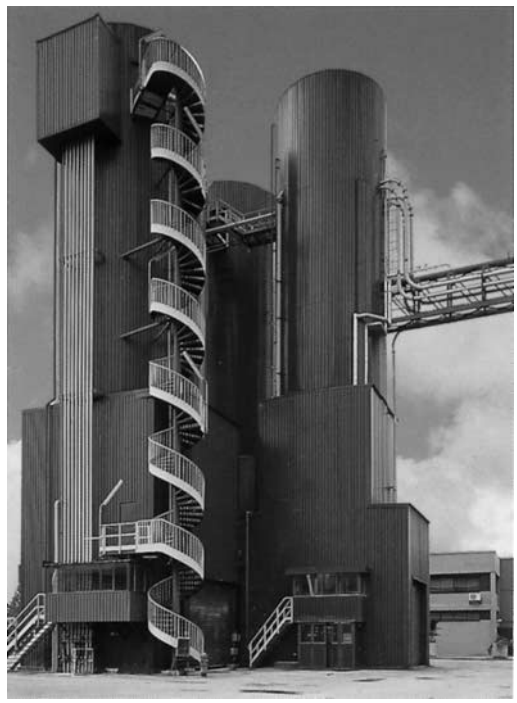


FIG. A-99 Group of silos for short-term storage of reagents and reaction products from an FGD plant. (Source: Alstom.)



FIG. A-100 The heart of the Fläkt DEPAC system is the dust transmitter, in which the product is fluidized by means of compressed air. (Source: Alstom.)

Combining Unit Operations

Total turnkey solutions for emission problems combine the operations of various units. The example illustrated is the TCR system. This system is designed to clean flue gas from waste incineration and produce certain recyclable end products.

In a similar manner, units are combined to form complete flue gas treatment trains in modern power plants.

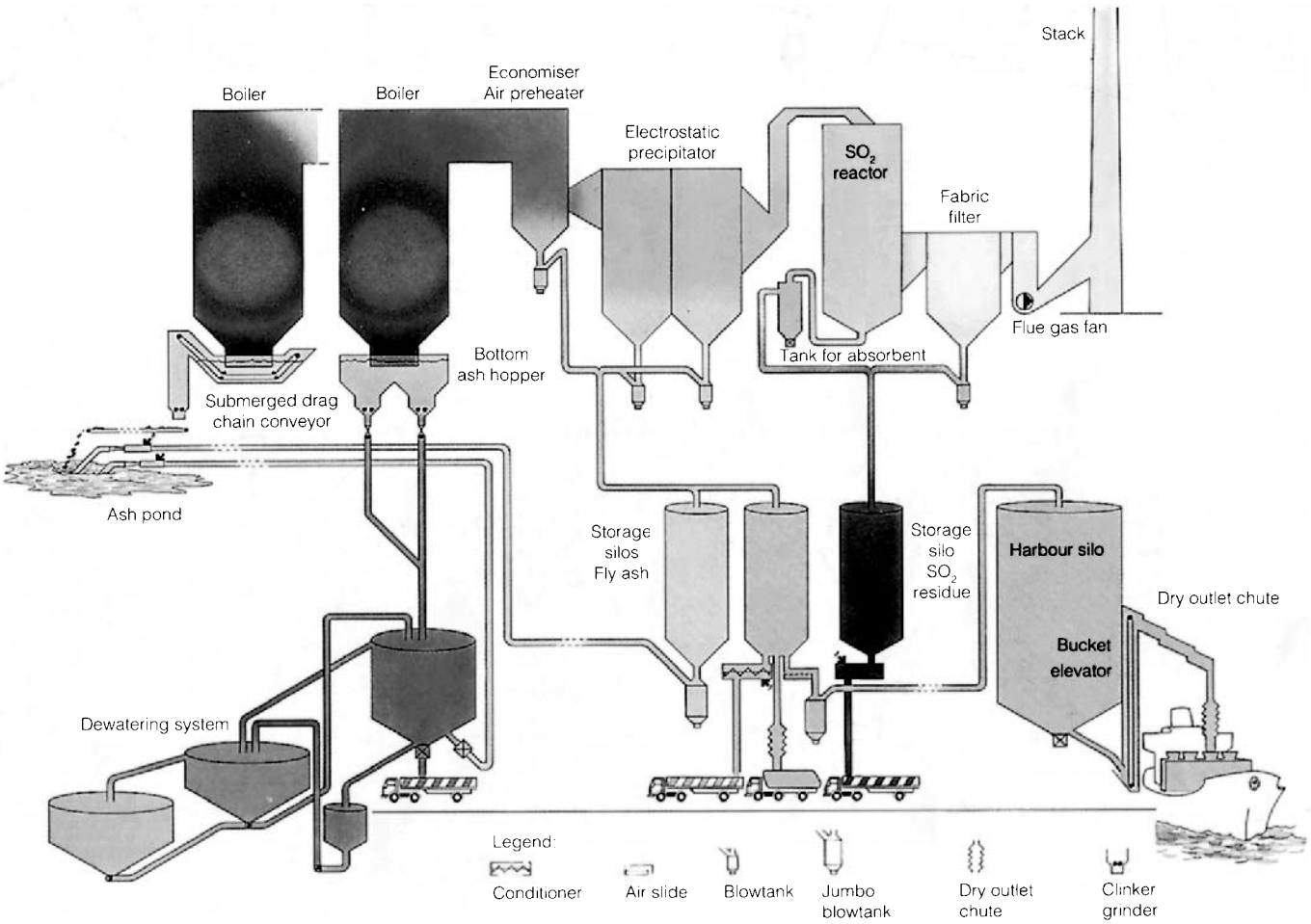


FIG. A-101 Schematic for end-products handling for boiler fixed plant. (Source: Alstom.)

Such treatment trains could combine $DeNO_x$ and wet FGD units in, for example, a large coal-fired power plant with an ESP for full emission control of the flue gases. Because emissions of heavy metals such as mercury and cadmium are common problems associated with coal firing, Filsorption units may be utilized to curb such emissions.

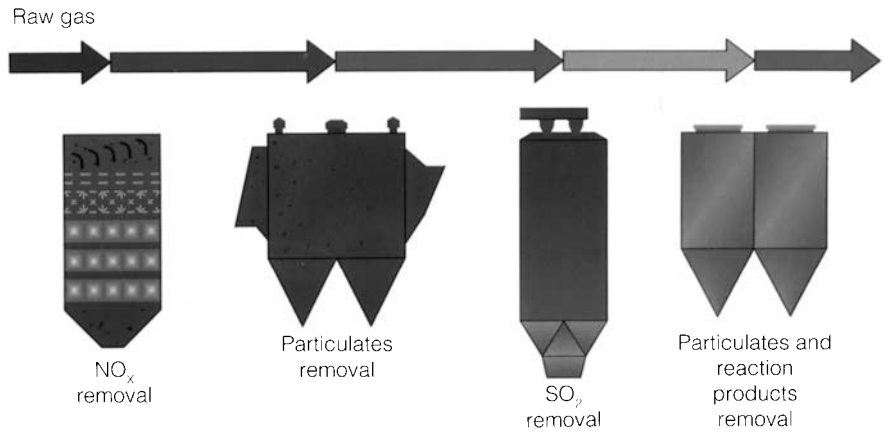
Also industrial application often contains several unit operations for full emission control. A steel process plant, for example, has many emission sources that all require a separate solution.

TCR

The TCR system contains basically three unit operations: (i) *Filpac,* (ii) *Wetpac*, and (iii) *Catpac*.

The Filpac stage (shown here is the Filsorption II process) is used for separation of submicron particles, heavy metals, and toxic hydrocarbons by a combination of filtering, "sorption," and chemical reaction.

The Wetpac® process is an absorption stage collecting the acid gas components and producing recyclable products, such as hydrochloric acid, chloride, and sulfate compounds.



The above unit processes can be combined into a flue gas treatment train (below).

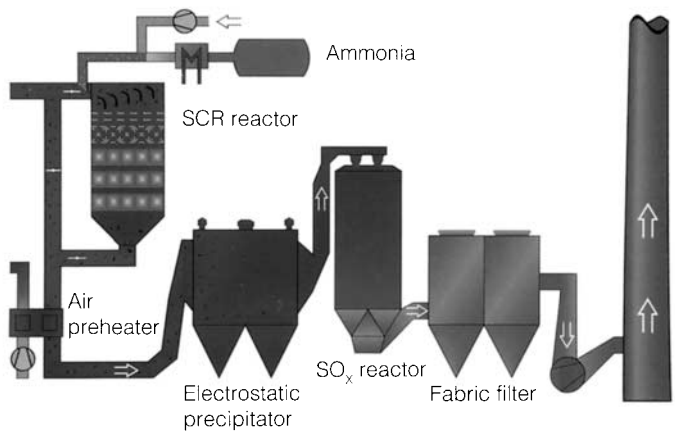


FIG. A-102 Combined NO_x and FGD units. (Source: Alstom.)

Catpac is used to reduce nitrogen oxides but may also incinerate hydrocarbons or dioxins.

With the TCR approach we combine well-proven unit operations into modularly built and fully optimized APC solutions for true eco-engineering.

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.

Air Purification; Air Sterilization

Certain specialized processes require air that is a great deal cleaner than outlined in the previous section on air pollution control. Examples include food processing and pharmaceuticals production. The detailed methodology needs to be worked out with equipment vendors, but basically it involves:

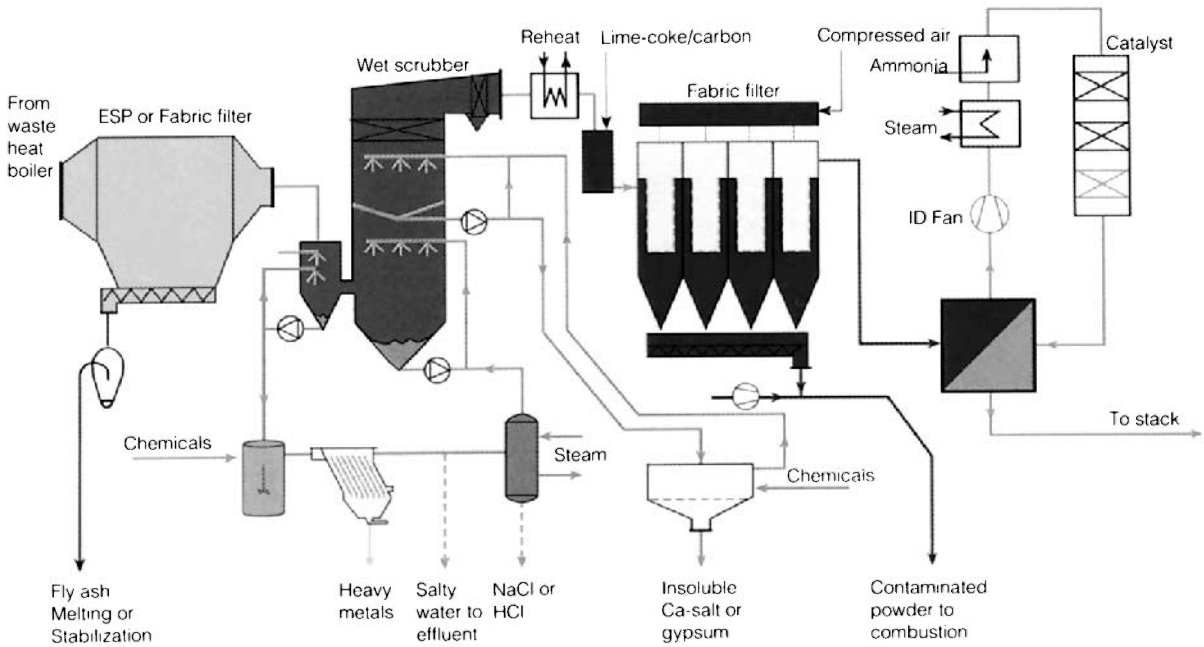


FIG. A-103 Unit for separating submicron particles, heavy metals, and toxic hydrocarbons. (Source: Alstom.)

1. Filtration (the end process may require removal down to 0.1 μm; 1 to 5 μm is common).
2. Electrostatic precipitation (can remove up to 90% plus of the particles in the air).
3. Air washing (can remove between 50 and 80% plus of the microorganisms in the air).
4. Ultraviolet irradiation. Different microorganisms have different sensitivities.
5. Heat and compression. Heat helps the sterilization process. Compression produces work, which, in turn, produces heat that also can contribute to sterilization.

B

Balancing; Onspeed Balancing of a Rotor

Balancing generally refers to the balancing of a turbomachinery rotor. Balancing can, in some cases, be done in the field. Maintenance staff can be trained, for instance, to balance a pump in situ in the plant, if their readings with their vibration analysis equipment confirm that this is what needs to be done. For more critical items, such as process compressors, this process is best done in the overhaul facility of the original equipment manufacturer (OEM). The exception to this would be if the end user had his or her own balance equipment and had trained staff that was capable of handling the rotor in question.

Most balancing of rotating machinery rotors or components of rotors (such as turbine wheels and so forth) is done in a balancing machine at speeds in the neighborhood of 1800 to 2000 r/min in atmospheric conditions. In certain rare instances, balancing at these speeds does not remove the imbalance (that was causing rotor vibration in the first place). As a last resort, the process engineer may have to specify onspeed balancing of this rotor. This needs to be done in a vacuum chamber and is expensive. Also, there are very few suitable vacuum test facilities in the world. Before getting into this additional expense, the process engineer is probably well advised to consult a rotating machinery engineer.

Balancing Problems, Troubleshooting (Turbomachinery) (*see* Condition Monitoring)

Batteries (*see* Cells)

Bearings* (*see also* Lubrication)

Bearings permit relative motion to occur between two machine elements. Two types of relative motion are possible, rolling or sliding, each of which depends upon the design of the mechanical bearing element. Thus bearings are classified into two general types: the rolling-contact type (rolling) and the sliding-contact bearing design in which the bearing elements are separated by a film of oil (sliding). Both can be designed to accommodate axial and/or radial loads. Each has a wide variety of types and designs to fit a wide variation in uses. The selection of a bearing type for application to a particular situation involves a performance evaluation and cost consideration.

There is ample literature available to determine the relative merits of each type. This subsection will provide a general overview of the types of bearings presently encountered in the equipment covered in this edition. Changes take place frequently because many scientists and engineers work constantly to improve the state of the art in bearing design.

* Source: Demag Delaval, USA.

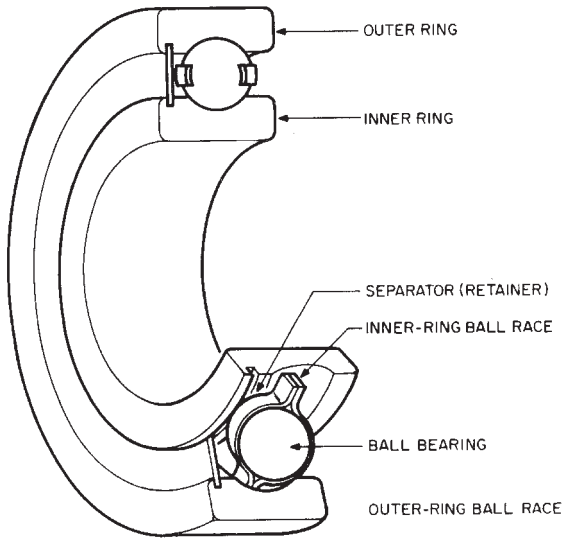


FIG. B-1 Nomenclature of a ball bearing. (Source: Demag Delaval.)

Rolling-Contact Bearings

Rolling-contact bearings include ball bearings, roller bearings, and needle bearings. Within each category several variations have been developed for specific applications. Variations in the amounts of radial and thrust load capabilities also exist between specific types. Self-aligning ball or roller bearings, by virtue of their spherically ground outer race, can tolerate misalignment of the shaft or housing.

Rolling-contact bearings consist of four principal components: an outer race, an inner race, rolling elements, and a separator, or spacer, for the rolling elements. The inner ring is mounted on the shaft. The outer ring securely fits in a stationary housing. The facing surfaces of the inner and outer rings are grooved to conform to the rolling-element shape. The rolling elements (with separator) accurately space the inner and outer races and thus enable smooth relative motion to occur (see Fig. B-1).

Sliding-Contact Bearings

Sliding-contact bearings are classified into two general types: journal bearings and thrust bearings. Journal bearings support radial loads imparted by the rotating shaft and may also be required to arrest or eliminate hydraulic instabilities that may be encountered in lightly loaded high-speed machinery. The thrust bearings are used for loads parallel to the shaft and may be required to support the full weight of the rotor in cases of vertical machinery.

Journal Bearings

The common types of journal bearings are:

- Plain journal bearings
- Three-lobe journal bearings
- Tilting-pad journal bearings

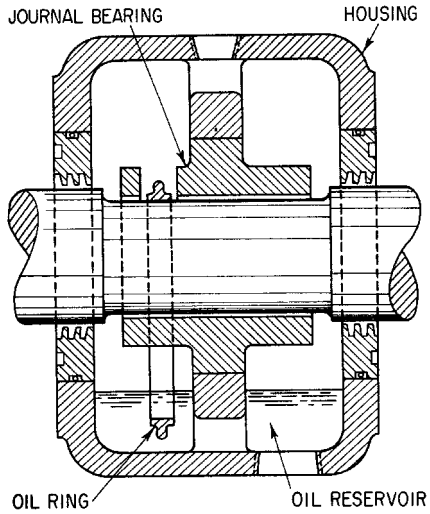


FIG. B-2 Ring-oil lubrication. (Source: Demag Delaval.)

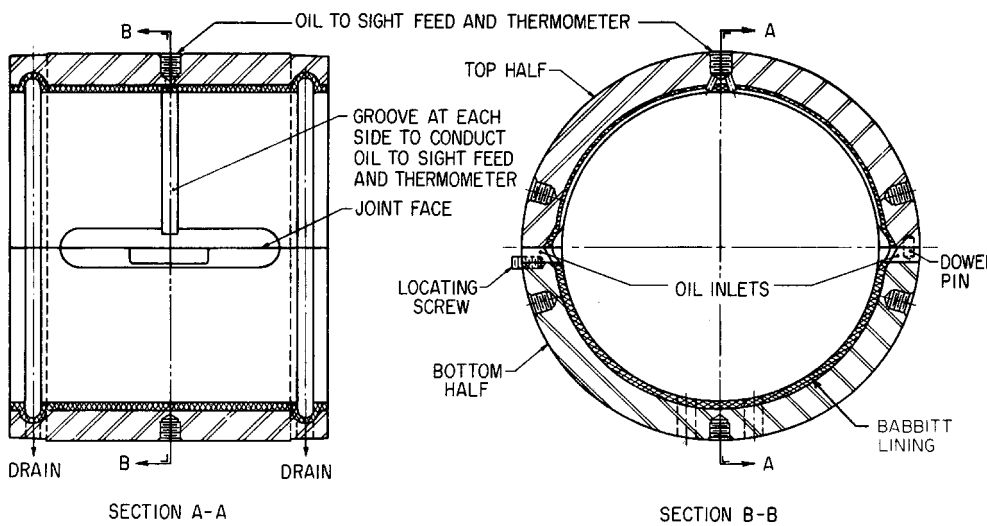


FIG. B-3 Typical design of journal bearing. (Source: Demag Delaval.)

The *plain journal bearing* may be either force-fed from a pressure-lubrication system or ring-oiled by means of a free ring that rests on and rotates with the shaft to serve as a pumping medium, as shown in Fig. B-2. A section of a typical force-fed journal bearing is shown in Fig. B-3. This shows a babbitt-lined split-type bearing. Journal-bearing clearance for high-speed machinery (i.e., turbines, centrifugal compressors, pumps, etc.) should be not less than 0.0015 in/in of diameter for satisfactory operation. However, a complete analysis should be made to obtain the proper clearance for each case.

The *three-lobe journal bearing* shown in Fig. B-4 is applied on high-speed machinery, usually when the bearing is end-fed and used in combination with a face-contact seal or when the bearing itself is a shaft seal. The three-lobe design provides, intrinsically, a stable three-point support that is particularly adaptable to very light loads in either horizontal or vertical shafts. Because of the small

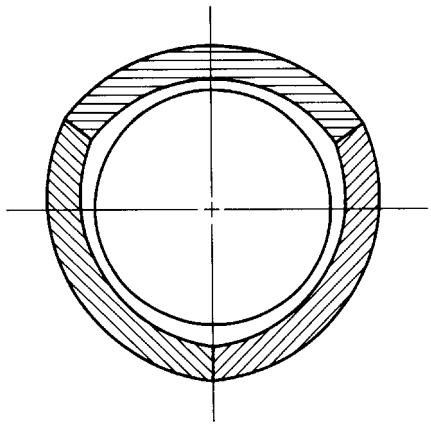


FIG. B-4 Sketch of a three-lobe journal bearing. (Source: Demag Delaval.)

amount of additional clearance, the end-leakage effect of this bearing can be well controlled and provides good impurity tolerance from the oil system with a means for continuous flushing.

Another design of a three-lobe bearing has three equal-diameter bores that are displaced 120° from each other by an amount equal to the bearing clearance less the oil-film thickness. This bearing is center-fed. It is used to support lightly loaded shafts, especially light gear shafts in which the load direction varies with the transmitted torque.

The *tilting-pad journal bearing* consists of three or more separate shoes equally spaced around the circumference of the journal. Its mechanical design and performance follow closely the theory of the tilting-pad thrust bearing. A typical bearing of this kind utilizes five pads, of which one pad is on the bottom, dead center for horizontal shafts, to provide a fixed means of supporting the shaft during alignment. This bearing is considered to be the best design for attaining rotor stability. It also is inherently more tolerant of impurities in the oil system without bearing failure. A typical section of this bearing is shown in Fig. B-5. This bearing is pressure-fed from a circulation system, and the amount of oil passing through the bearing is controlled by a restriction at the inlet to the bearing shell.

Thrust Bearings

The simplest type of thrust bearing is the sliding-contact type with plain babbitted face, which is used in lightly loaded thrust applications. The second, more complicated design for intermediate thrust loading is found in the tapered-land thrust bearing. This design has a multiple radial series of fixed lands having a tapered inlet edge in the direction of rotation to develop a hydrodynamic film to support the thrust load.

The most common thrust bearing found in high-speed machinery is the tilting-pad thrust bearing. This bearing contains independent-acting shoes with a hardened, pivoted pad backing. Each shoe acts to establish a hydrodynamic film proportional to the speed and loading of the shaft to support the thrust load. This bearing may be single-sided in the case of turbines in which unidirectional thrust is encountered or double-sided in the case of compressors or pumps in which the thrust loading may shift, depending on the operating conditions.

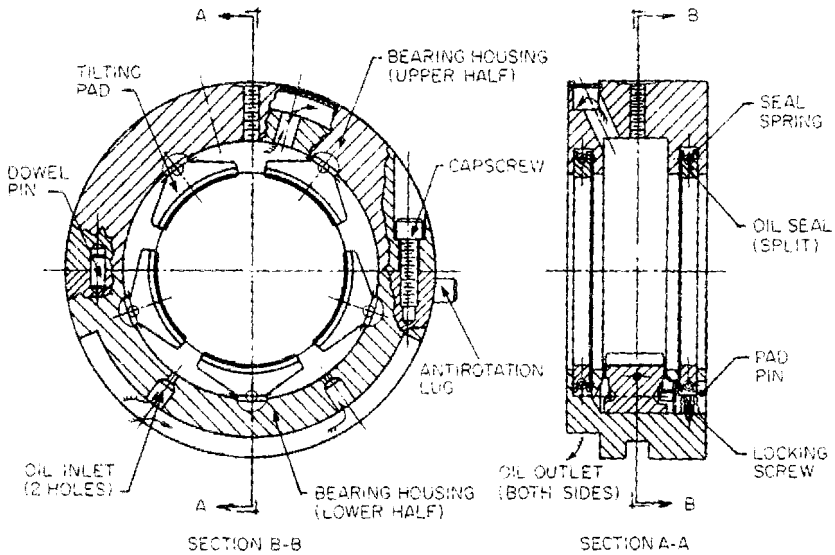


FIG. B-5 Typical tilting-pad bearing. (Source: Demag Delaval.)

Magnetic Bearings*

Magnetic bearing systems represent a completely different approach to the support of rotating equipment. They are a *noncontacting technology*, which means negligible friction loss and wear and higher reliability. It also enables previously unachievable surface speeds to be attained. Lubrication is eliminated, meaning that these bearings can be incorporated into processes that are sensitive to contamination, such as the vacuum chambers in which many semiconductor manufacturing processes take place.

Magnetic bearings incorporate three distinct technologies: *actuators and sensors* are the electromechanical hardware by which input signals are collected and supporting forces applied to the machine on which they are installed. The *control system* provides the power and control electronics for signal conditioning prior to calculation of correcting forces and resultant commands to the power amplifiers for each axis of control. *Control algorithms* are the software programs used in digital magnetic bearing system control including the processing of the input signals after conditioning and calculation of the command signals to the power amplifiers.

System components and operation

Magnetic bearings provide *attractive electromagnetic suspension* by application of electric current to ferromagnetic materials used in both the stationary and rotating parts (the *stator* and *rotor*, respectively) of the magnetic bearing *actuator*. This creates a flux path that includes both parts, and the air gap separating them, through which noncontact operation is made possible.

As the air gap between these two parts decreases, the attractive forces increase; electromagnetic bearings are inherently unstable. A *control system* (Fig. B-6) is needed to regulate the current and provide stability of the forces and rotor position.

---

*Source: Revolve, Canada. Adapted with permission.

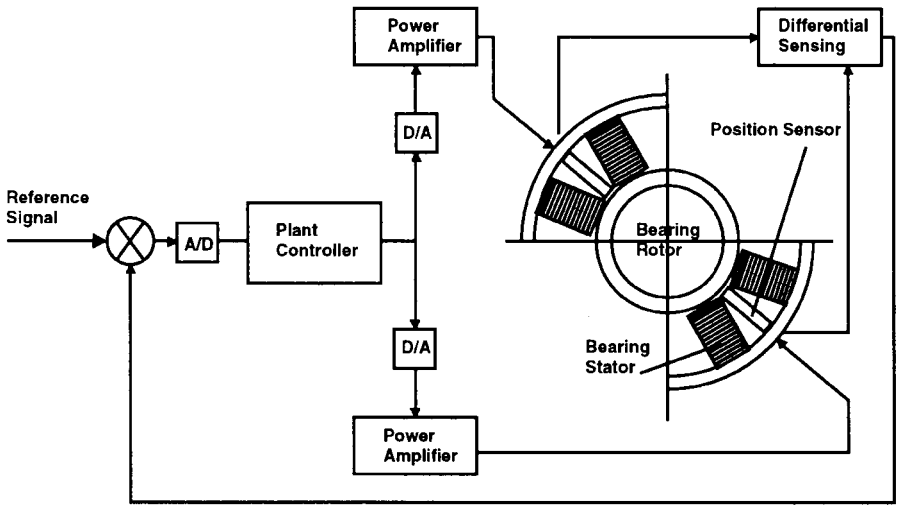


FIG. B-6 Schematic for a magnetic bearing. (Source: Revolve.)

The control process begins by measurement of rotor position with a *position sensor*. The signal from this device is received by the control electronics, which compare it to the desired position programmed during machine testing. Any difference between these two signals results in calculation of the forces necessary to pull the rotor back into the desired position. This is translated into a command to the *power amplifier* connected to the magnet actuator. The current is increased, causing an increase in magnetic flux, an increase in the forces between the rotating and stationary components, and, finally, movement of the rotor toward the stator along the *axis of control*.

The entire process is repeated thousands of times every second, enabling precise control of machinery rotating at speeds in excess of 100,000 rpm.

Actuators and sensors. This represents closed-loop control of one axis on a radial bearing. To provide support in more than one direction, magnetic poles are oriented about the periphery of a radial bearing. This is shown in Fig. B-7A, along with a photograph of a radial magnetic bearing rotor and stator (Fig. B-7B).

Radial actuator construction is very similar to that of an electric motor, involving the use of stacked laminations of steel, around which power coils are wound. Stacked laminations are also used in the rotor to minimize eddy current losses, which cause heating and otherwise represent inefficiencies between power supplied to the bearings, and the useful work performed by the bearing system.

The sensors are also oriented about the periphery of the stator, usually inside a ring that is mounted adjacent to the actuator poles. The photograph shows the sensor ring mounted on the end face of the radial stator. Inductive sensors are used that measure the inductance of the gap between the sensor and the rotor laminations. Two measurements are taken for each radial axis and the rotor center position calculated by means of a bridge circuit.

A typical rotating machine will experience forces in both the radial and axial directions. Typically, a five-axis orientation of bearings is used, incorporating two radial bearings of two axes each and one thrust bearing. The orientation of axes is shown in Fig. B-8.

Thrust bearings provide a magnetic flux path in the axial direction, between two stators oriented on either side of a *thrust rotor*, or *disc*, mounted on the

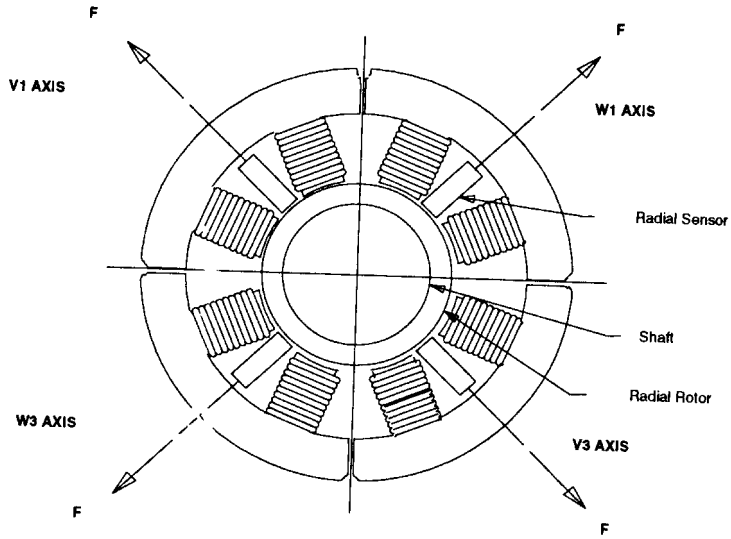


FIG. B-7A Magnetic poles oriented about the periphery of a radial bearing. (Source: Revolve.)

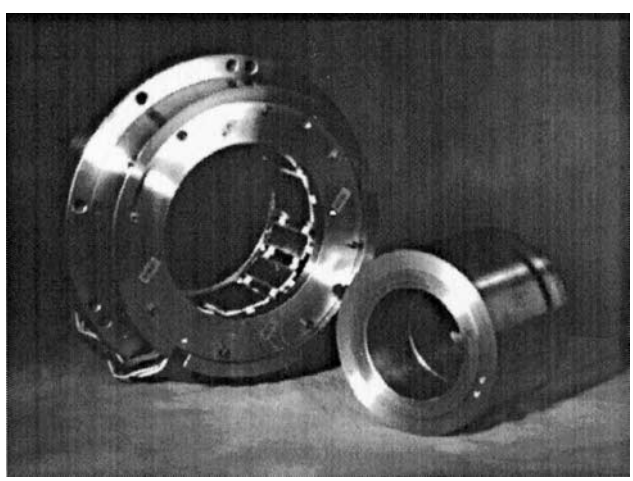


FIG. B-7B Radial magnetic bearing rotor and stator. (Source: Revolve.)

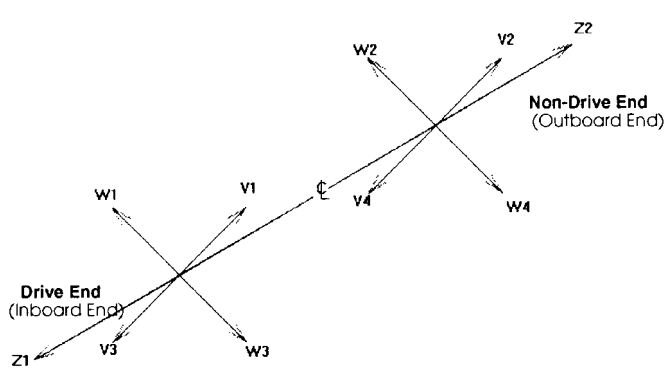


FIG. B-8 Vector diagram of bearing forces. (Source: Revolve.)

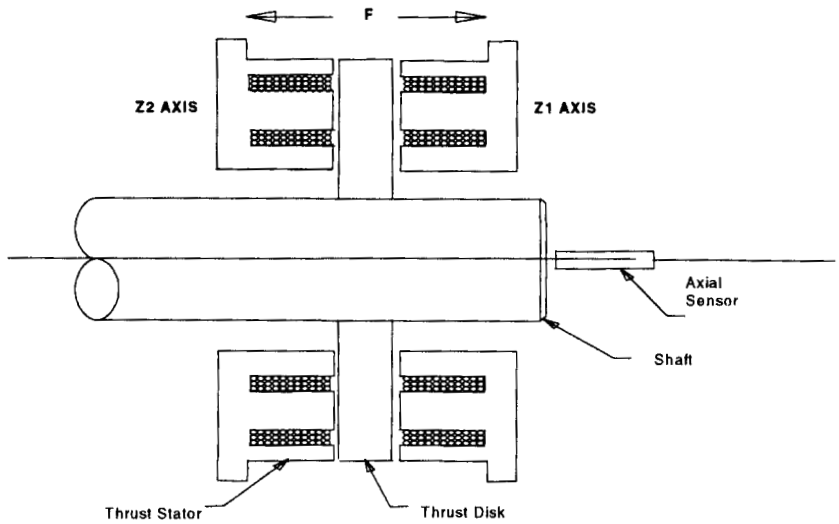


FIG. B-9A Thrust bearing schematic. (Source: Revolve.)

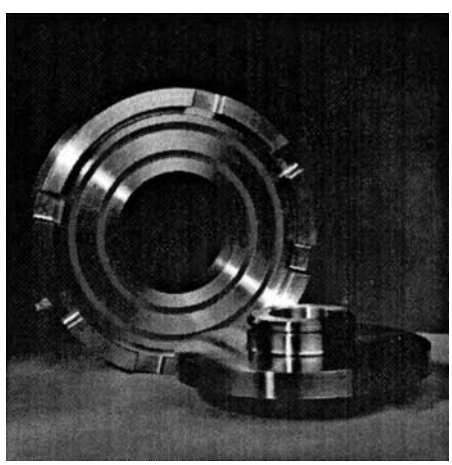


FIG. B-9B Thrust bearing. (Source: Revolve.)

rotating shaft as shown in Fig. B-9. An axial sensor measures the position of the shaft.

Control system. The control system consists of the *plant controller* (or control head), *power amplifiers*, and a *power supply*. Additional circuitry is present for conditioning of the signals from the position sensors and conversion of the digital



FIG. B-10A Low load digital control system. (Source: Revolve.)



FIG. B-10B High load digital control system. (Source: Revolve.)

outputs from the plant controller to the analog amplifiers. Finally, a *user interface* allows input of the desired rotor position information and logic to be coordinated with other machine systems (startup, warmup, shutdown, etc.).

Control systems for magnetic bearing systems can be of the analog or digital type. *Analog control systems* have been used in magnetic bearing control for over 30 years, but are rapidly being displaced by digital control systems, suitable for low load applications (Fig. B-10A) and high load (Fig. B-10B).

Low load control systems provide as much as 1 kW of power for applications such as turbomolecular pumps in an enclosure about the size of a shoe box. Higher load

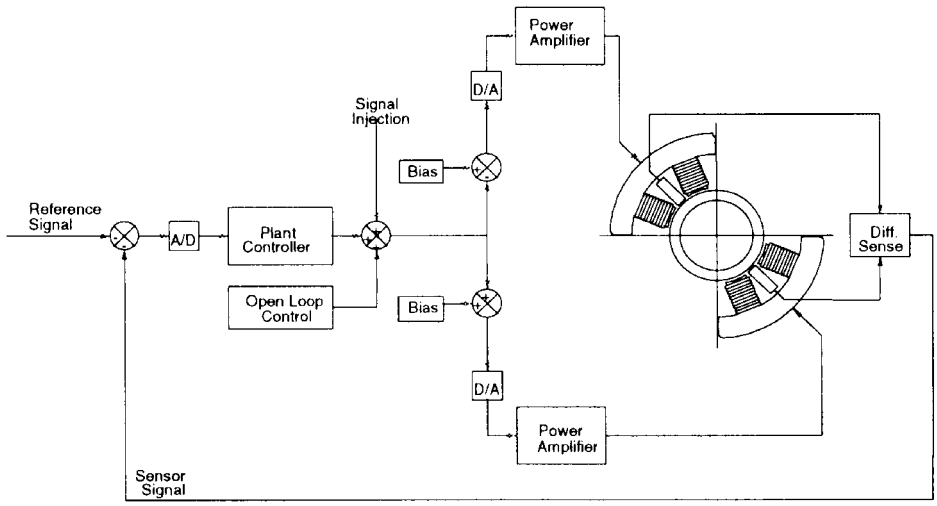


FIG. B-11 Open-loop control system. (Source: Revolve.)

applications include large-frame turbomachinery, such as centrifugal compressors, pumps, and turbines, and require up to 10 kW of power. These systems are designed to meet industrial electronic rack standards.

Control algorithm. For any particular bearing system, a control scheme is required that takes into account the behavior of the suspended (rotating) weight over the operating range (speed, acceleration, dynamic forces). These characteristics are revealed during a *rotor dynamic analysis* of the system. The knowledge gained results in composition of a *transfer function* for the control system that enables stability to be achieved over the desired operating range.

This transfer function is present in analog control systems in the form of hard-wired control circuits, which must be designed and built for each specific purpose or application. Their tolerance to system changes is limited, and modifications or changes are difficult and expensive to make.

Digital control systems allow transfer functions to be programmed in software, which can be modified or changed more easily and for less expense. They also allow *open-loop control* schemes to be used; these are control schemes that use actively changing external signals, as well as the reference and sensor signals used in closed-loop systems. (See Fig. B-11.)

Advantages and benefits

1. *Noncontacting*. Separation of the rotating and stationary parts means that the only friction is that provided by the air or other gas present. This is negligible compared to conventional oil or grease lubricants. Machinery operates more efficiently, with virtually no wear, requiring less maintenance and therefore achieving higher reliability.
2. *Contamination free*. Magnetic bearings can be used in processes that will not withstand the presence of any lubricant—such as refrigeration compression, pure gas processing, and vacuum processes such as those used in semiconductor manufacturing.
3. *Higher rotational speeds*. Noncontacting also means that higher rotational speeds are possible, as friction and surface galling are no longer factors to be

concerned with. Speeds are limited only by material stresses. This has significant implications for applications such as machine tool spindles, with potential for higher productivity and better surface finish quality.

4. *Power consumption*. Power consumed by a magnetic bearing system is minimal. Even for large turbomachinery, involving radial bearings supporting rotating assemblies of up to 15,000 N, power consumption is less than 4 kW. This is because the power is transferred back and forth between the bearing coils, and capacitors that straddle the power amplifiers. The only real power consumed is in resistance losses of the bearing coils and control system conductors, and eddy current losses in the bearing laminations. Compare this to over 250 kW of power consumed by a conventional system, which represents the sum of the lube oil shear losses, pump, and cooling fan power.

Blades and Vanes (for a Turbine) (*see* Metallurgy; Turbines)

Boilers

In conventional plants today, the term *boiler* is used synonymously with steam generator, which covers the boiler, the fuel system, and fuel firing mechanisms and controls, as well as all other accessories and controls.

Boiler Emissions

For allowable emissions limits in a sample country, *see* Emissions.

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.

Borescopes

A *borescope* is an instrument that, basically, uses fiber optics and flexible hose to "see" inside a critical piece of machinery. The objective with machinery is generally to run it "on condition" for as long as possible or, in other words, to use predictive instead of preventive maintenance (*see* Condition Monitoring). The borescope helps one do this by allowing maintenance personnel to look at the inside of a critical item of machinery. The OEM designers normally provide appropriate ports for this look. A qualified observer can then determine items such as: is there evidence of overtemperature on the turbine blades, are combustor can cross-rods cracking, is there erosion on fuel nozzles, and so forth? There is some distortion afforded by the fish eye at the end of the borescope's flexhose, but seasoned observers can interpret what they see. Interpreting borescope views is more an art than a detailed science.

Brakes

Hydrodynamic*

Hydrodynamic brakes are similar in construction to hydrodynamic couplings. The force or torque delivered by the machine to be braked is applied to the pump

*Source: J.M. Voith GmbH, Germany. Adapted with permission.

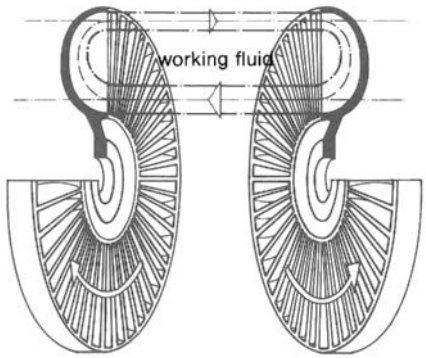


FIG. B-12 Hydrodynamic brake schematic. (Source: J.M. Voith GmbH.)

wheel and from there, by way of the working fluid, to the turbine wheel. (See Fig. B-12.)

The difference in the case of hydrodynamic brakes is that the turbine wheel is not free to rotate.

In order to achieve the highest possible braking force, the blades of the rotor and stator are set at an angle. The hydrodynamic brake model discussed in this section (made by Voith) achieves its braking power by retarding the inertial force of the working fluid that is accelerated radially and tangentially in the driven part (rotor) and decelerated in the nonrotating part (stator). The kinetic energy is thus transformed into heat that is taken up by the working fluid.

The braking process is initiated by filling the brake with working fluid and is ended by draining the fluid from the brake.

By varying the amount of fluid in the brake or the filling pressure, infinite variation of the braking torque within the performance range of the brake concerned can be obtained.

It is possible to influence the shape of the performance curve by various control and regulation means.

Control and energy dissipation

In hydrodynamic brakes, the working fluid is used simultaneously as a braking medium and as an energy carrier. In order to dissipate the heat energy generated by the braking process, a suitably dimensioned flow of working fluid with a defined input temperature must be ensured at all times.

A hydrodynamic brake can either be controlled with the working fluid in closed circuit or in open circuit and, by adjusting the amount of fluid in the brake, it is possible to work at any point within the performance range of the brake concerned.

A closed-circuit hydrodynamic brake (Fig. B-13) is directly connected with one or more heat exchangers and a given amount of working fluid once introduced into the system circulates continuously between the brake and the heat exchanger. To change the amount of fluid in the brake, fluid is either added to or withdrawn from the circuit by way of valves. This alters the pressure within the system and establishes the required performance within the working range.

With an open-circuit hydrodynamic brake (Fig. B-14) the working fluid is fed to the brake from a tank by means of a pump. The required amount of fluid in the brake is maintained by a control element in the feed line.

The heated working fluid flows back to the tank through the return line and a second circuit provides for cooling the fluid in the storage tank.

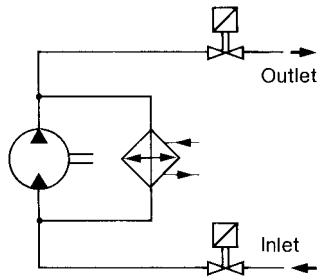


FIG. B-13 Closed-circuit hydrodynamic brake. (Source: J.M. Voith GmbH.)

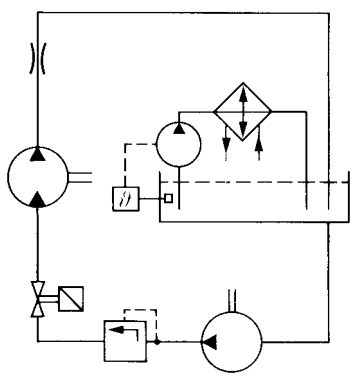


FIG. B-14 Open-circuit hydrodynamic brake. (Source: J.M. Voith GmbH.)

Where water is used as a working fluid it is possible for the water to be drawn direct from the water mains and for it to be discharged to the drains after use.

Hydrodynamic brakes are used wherever substantial masses have to be braked without the braking mechanism being subject to wear and where mechanical brakes would be overtaxed. Hydrodynamic brakes are particularly well suited to applications involving continuous operation. Hydrodynamic brakes have been used over several decades as a means of measuring power, particularly as absorption dynamometers (fluid friction brakes) in engine test stands.

Applications for hydrodynamic brakes

Test stands of all types, for instance

- Axle test stands
- Torque converter test stands
- Engine test stands
- Transmission test stands
- For the testing of hydrostatic units
- Ski slope servicing vehicle test stands

Coal crushers

Protective brakes in steel hydraulic mechanisms

Gas turbine drives

Deep drilling rigs
Downward sloping belt conveyors
Winding engine construction
Hydrostatic drives
Shipbuilding
Braked trailers (load simulators)

Advantages of hydrodynamic brakes

Simple construction
No fading
Virtually maintenance-free
Long service life
Low weight for high capacity
Small rotating masses and small inertia moment
Wear-free operation
Highly economic
Infinitely variable braking torque
Simple control options
High degree of stability of the set working point
Low and controllable working temperatures

Specific hydrodynamic brake models (VHBI, VHBM*: construction and variants)

VHBI hydrodynamic brakes. These brakes are characterized by performance with small overall dimensions as a result of the inclined blading of the rotor and stator. The hydrodynamic brake is effective in only one direction of rotation.

The housing elements, rotor, and stator are of spheroidal graphite iron or of material that is resistant to cavitation and corrosion. The shaft is of high-quality, heat-treatable steel and the shaft extensions are normally finished to DIN standards although other arrangements are possible. Oil is used as a lubricant for the bearings and radial-lip-type shaft seals or labyrinth seals are normally fitted. Mechanical end-face seals can be provided.

VHBM hydraulic brakes for power measurement. The VHBM is similar in construction and execution to the VHBI. The braking torque is transmitted through the housing, which is free to move by way of a lever to an electrical, pneumatic, or mechanical force measuring device while the speed of the shaft is detected at the same time and peripheral units provide for registering, processing, indicating, and recording the measurement data. Suitable control devices provide for adapting the brake characteristics to the application concerned.

Non-self-supporting models. Where the hydrodynamic brake is fitted in a machine or drive element, use is made of the existing bearings. The rotor is fitted overhung on the shaft and the braking torque is transmitted to the machine housing.

---

*VHBI and VHBM are trademark model designations associated with Voith brakes.

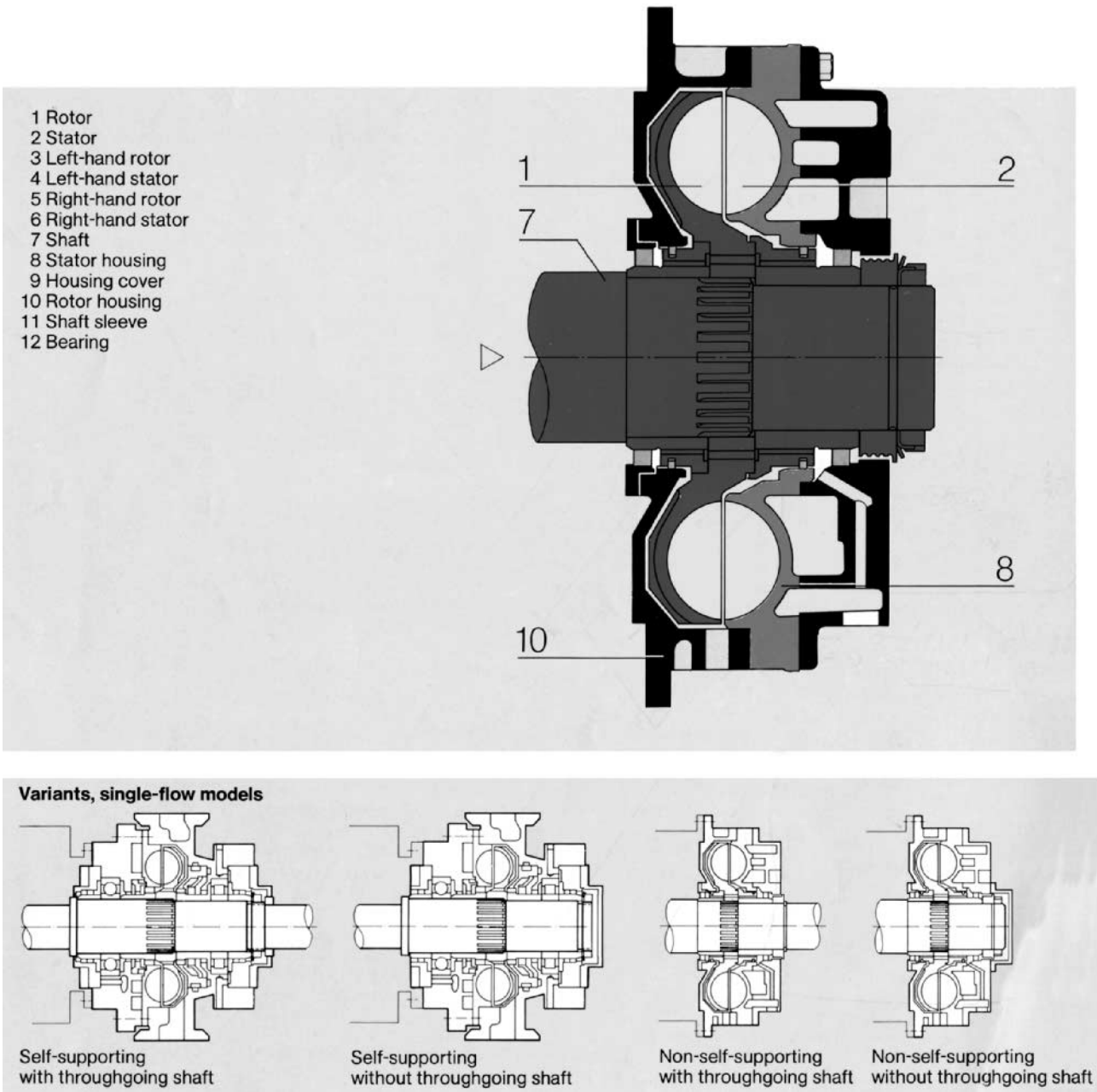


FIG. B-15 Hydrodynamic brake components and brake types (Voith VHBM type). (Source: J.M. Voith GmbH.)

Self-supporting models. Here the hydrodynamic brake is provided with its own bearings—roller bearings or, if requested, plain bearings. The hydrodynamic brake is either flange-mounted or separately mounted on the base plate as one of the elements in a drive assembly. See Fig. B-15.

After reaching a certain profile diameter, hydrodynamic brakes are supplied as double-flow models. Such models are indicated by a letter "D" in the type designation.

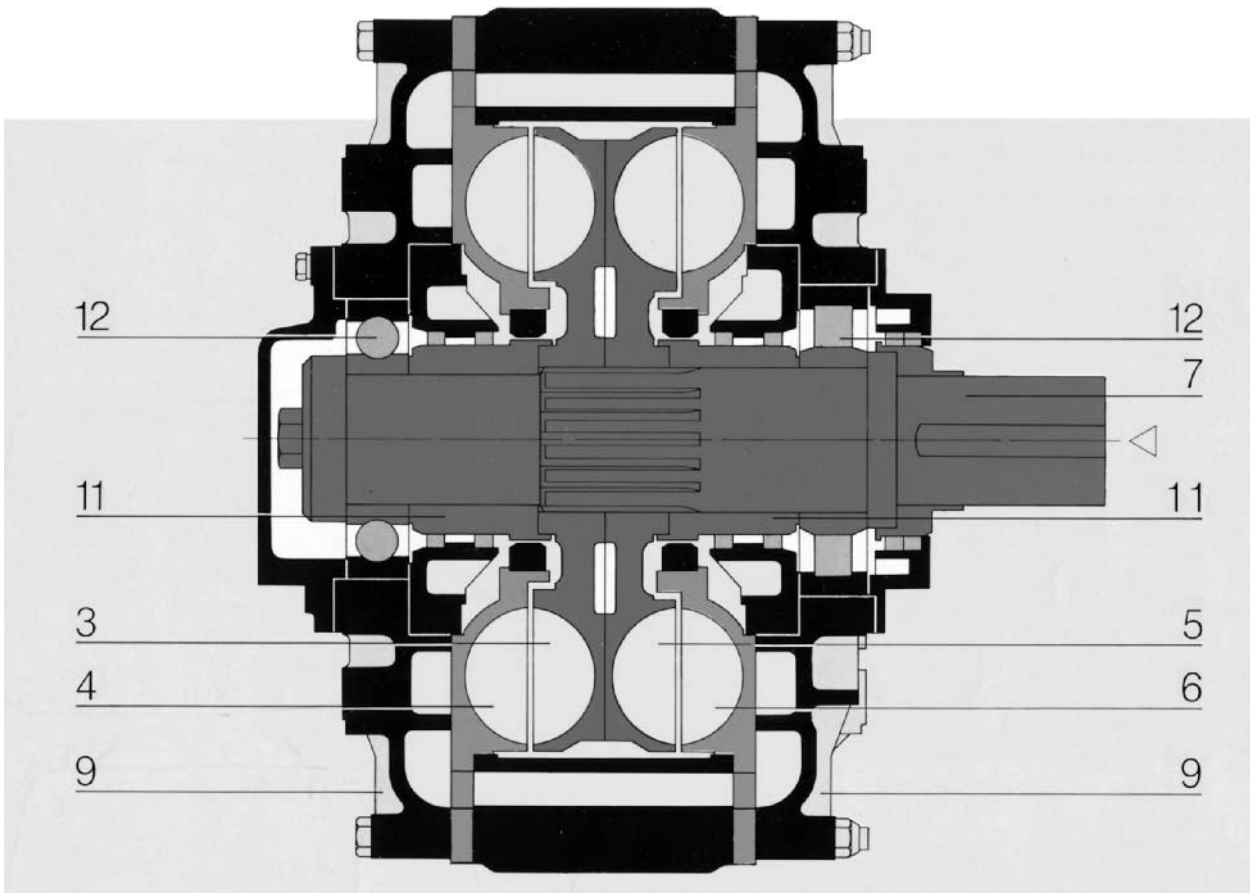


FIG. B-15 (*Continued*)

Controls for hydrodynamic brake types

The control schematics shown in Figs. B-16 through B-21 are some of the many practical applications of controls for hydrodynamic brakes.

Performance—braking torque

Separate performance characteristics are indicated in Figs. B-22 through B-25 for brakes using water and oil as a working medium.

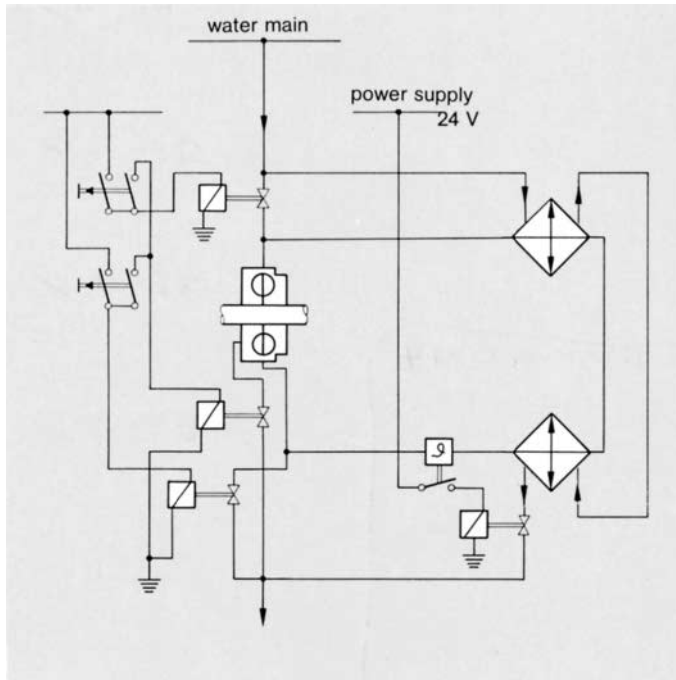


FIG. B-16 Heat-exchanger control. The supply is from the water main. Solenoid valves control the extent to which the closed circuit is filled. In order to reduce the water consumption, the heat energy is dissipated through a heat exchanger through which the flow is controlled in relation to the temperature. (Source: J.M. Voith GmbH.)

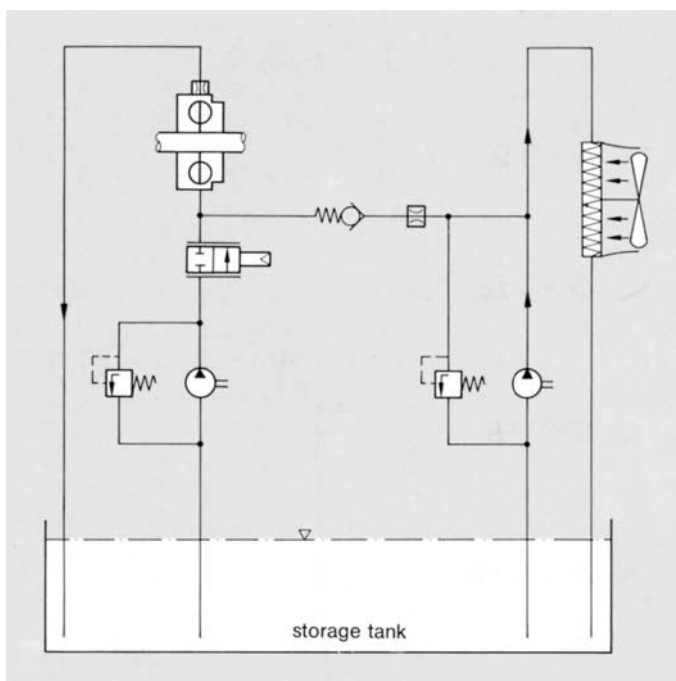


FIG. B-17 Pump control with separate cooling for the storage tank. The working fluid is drawn from the storage tank and fed to the hydrodynamic brake by means of a pump and by way of a pressure relief valve—to maintain the pressure at a constant level—and a solenoid valve—to turn the supply on and off. The heated working fluid flows back to the storage tank and is cooled in a separate circuit that also provides for the constant cooling and lubrication of the hydrodynamic brake. (Source: J.M. Voith GmbH.)

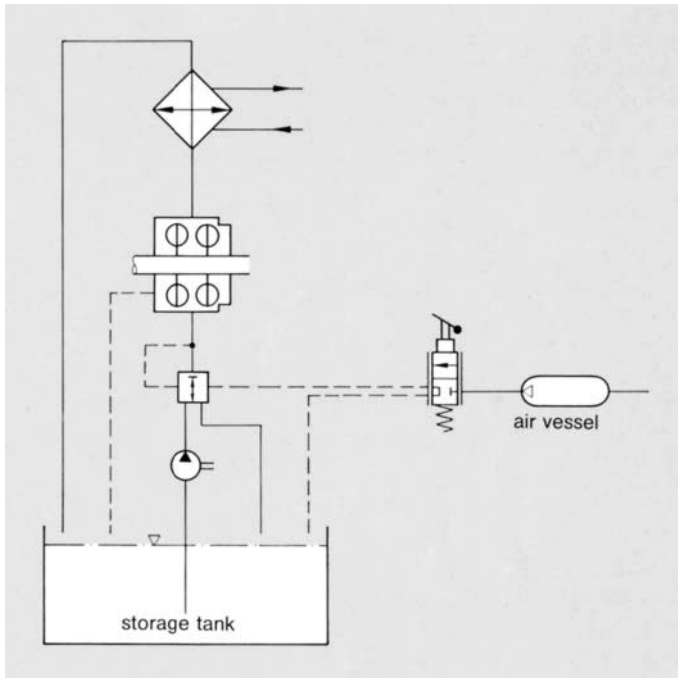


FIG. B-18 Pump control with integral cooling. The supply to the hydrodynamic brake is effected by a pump by way of a pneumatically controlled pressure-regulating valve. The heated working fluid is cooled in a heat exchanger in the return line. (Source: J.M. Voith GmbH.)

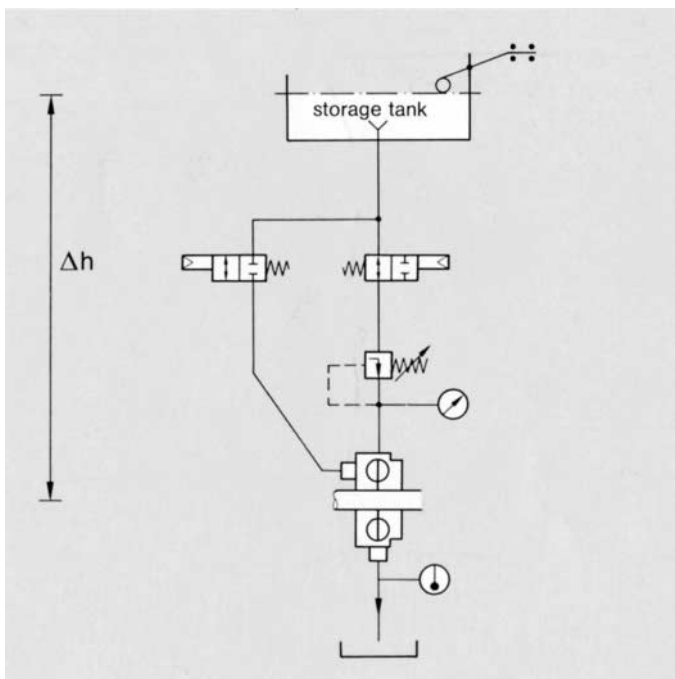


FIG. B-19 Through-flow control. Here the hydrodynamic brake is fed with water from a water main or from an elevated storage tank by way of a pressure-regulating valve and solenoid valve. The heated water is discharged to a drain. (Source: J.M. Voith GmbH.)

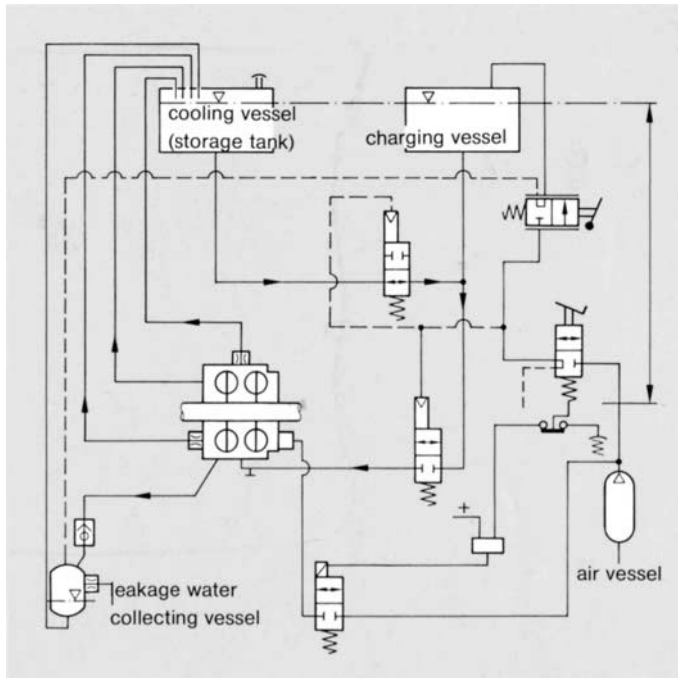


FIG. B-20 Air vessel control—with capacitative heat storage in the storage tank. The charging vessel is filled by gravity. On braking, the connection between the two vessels is interrupted. Compressed air is applied to the charging vessel and the working fluid is forced into the brake. The filling pressure is manually preselected. The heated working fluid flows back into the cooling vessel. Any leakage water flows into a separate collecting vessel and is then fed back to the cooling vessel by applying compressed air. (Source: J.M. Voith GmbH.)

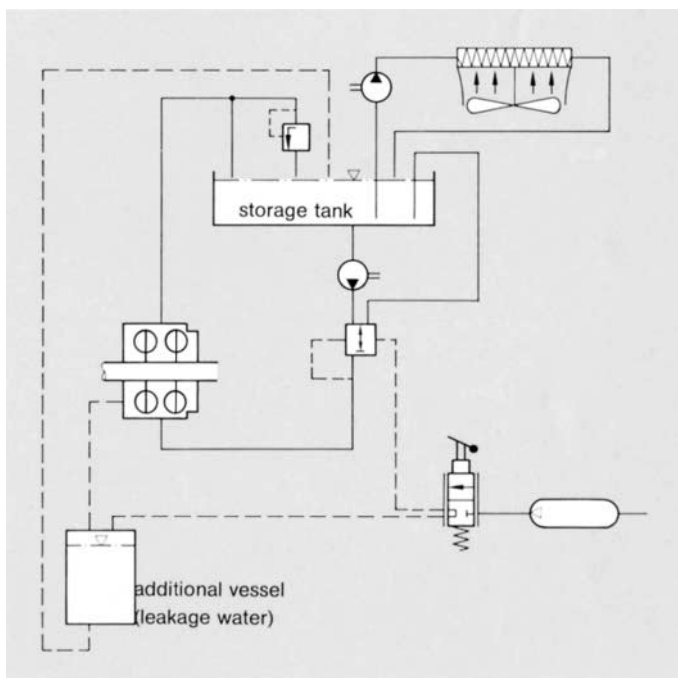


FIG. B-21 Pump control with separate cooling circuit. The hydrodynamic brake is supplied with fluid by a pump by way of a pneumatically controlled pressure-regulating valve. An additional vessel is provided to collect any leakage water, which is then forced back into the storage vessel by compressed air. The storage vessel is cooled by way of a separate cooling circuit. (Source: J.M. Voith GmbH.)

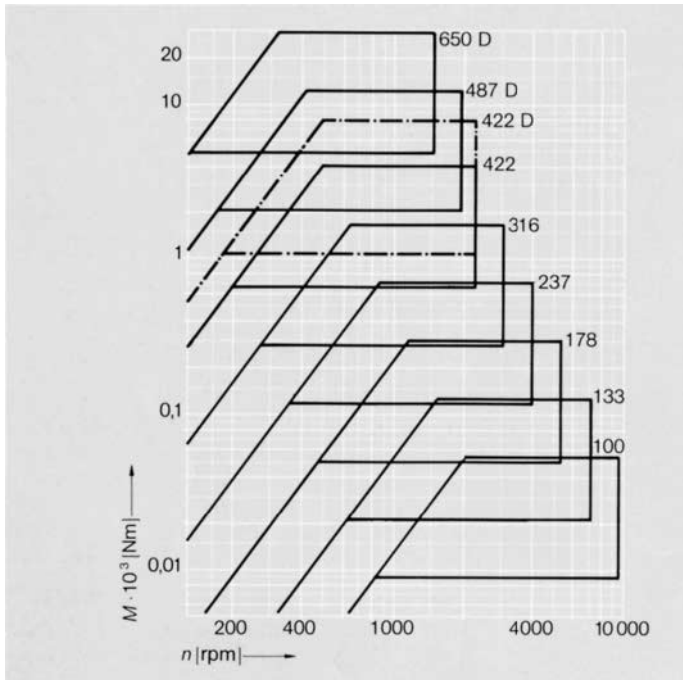


FIG. B-22 $M = f(n)$, **braking torque as a function of speed. (Source: J.M. Voith GmbH.)**

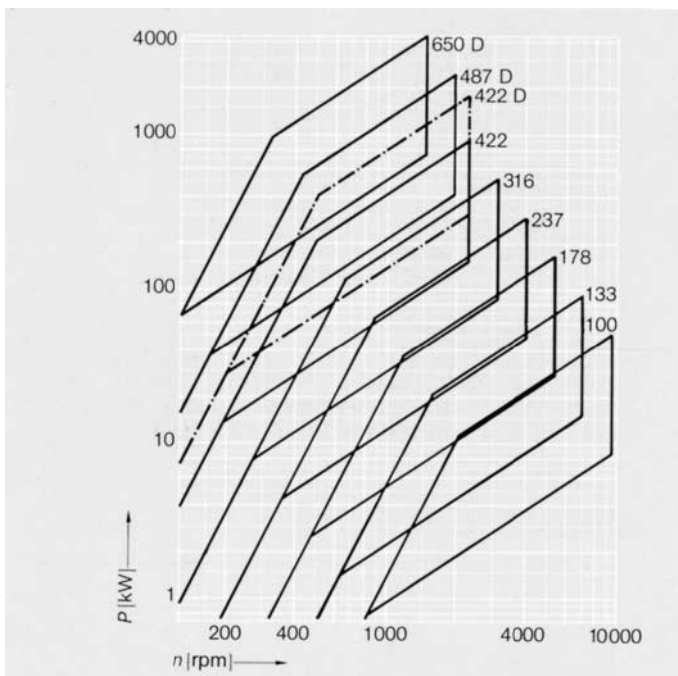


FIG. B-23 $P = f(n)$, **braking power as a function of speed. (Source: J.M. Voith GmbH.)**

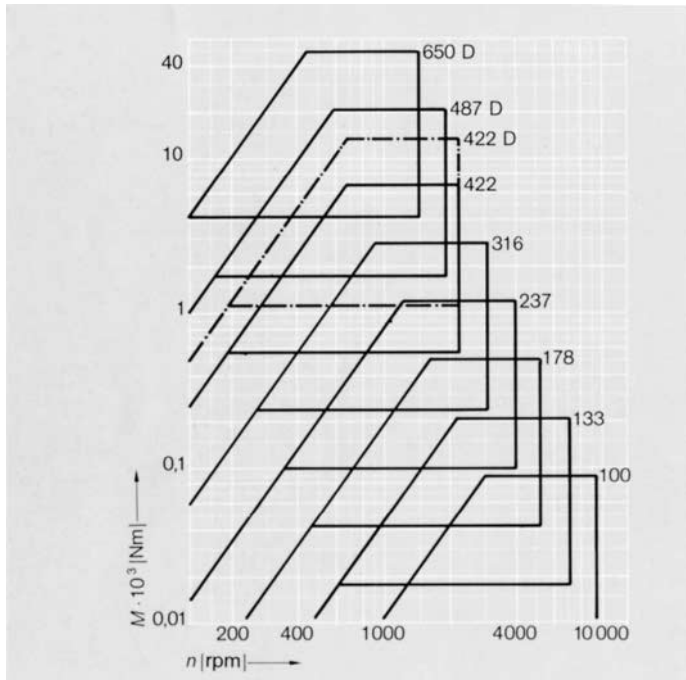


FIG. B-24 $M = f(n)$, **braking torque as a function of speed. (Source: J.M. Voith GmbH.)**

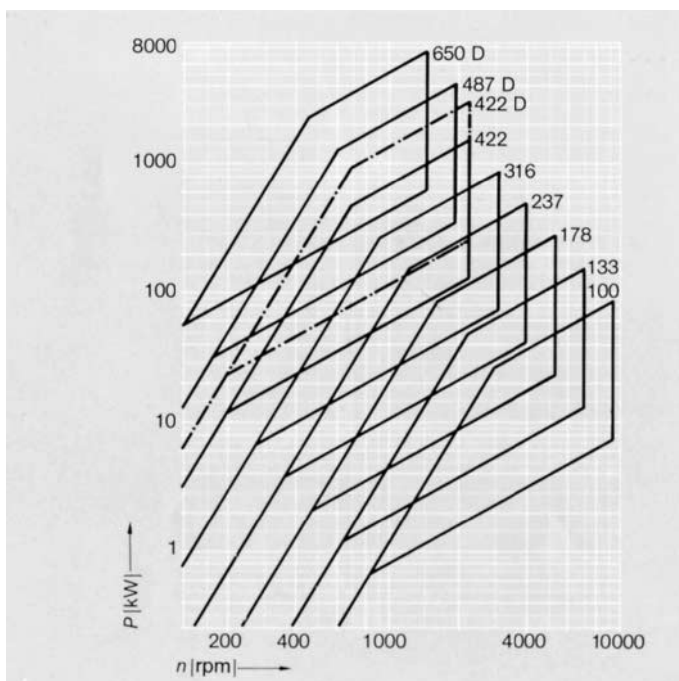


FIG. B-25 $P = f(n)$, **braking power as a function of speed. (Source: J.M. Voith GmbH.)**

Single-flow models

Double-flow models

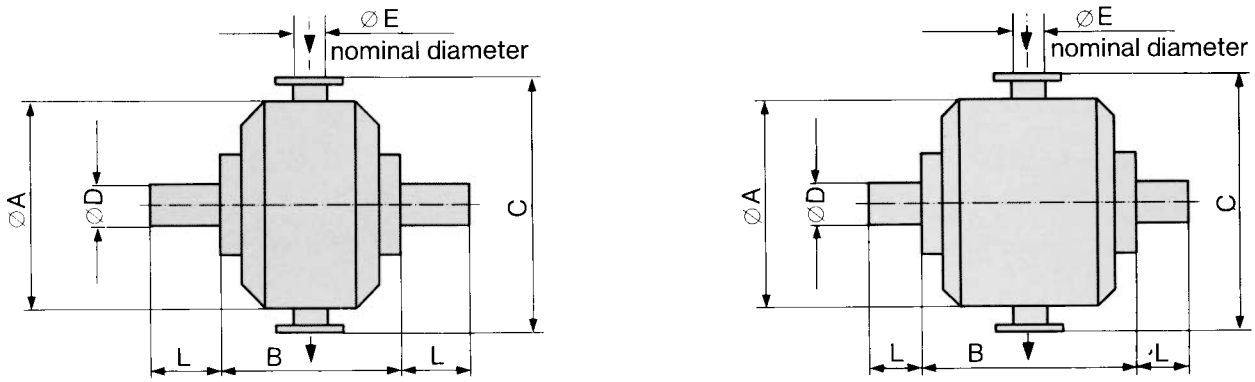


FIG. B-26 Reference dimensions of different brake models. (Source: J.M. Voith GmbH.)

Dimensions. See Fig. B-26 for a diagrammatic definition of reference dimensions.

Single-Flow Models

Brake Type	A (mm)	B	C	D	E	L	Weight (kg)
100	130	130	150	28	20	42	6
133	170	170	195	30	25	58	13
178	225	225	260	40	32	82	32
237	300	300	345	55	40	82	76
316	400	400	460	75	65	105	180
422	535	535	615	95	80	130	430

All dimensions and weights given represent maximum values. The dimensions "D" and "L" represent the dimensions necessary for the transmission of the indicated maximum torque.

Double-Flow Models

Brake Type	A (mm)	B	C	D	E	L	Weight (kg)
100 D	125	133	150	30	25	58	7
133 D	170	176	195	38	32	58	15
178 D	225	236	260	55	50	82	37
237 D	300	315	345	70	65	105	88
316 D	400	420	460	90	80	130	208
422 D	535	560	615	120	100	165	495
487 D	620	646	710	140	125	200	761
650 D	825	865	950	190	150	280	1809
826 D	1050	1100	1210	240	200	330	3712

All dimensions and weights given represent maximum values. The dimensions "D" and "L" represent the dimensions necessary for the transmission of the indicated maximum torque.

Technical data

Single-Flow Models

Brake Type	M_{BR} max. (N·m)	n Abreg. (min^{-1})	n max. (min^{-1})
Working medium: oil			
100	86	2,868	9,549
133	203	2,164	7,180
178	484	1,620	5,365
237	1,157	1,215	4,029
316	2,712	910	3,022
422	6,494	684	2,263
Working medium: water			
100	52	2,037	9,549
133	121	1,541	7,180
178	291	1,151	5,365
237	681	869	4,029
316	1,634	652	3,022
422	3,831	490	2,263

Double-Flow Models

Brake Type	M_{BR} max. (N·m)	n Abreg. (min^{-1})	n max. (min^{-1})
Working medium: oil			
100 D	172	2,868	9,549
133 D	406	2,164	7,180
178 D	968	1,620	5,365
237 D	2,314	1,215	4,029
316 D	5,424	910	3,022
422 D	13,032	684	2,263
487 D	20,000	598	1,961
650 D	47,863	444	1,469
826 D	97,067	349	1,156
Working medium: water			
100 D	104	2,037	9,549
133 D	242	1,541	7,180
178 D	582	1,151	5,365
237 D	1,362	869	4,029
316 D	3,268	652	3,022
422 D	7,662	490	2,263
487 D	12,069	423	1,961
650 D	28,292	316	1,469
826 D	64,050	249	1,156

Control characteristic curves—examples of application

Controls can be provided to give various characteristic curve forms. The characteristic curve form should be so selected that load moment line intersects the brake torque line at a defined point. (See Figs. B-27 and B-28.) Figure B-29 shows a brake assembly schematic. Figures B-30 through B-36 show applications of brake assemblies in various industries.

Brick-Lined Process Equipment

The thermal insulation value and other properties of the brick and protective membrane layer between the brick and the process fluid it contains can be determined from property values (such as corrosion resistivity) listed in catalogs supplied by the brick manufacturers or relevant engineering standards. *See* Some Commonly Used Specifications, Codes, Standards, and Texts.

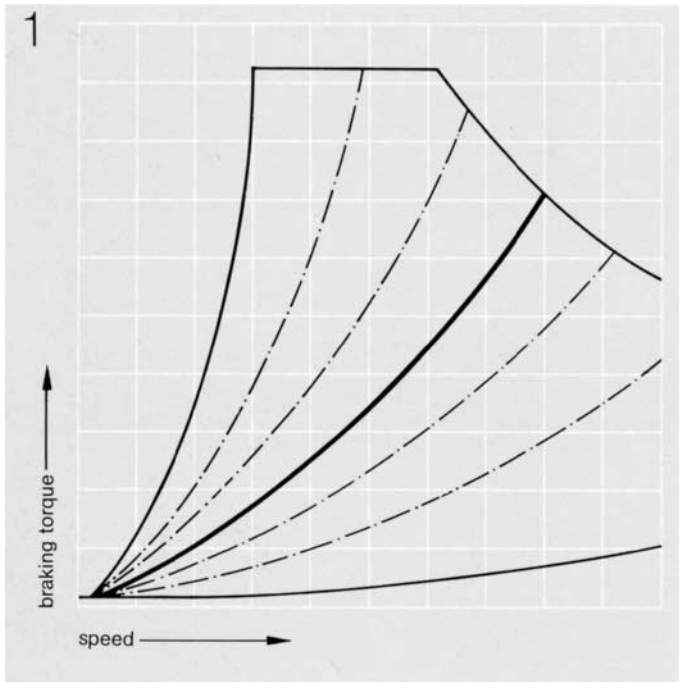


FIG. B-27 Control for a parabolic braking torque. (Source: J.M. Voith GmbH.)

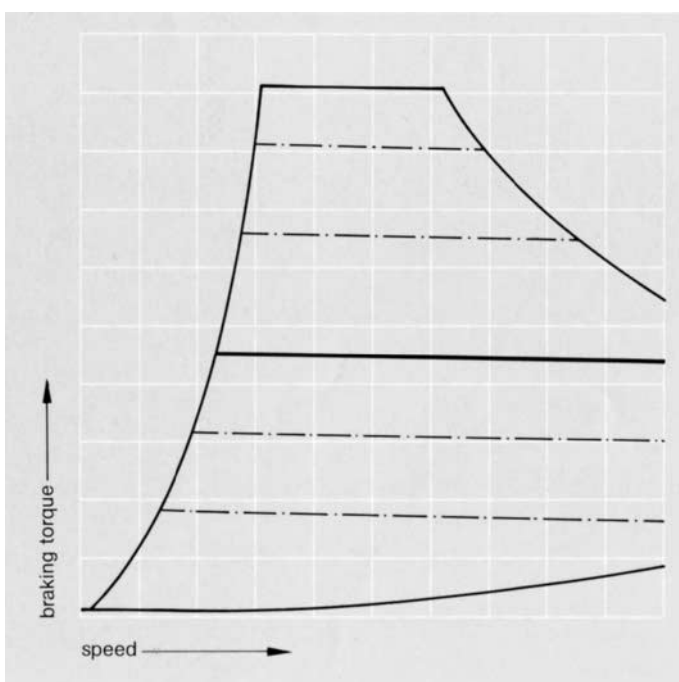


FIG. B-28 Control for a constant braking torque. (Source: J.M. Voith GmbH.)

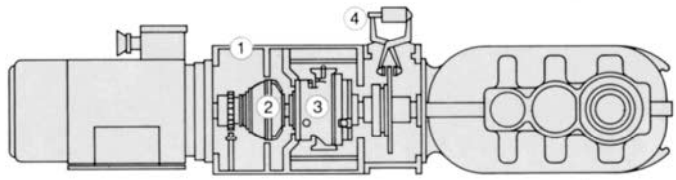


FIG. B-29 Schematic of a brake assembly. 1, connecting housing; 2, turbo coupling TV; 3, hydrodynamic brake VHBI; 4, mechanical brake. (Source: J.M. Voith GmbH.)



FIG. B-30 VHBI 422 D, fitted in the drive to the winch of a transportable drilling rig suitable for drilling to a depth of approximately 5000 meters, Texas, USA. (Source: J.M. Voith GmbH.)

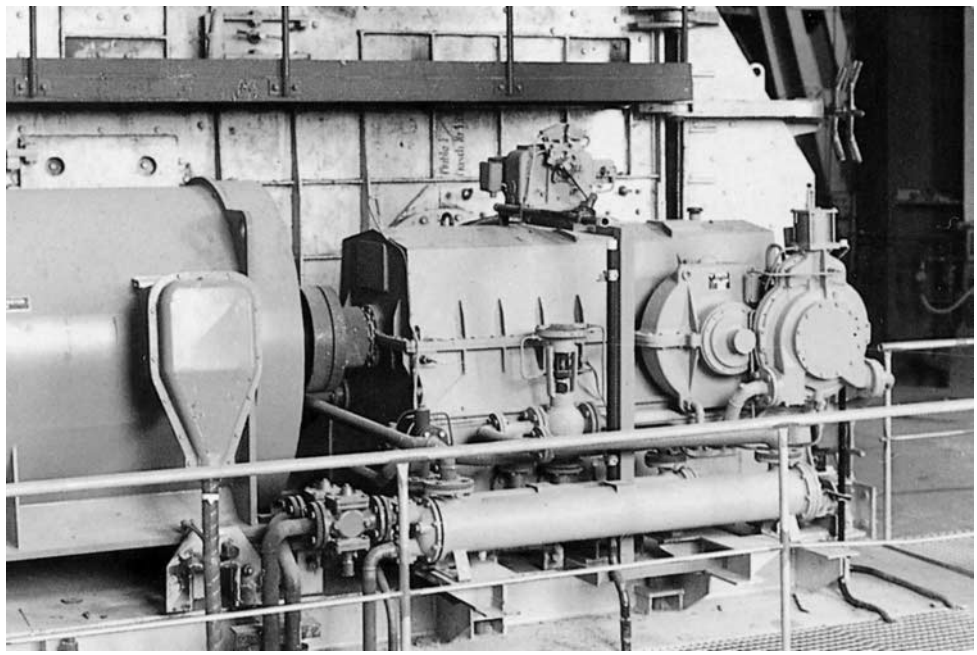


FIG. B-31 VHBI 316 with stepup gear, flange-mounted to a Voith variable-speed geared turbo coupling R 750 B in the drive to a coal crusher in a power station in West Germany. (Source: J.M. Voith GmbH.)

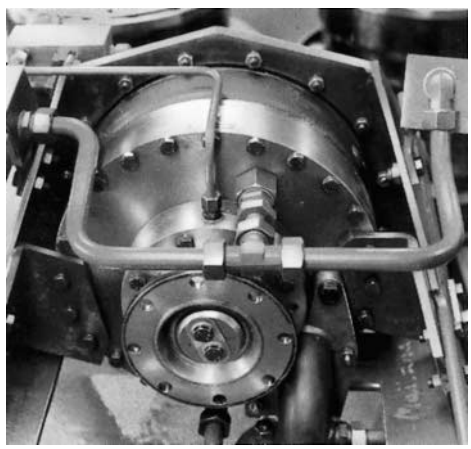


FIG. B-32 VHBI 316 working in connection with a Voith coupling 562 SVL as an element in a performance test stand for electric motors in Italy. (Source: J.M. Voith GmbH.)

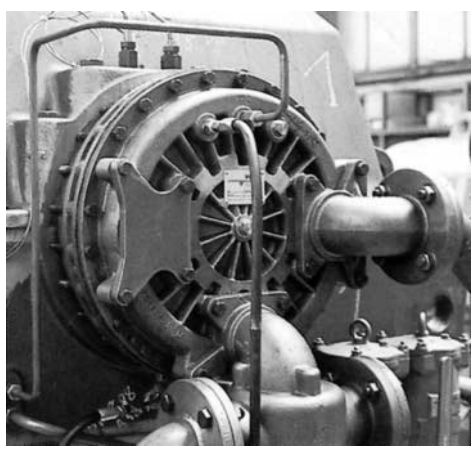


FIG. B-33 VHBI 422 flange-mounted to a Voith variable-speed geared turbo coupling R 866 B 5 in the drive to a coal crusher in a power station in Yugoslavia. (Source: J.M. Voith GmbH.)

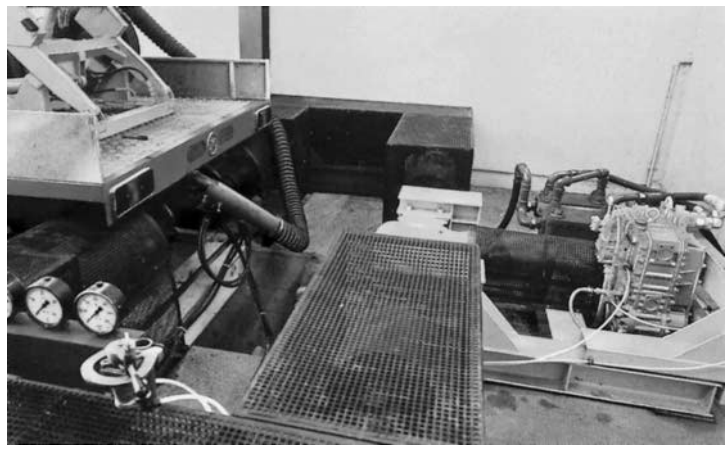


FIG. B-34 VHBK 130 fitted in a test stand for testing the hydrostatic drive system of ski slope maintenance machines in West Germany. (Source: J.M. Voith GmbH.)

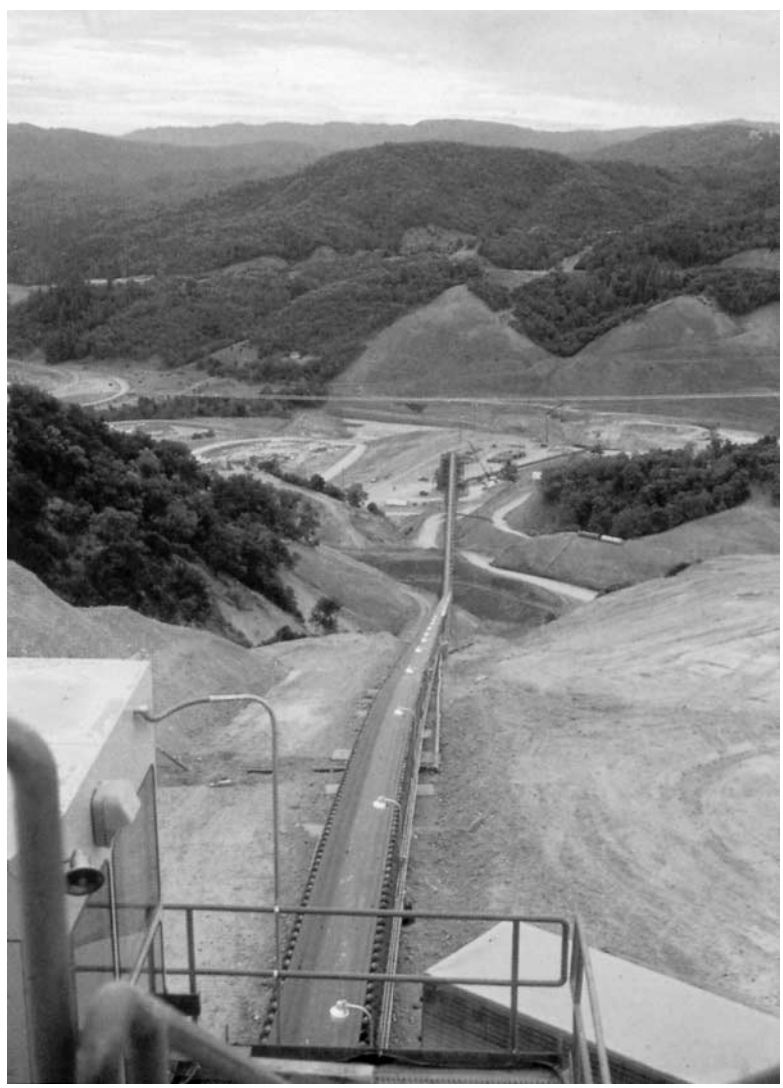


FIG. B-35 VHBI 422 with throughgoing shaft, fitted to a Voith variable-speed turbo coupling 650 SVNL in the drive to a 1-km-long downward sloping conveyor belt feeding material to an imponding dam in California, USA. (Source: J.M. Voith GmbH.)



FIG. B-36 VHBI 316 with stepup gear, flange-mounted to a reduction gear in one of the drive stations in a 10-km-long conveyor plant, made up of several conveyor belts, for the conveyance of bauxite in West Australia. (Source: J.M. Voith GmbH.)

Briquette Machines (Similar to Pelletizing and Balling Machines)

These types of machinery are custom built for different products and processes. Basically, work energy is used to make briquettes of varying size and shape. These machines are common in agricultural, domestic, and chemicals industries. *See* Some Commonly Used Specifications, Codes, Standards, and Texts.

C

CAD/CAM (*see also* Machining)

CAD/CAM is an acronym for computer-automated design and computer-automated machining. CAM is generally conducted in conjunction with computer numerical control (CNC) for metal-working processes such as blade-tip robotic welding, CNC tig welding, and CNC lathing, milling, and/or machining. All these numerical and computational methods contribute to a field that is now a science in its own right. *See* Metallurgy for information on blade-tip robotic welding, an example of a CNC CA (in this case welding) process.

CFD (Computational Fluid Dynamics)*

CAD is frequently used in conjunction with computational fluid dynamics (CFD) where air- or gasflow is involved. For illustrative purposes, some literature on CFD, as a component of CAD, follows.

CFD can be used for analyzing, for instance

- Fluid handling
- Measurement and controls
- Heat and mass transfer
- Filtration
- Distillation
- Mixing
- Separation
- Fluidization
- Sedimentation
- Reaction
- Polymerization
- Drying
- Forming
- Ventilation
- Emission control
- Incineration
- Combustion
- Materials processing

*Source: Fluent Inc., USA. Adapted with permission.

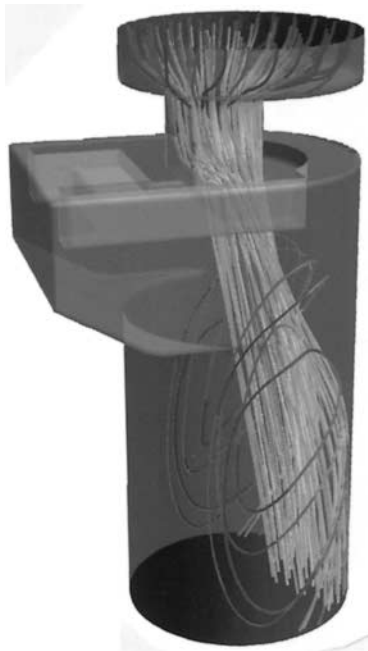


FIG. C-1 Streamlines depict the flow of regenerated catalyst through a slide valve, revealing the source of erosion problems. Courtesy of REMOSA. (Source: Fluent Inc.)



FIG. C-2 Instantaneous solids concentration in a riser, showing how a baffle plate can be used to distribute flow and produce uniform delivery of catalyst to the reactor. (Source: Fluent Inc.)

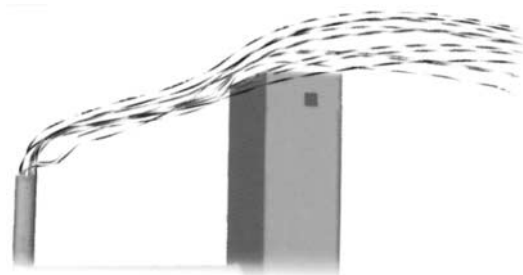


FIG. C-3 Prediction of exhaust plume trajectories can provide quantitative information about downstream impacts. (Source: Fluent Inc.)

Figures C-1 through C-11 are modeling representations of eight individual case studies.

The work done by CFD firms is frequently unique to a specific firm and any research alliances it may have formed with individual OEMs.

Some Generic CFD Applications

Fluid handling and flow distribution

Transport and storage of gases, liquids, or slurries represents a large capital and operating expense in process plants. CFD software helps to design for flow uniformity, balance flows in manifolds, minimize pressure drop, design storage tanks, and accurately size blowers, fans, and pumps. High-speed nozzles and spray systems can be analyzed in order to optimize performance.

Reactor modeling

CFD software helps you to quantify residence times, mixing rates, scaling effects, and overall chemical conversion in a wide range of reactor systems, including packed beds, fluidized beds, recirculating beds, plug flow or tube reactors, and stirred tank reactors. This provides the flexibility you need for description of reactions and the sophistication you need for prediction of gas-solid, gas-liquid, or liquid-solid multiphase systems.

Ventilation and safety

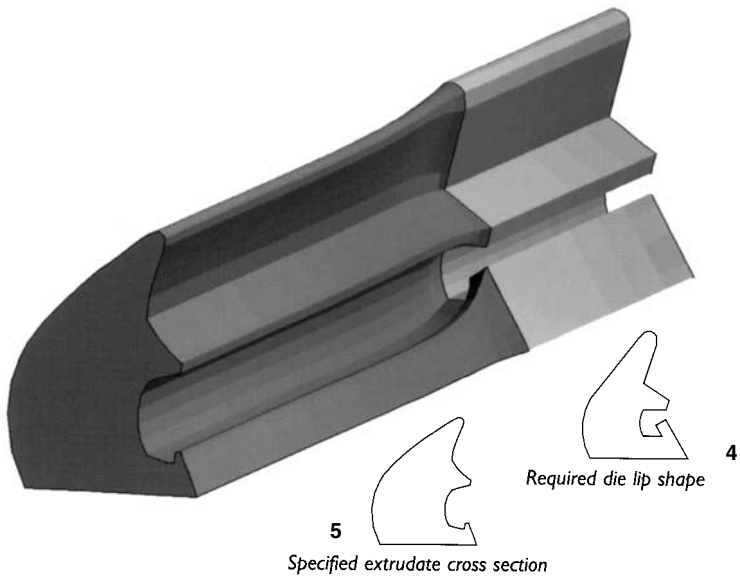
CFD software allows you to reliably and easily determine the trajectory of environmental releases and examine building ventilation system performance. You can quantify the exposure of personnel to specific contaminant levels and analyze the effectiveness of planned responses.

Mixing

Mixing in agitated vessels, static mixers, jet mixers, t-mixers, and other devices is important to the performance of most chemical and process plants. The capability for the analysis of stirred tank mixers is unsurpassed, with interactive automated model generation and mixing-specific data analysis tools.

Materials processing

Extrusion, mold filling, fiber forming, thermoforming, and coating processes involve complex fluid rheology and deforming free surfaces that not all CFD codes can



Prediction of extrudate shape. Inverse die design capability allows determination of the required die lip (FIG. C-4) for a specified extrudate shape (FIG. C-5). (Source: Fluent Inc.)

model. CFD software includes powerful tools that are customized to excel for these applications.

Separation and filtration

CFD can provide a complete range of tools for modeling of phase separation, solids settling, and particle dispersion and classification. Inertial separation using chevrons or cyclonic separators and filtration systems using filter media can also be modeled.

Combustion, incineration, emissions, and environmental control

CFD can provide state-of-the-art models for prediction of combustion and pollutant formation, including built-in NO_x prediction. Optimization of environmental control equipment, from incinerators to scrubbers, filters, and collectors, can help ensure compliance and reduce capital costs.

Some Examples of Specific CFD Case Studies

This material is proprietary to the information source company and therefore contains mention of trademarks specific to this designer.

Process industry modeling

The European Commission has funded the European office of this information source to work on two projects related to process industry applications.

OLMES is a project that looks at the application of CFD to the design of membrane separation devices used in the production of reformulated gasoline. To date, the primary method available for olefin reduction in FCC-derived gasoline is

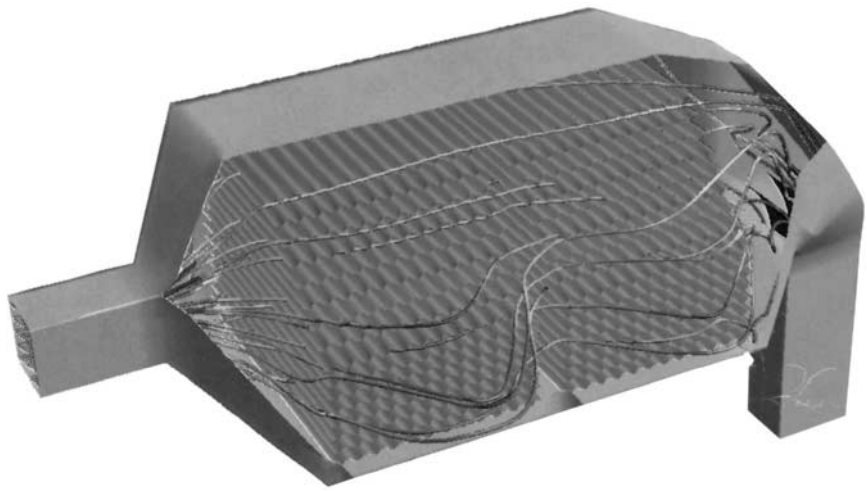


FIG. C-6 Flow streamlines in an electrostatic precipitator confirm the effectiveness of vanes for improving uniformity of flow through the precipitator plates. (Source: Fluent Inc.)

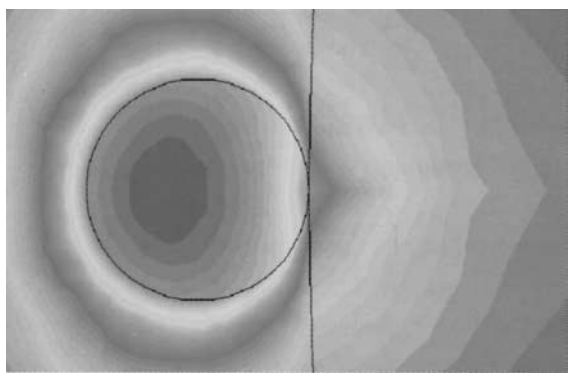


FIG. C-7 Temperature prediction in the vicinity of two catalyst-impregnated particles, studied as part of the CATAPOL project on the application of CFD to polymerization. (Source: Fluent Inc.)

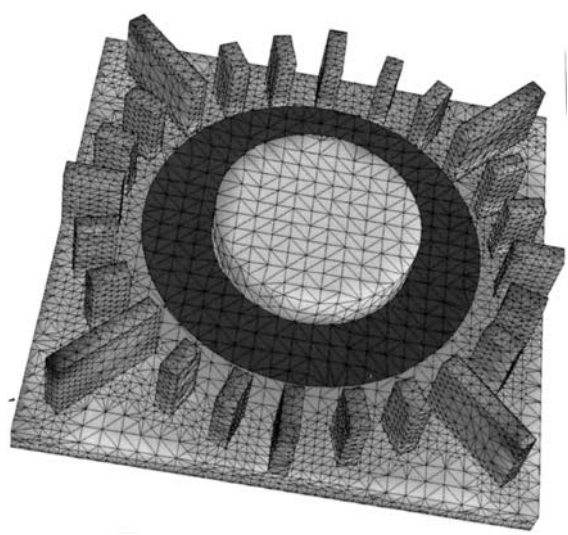


FIG. C-8 Tetrahedral mesh used in IcePak 2.2 to model a complex fan/heat sink with radial fins. (Source: Fluent Inc.)

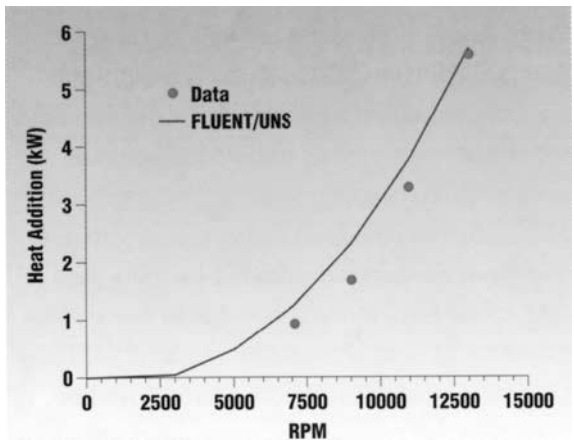
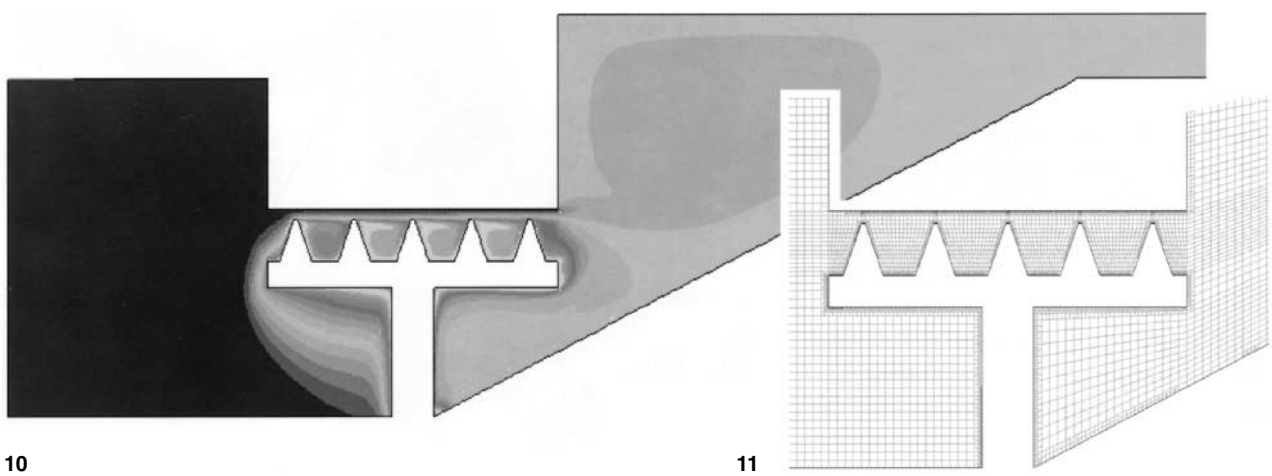


FIG. C-9 The FLUENT/UNS prediction yielded good agreement with the measured heat addition due to windage heating. (Source: Fluent Inc.)



10 11

FIG. C-10 Temperature prediction showing the effect of windage heating. (Source: Fluent Inc.)

FIG. C-11 Mesh adaption with embedding was used to ensure grid independence of the predictions. (Source: Fluent Inc.)

the hydro-treating process. However, hydro-treatment is a very energy-intensive process and significantly reduces octane quality. Membrane separation is now becoming an attractive alternative, but many practical design problems remain. The OLMES project aims to build simulation software that will aid the design process by modeling the fluid flow and mass transfer in the many fine passageways that make up a membrane separator.

CATAPOL is a project that applies CFD to the modeling of heterogeneously catalyzed gas-phase polymerization, involving the injection of catalyst particles into a fluidized bed where they react with monomer gas to grow polymer particles. Using CFD modeling of individual particle behavior and fluidized bed hydrodynamics, the project addresses potential problems such as reactor stability and thermal runaway.

New core technology for IcePak

IcePak, a specialty CFD product for electronics cooling applications, continues to evolve at a fast pace. Jointly developed by this information source and ICEM CFD, new releases of IcePak are delivering on the commitment to make this information source's latest core CFD technology available for electronics thermal management.

The first adaptation of this product, "Version 2.1," released in October 1997, delivered speed improvements by incorporating the solver engine from FLUENT/UNS (an average of 7.5 times faster during regression testing).

The next adaptation of this product, "Version 2.2," was intended to deliver more flexible model building capability, with tetrahedral as well as hexahedral meshes supported. Tetrahedral meshes can handle extremely complex geometries. The combination of automated hex and tet meshing gives IcePak users better strategies when confronted with difficult modeling.

Predicting windage heating in labyrinth seals

Labyrinth seals are commonly used in rotating equipment to restrict cooling flow between rotating and stationary components. One of the issues in the design of such seals is the accurate prediction of the temperature rise of the cooling flow due to windage heating effects. Accurate prediction of this heating allows designers to maintain the structural integrity of the engine with the minimum amount of cooling flow, thereby maximizing the efficiency of the engine.

In order to validate the accuracy of this information source's product ("FLUENT/UNS") for this kind of flow prediction, a straight five-fin labyrinth knife seal with a nominal clearance of 1.11 mm between the labyrinth seal and the shroud was analyzed. The computational model was axisymmetric, with specification of the pressure ratio (1.5) across the seal as imposed boundary conditions. The working fluid was assumed to be air, with density computed via the ideal gas law and fluid properties (viscosity, thermal conductivity, and heat capacity) expressed as a function of temperature. Turbulence was modeled using the RNG k-ε model. The CFD model was run at several different rotor speeds and the windage heating was computed and compared with the experimental data.

Grid independence using adaption. Grid-independent solutions were obtained at each operating speed by using the solution-based mesh adaption capability in FLUENT/UNS. The mesh was adapted based on predicted y^+ values and on gradients of total temperature. The initial mesh for each simulation started out with approximately 9400 quadrilateral cells and after adaption the final cell count was approximately 12,500 cells. A typical mesh after the solution-adaptive refinement is shown below.

Carbon; Carbon-Graphite Mix Products

When mixed with graphite, carbon has lubricant-, strength-, and temperature-resistance properties that are useful in many applications. The properties of the mix depends on the variation in the mix, as well as the bonders and adhesives used. These properties will therefore vary between companies that make these products. For illustrative purposes, one company's product line, with applications, is outlined here.*

*Source: Advance Carbon Products, USA. Adapted with permission.

TABLE C-1 AD-CARB Materials

Grade	Composition	Density, g/cm^3	Shore Hardness	Compressive Strength, psi	Flexual Strength, psi	C.T.E., in/in/°F × 10^{-6}	Modulus of Elasticity, psi × 10^{-6}	Temperature Limit in Air, °F
AC-1	CG	1.59	40	8,000	3,600	2.5	1.7	700
AC-1F	CG-R	1.75	55	11,000	5,000	3.0	1.9	400
AC-2	CG	1.71	60	11,000	6,700	2.5	2.0	700
AC-2F	CG-R	1.80	70	15,000	8,000	3.0	2.2	400
AC-3	CG	1.66	50	12,000	4,600	2.4	1.7	700
AC-4	CG-R	1.75	65	20,000	6,000	2.9	1.9	400
AC-8	G	1.76	40	9,000	5,000	1.6	2.0	750
AC-9	G	1.64	35	4,000	1,900	2.2	1.6	750
AC-10	G	1.72	35	5,600	3,500	2.1	1.8	750
AC-20	G	1.78	45	9,300	4,300	2.3	1.6	750
AC-22	G	1.71	35	7,200	3,500	2.2	1.5	750
AC-23	G-P	1.71	35	7,200	3,500	2.2	1.5	750
AC-25	G	1.72	40	7,000	4,000	1.6	1.6	750
AC-26	G	1.70	45	8,000	4,500	2.6	1.5	750
AC-27	G	1.90	85	18,000	10,000	3.5	1.9	750
AC-30	G	1.81	45	9,000	4,800	1.4	1.5	750
AC-40	CG-B	3.00	40	13,000	6,000	2.3	2.9	400
AC-50	CG-BR	2.85	45	17,000	7,500	2.6	2.9	500
AC-52	CG-BR	2.55	60	25,000	8,500	2.6	3.1	500
AC-60	CG-CU	2.85	50	16,000	7,500	3.1	2.8	750
AC-62	CG-CU	2.50	60	24,000	8,500	3.0	2.8	750
AC-80	CG-AG	3.40	45	16,000	7,000	3.1	2.7	750
AC-82	CG-AG	2.60	65	24,000	8,000	3.0	2.9	750
AC-101	CG-NI	2.40	60	23,000	7,500	2.8	2.8	750
AC-106	CG-NI	2.85	50	15,000	6,500	2.7	3.1	750
AC-236	CG-R*	1.85	85	26,000	9,500	2.9	3.2	500
AC-240	CG	1.70	85	30,000	9,000	2.1	2.5	700
AC-243	G	1.85	90	33,000	12,000	2.8	3.0	850
AC-245	CG-AG	2.45	75	36,000	12,000	2.8	4.0	700
AC-246	CG-R	1.80	95	40,000	12,500	3.2	3.3	500
AC-250	CG	1.75	65	20,000	7,500	2.0	2.3	700
AC-253	CG-AT	2.10	75	27,000	10,000	2.3	3.9	700
AC-266	CG-BR	2.40	80	40,000	13,000	3.7	4.1	500
AC-270	C-B	2.40	65	26,000	10,000	3.7	3.0	400
AC-540	G	1.77	50	13,000	5,500	1.8	1.3	750
AC-562	G-P	1.88	70	21,000	14,000	4.3	1.6	850

Code: CG = Carbon-Graphite; G = Graphite; P = Purified; R = Resin; B = Babbitt; BR = Bronze; CU = Copper; AG = Silver; AT = Antimony; NI = Nickel-Chrome; * = Pressure Tight after Machining

Example of a Carbon Mix Material

AD-CARB is a self-lubricating, low-friction carbon material that has high mechanical strength and can withstand temperatures up to 750°F without oxidizing.

Through the impregnation of metals and/or resins, this material can solve many lubrication problems. See Table C-1.

Product properties

1. Self-lubricating. The product acts as an effective dry lubricant, forming a thin film on mating parts, which allows the material to properly function without additional lubrication. The product may be used with light lubrication to enhance its load-carrying capabilities. The material is nongalling.

2. Oxidization resistant. The product maintains its properties up to 750°F without oxidizing. The material may be used up to 1500°F in an inert atmosphere, depending upon which grade is used. Certain materials may be treated to operate successfully up to 1100°F in an oxidizing atmosphere.

3. High mechanical strength. The product possesses high mechanical strength and hardness to create a long-wearing material. The strength of the material actually increases with an increase in temperature.

4. Chemical resistant. The product is chemically stable and does not react with most chemicals: acids, alkalis, salts, and organic solvents.

5. Low coefficient of friction. The product has a coefficient of friction of approximately 0.05 to 0.20, depending upon the lubrication and surface finish of the mating surface.

6. Low coefficient of thermal expansion. The product has a coefficient of thermal expansion of between 1.5 and 4.0×10^{-6}/°F, allowing for precisely controlled running clearances from room temperature up to maximum operating temperature.

7. Material classification. The product is a mixture of carbon and graphite that may be impregnated with various metals and resins to enhance the wear properties and improve the life of the materials.

Design factors

1. Environment. The carbon material and its impregnation must be checked to ensure that they are compatible with the environment.

2. Temperature. Though carbon does increase in strength with an increase in temperature, the maximum operating temperature of the material must not be exceeded. When the maximum temperature is exceeded, carbon becomes soft on the exposed surfaces from oxidization and will wear quite rapidly.

3. Mating materials. The product may be run against most materials with better results when run against the harder surfaces. Ceramics and chrome-plated surfaces are excellent mating materials. A minimum Rockwell C of 55 is recommended. Longer life will also be obtained if the mating material is ground and polished to a 16 microinch finish or better.

4. Load. The bearing load, in pounds per square inch, is calculated by figuring the total bearing load divided by projected area of the bearing (length × diameter). The maximum load that a bearing can withstand is related to the velocity (feet/minute) by the following formula: PV = 15,000 (dry).

If the bearing is thoroughly lubricated, the PV may be increased up to 150,000 or more, depending upon the circumstances. See Fig. C-12.

5. Interference fit. Since carbon bearings are usually installed in metal housings and run against metal shafts, the designer must be aware of the difference in the coefficient of thermal expansion of the materials and take this into consideration when designing the size of the bushing. At maximum operating temperature, the bushing must have at least a 0.0015 in interference fit per inch of diameter.

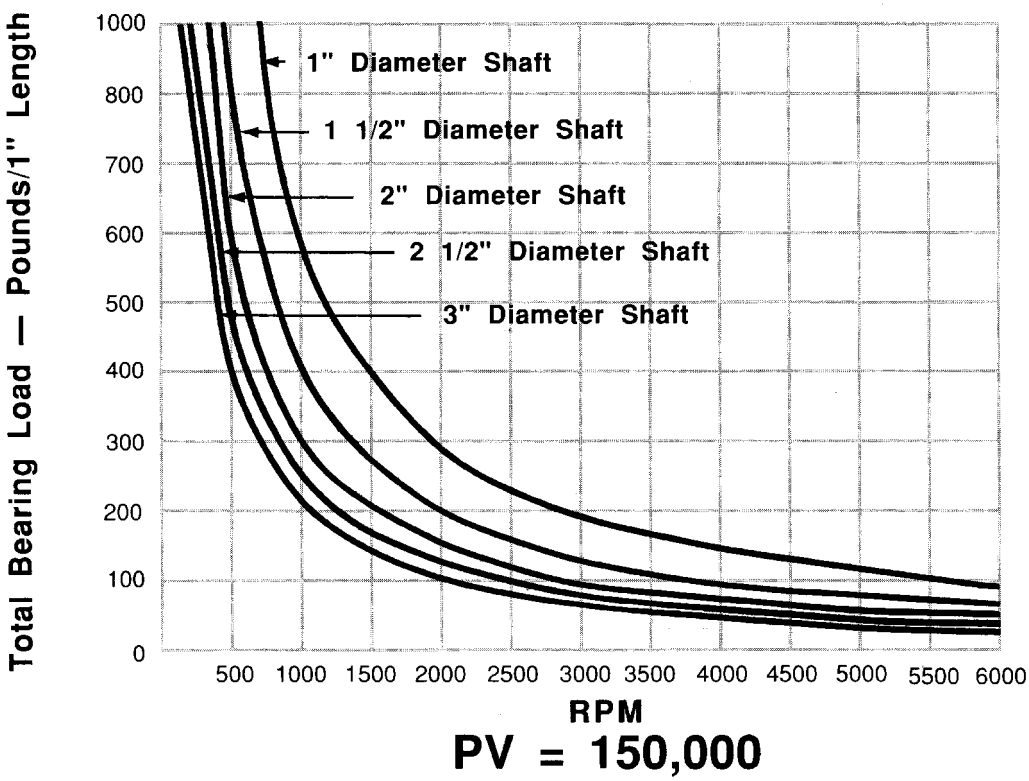


FIG. C-12 Maximum allowable load for thoroughly lubricated bearings. Reduce load to 10 percent of graph value for dry running. (Source: Advance Carbon Products.)

6. Close-in. When a carbon bearing is either pressed into a housing or heat-shrunk into a housing, the inside diameter will shrink in relation to the amount of interference on the outside diameter. This shrinkage must be taken into account when designing the inside diameter. For heavy-wall bearings (approximately ½ in), the inside diameter will shrink approximately 40 percent of the interference fit on the outside diameter; for thin-wall bearings (approximately ⅛ in), the inside diameter will shrink approximately 90 percent of the interference fit on the outside diameter.

7. Running clearance. The running clearing will vary depending upon the application requirements; however, a general guideline is to have 0.002 in/in of shaft diameter with a minimum of 0.003 in. This minimum clearance is to allow foreign particles to escape through the bushing, with minimal damage. These clearances are measured after the bearing is installed in the housing to take into account the inside diameter close-in. With high surface speeds the clearance may be reduced, and with slow surface speeds the clearance may be increased.

8. Bearing length. The bearing length should be one to three times the shaft diameter, with 1½ times the preferred length.

9. Preferred design. The preferred design is for the bearing to be a straight sleeve type without shoulders and grooves. Shoulders create stress points and can be a cause of failure. Grooves can be used quite successfully if they are installed with a radius tool bit and do not cut so deep into the wall as to weaken the bearing. The

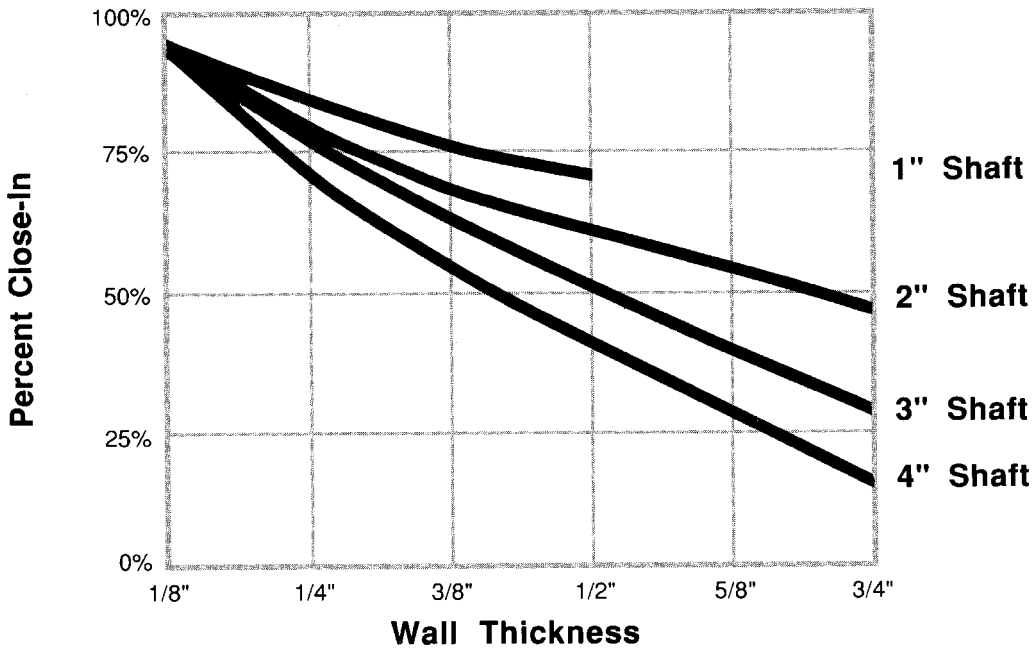


FIG. C-13 Close-in of inside diameters (approximation due to press or shrink fit). (Source: Advance Carbon Products.)

minimum wall thickness should be $\frac{1}{8}$ in, with $\frac{3}{16}$ in preferred. The recommended wall thickness is $\frac{1}{8}$ in + 10 percent of the bearing inside diameter. See Fig. C-13.

10. Tolerances. The product bearings can be machined to very close tolerances. The recommended tolerances are listed below, though closer tolerances can be machined at additional cost.

OD: ±0.001 in
ID: ±0.001 in
Length: ±0.010 in
Concentricity: 0.002 in TIR

Application example

Lube oil injection pump:

235°F operating temperature
316 stainless steel housing
316 stainless steel shaft, hard faced 0.020 deep, ground and polished
1770 rpm
3.092/3.090 housing diameter
2.4375 shaft diameter (maximum)

A. Material recommendation is AC-52, good up to 500°F.
B. Calculate expansion, close-in, and running clearance:

1. Housing: 316SS coefficient of thermal expansion is 8.3×10^{-6} in/in °F; @ 235°F, stainless steel housing expansion =
(235–68°F) (3.092 dia.) (8.3×10^{-6} in/in °F) = 0.0043 in housing expansion

2. Bearing: AC-50 coefficient of thermal expansion is 2.6×10^{-6} in/in °F;
   @ 235°F, carbon OD expansion =
   (235–68°F) (3.092 dia.) (2.6×10^{-6} in/in °F) = 0.0010 in bearing expansion
3. Minimum interference fit =
   0.0015 in/in of diameter
   ×3.092 in OD
   0.0046 in min. OD Interference
4. Calculate OD:

	+0.0043	Housing expansion
	–0.0010	Bearing expansion
=	+0.0033	Expansion difference
add	+0.0046	Minimum interference fit
=	+0.0079	Total addition to OD
add	+3.0920	Housing ID
=	+3.0999	Bushing OD, minimum

   Since the housing has a 0.002 in tolerance, the tolerance for the bearing should be reduced to 0.001 in, so as to prevent a buildup of excessive tolerances. Final bushing OD = 3.100, +0.001, –0.000.
5. Close-in of ID at room temperature, due to press fit:
   Calculate minimum and maximum interference fits:

	Minimum:	
	3.100	Minimum bearing diameter
	–3.092	Maximum housing diameter
=	0.008	Minimum interference
	×80%	Close-in percentage
=	0.0064	Minimum close-in
	Maximum:	
	3.101	Maximum bearing diameter
	–3.090	Minimum housing diameter
=	0.011	Maximum interference
	×80%	Close-in percentage
=	0.0088	Maximum close-in

   0.0088/0.0064 = 0.0076 average close-in
6. Running clearance:
   2.4375 in shaft diameter
   ×0.002 in clearance/in of shaft diameter
   = 0.0049 minimum running clearance
7. Calculate bearing ID:
   Bearing close-in = +0.0076 in
   Running clearance = +0.0049 in
   High-limit shaft diameter = +2.4375 in
   Inside diameter = +2.450 in (tolerance: ±0.001)
8. Bearing length:
   Shaft diameter 2.4375 in
   ×1.5 ratio
   3.656 in preferred length

Seals

Mechanical seals are custom machined to a specification. *Note:* Material specifications stated here are typical values, and will vary with the size of the material.

Products that can be made with carbon-based materials:

Brushes: motor/generator

- Carbon
- Electrographitic
- Metal-graphite
- Graphite

Brush holders

Brush seater and cleaner

Brush tension scales

Carbon specialties

- Seals
- Steam turbine seals
- Gas turbine seals
- Hydraulic turbine seals
- Labyrinth seals
- Spherical seals
- Compressor seals
- Pump and compressor blades
- Bearings
- Thrust bearings
- Sliding guides
- Porous carbon filters

Commutator dressing and finishing stones

Electrical contacts

- Graphite
- Silver-graphite
- Copper-graphite

Metal impregnated carbon—graphite

Carbon Dioxide (CO_2); CO_2 Disposal

Carbon dioxide, an inert gas, is a byproduct of combustion. Large volumes of CO_2 result from combustion of fossil fuels. Industrial users of fossil fuels include gas and steam turbines. Industrial activity has contributed increasing volumes of CO_2 to the atmosphere. CO_2 has been found to be a greenhouse gas. Greenhouse gases contribute to the phenomenon of global warming. There have been changes in atmospheric CO_2 content that have "corrected themselves" in planetary history, but those changes occurred over millions of years versus the current trend established over just a few centuries. Technologies are now being developed to remove CO_2 from atmospheric solution to lessen the amount of CO_2 released into the atmosphere.

Due to chemical composition (number of atoms of carbon in a molecule of fuel), some fuels produce less CO_2 on a unit-weight basis than others that may be more commonly accepted on the market. This is one of the best ways of mitigating emissions of this greenhouse gas.

Research project activity in the field of CO_2 mitigation includes liquefication of CO_2 experiments (liquid CO_2 can be used in dry cleaning). There may eventually

also be an industrial grade process to solidify CO_2 by "turning" it into limestone or calcium carbonate. What follows is a description of an industrial process developed in Sweden, where CO_2 resulting from fossil fuel combustion is reinjected into the ground.

Technology and Cost Options for Capture and Disposal of Carbon Dioxide from Gas Turbines: A System Study for Swedish Conditions*

The current massive dependency on fossil fuels—90 percent of the world population's commercial production and consumption of energy—together with predictions of a considerable increase in the total world energy consumption during the coming decades, implies that the emissions of carbon dioxide from human activities will rise significantly over that period. Carbon dioxide (CO_2) is the largest anthropogenic contributor to the greenhouse effect. There is a broad consensus among scientists that the current and increased CO_2 emissions will increase the global mean temperature and affect local climates significantly, with numerous and far-reaching economic and environmental consequences.

Among several options for limiting future CO_2 emissions, capture and disposal of CO_2 from combustion gases has been studied within the IEA Greenhouse Gas Implementing Agreement. The process components of CO_2 capture have been demonstrated, and a complete demonstration plant (200 ton CO_2 per day recovered from boiler flue gases) is in operation at Shady Point in the United States (formerly ABB Lummus Crest; as of 2000, ABB is part of the Alstom Corporation). Disposal of CO_2 into sandstone aquifers is now under demonstration on a commercial scale. Since 1996, Statoil injects 1 million ton CO_2 per year into the Utsira sandstone formation at the Sleipner natural gas field.

Financed by NUTEK (The Swedish National Board for Industrial and Technical Development), a system study has been performed with the objective to assess how recent knowledge on the technical and economic options for the capture and the disposal of CO_2 from combustion gases could be implemented into the Swedish energy system.

Aquifers suitable for disposal of carbon dioxide

Surveys of earlier geologic investigations have indicated that geologic formations—aquifers—that should be suitable for CO_2 disposal exist in the south of Sweden–Denmark (South West Skåne and the eastern part of Zealand) and in the Baltic Sea between Gotland and Lithuania. The aquifer in Skåne-Denmark has the most favorable location with an estimated storage capacity of up to 10 Gton CO_2 of which the part in Skåne is estimated to have a storage capacity of up to 3.5 Gton. This could be compared to the yearly Swedish CO_2 emissions, approximately 60 Mton in 1995.

Capture of carbon dioxide from gas turbine–based power plants

Large-scale electric power production (500 MW power) with CO_2 capture has been studied for natural gas combined cycle (NGCC) and coal-based integrated gasification combined cycle (IGCC) within the IEA Greenhouse Gas R&D Programme. Based on these studies, we have studied the possibilities of recovering

* Source: Vattenfall Utveckling AB, Sweden; also, this section is adapted from extracts from ASME paper 98-GT-443.

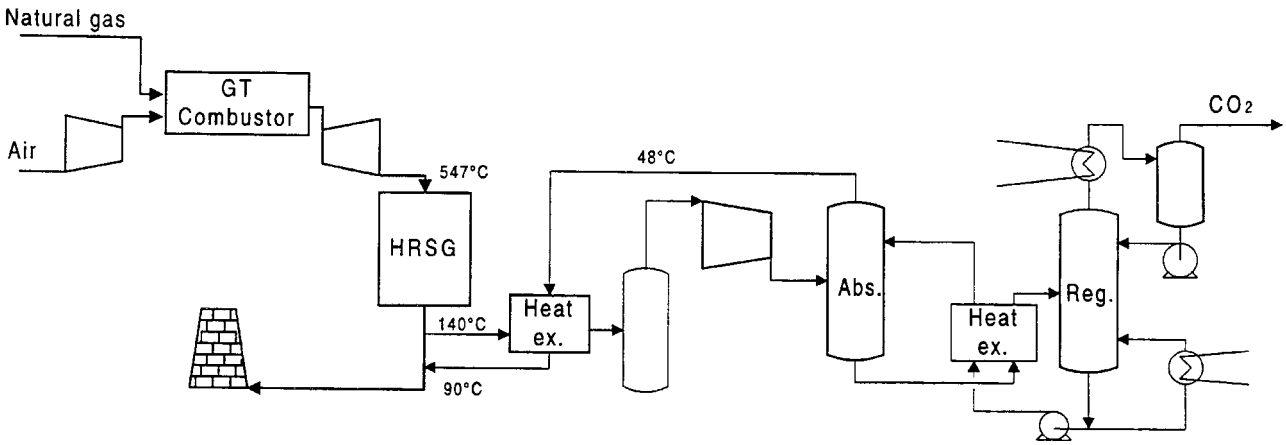


FIG. C-14 Process scheme for an NGCC (natural gas combined cycle) power plant with CO_2 capture. (Source: Vattenfall Utveckling AB.)

low-temperature heat from such processes as district heating. As a reference to the IGCC, performance and costs for pulverized coal combustion (PF, pulverized fuel) with and without CO_2 capture have also been estimated.

The process configurations, CO conversion rates, CO_2 removal efficiencies and other process parameters are the same as in IEA studies. Almost all process steps are based on proven technologies, and the process parameters have been chosen based on typically feasible designs and performances.

The NGCC plant consists of a single train of an advanced gas turbine. Like in some IEA studies, the calculations have been based on a Siemens 94.3A (turbine inlet temperature 1300°C and pressure ratio of 15–16) with a triple pressure reheat steam cycle (106 bar/30 bar/4.5 bar) (Fig. C-14). Due to the low partial pressure in the gas turbine exhaust gas, a CO_2 removal process based on chemical absorption, using a solvent such as MEA (MonoEthanol Amine), will be required. The assumed CO_2 removal efficiency is 85 percent. Regeneration of the solvent is performed by reboiling and stripping. Low-pressure steam for the regeneration is extracted from the steam cycle.

In the IGCC plant, coal is gasified at a high pressure (about 50 bar) and temperature (about 1400°C) with oxygen and steam in an entrained flow gasifier (Fig. C-15). In an IGCC plant without CO_2 capture, the fuel gas would be cooled and contaminants, such as dust and hydrogen sulfide, would be removed before burning the gas in the gas turbine combustor.

Due to its higher pressure and lower gas flow, it is advantageous to capture CO_2 in the fuel gas upstream of the gas turbine instead of from the exhaust gases. The fuel gas contains about 40 vol% CO, 28 vol% H_2, 18% H_2O, and 10 vol% CO_2. Since the CO (carbon monoxide) in the fuel gas would be emitted as CO_2 in the gas turbine exhaust gas, it must be converted to CO_2 prior to the CO_2 removal. This is achieved with steam according to the shift reaction, $CO + H_2O \Leftrightarrow CO_2 + H_2$. Medium pressure steam is extracted from the steam cycle. The steam demand is 0.5 kg H_2O/kg gas. The shift takes place in multiple catalytic reactors with intercooling at about 250–350°C.

After the sift, the CO_2 content in the gas has increased to 30 vol%. Because of the high pressure and concentration, a CO_2 removal process based on physical absorption, like the Selexol process, is most suitable for this application. The

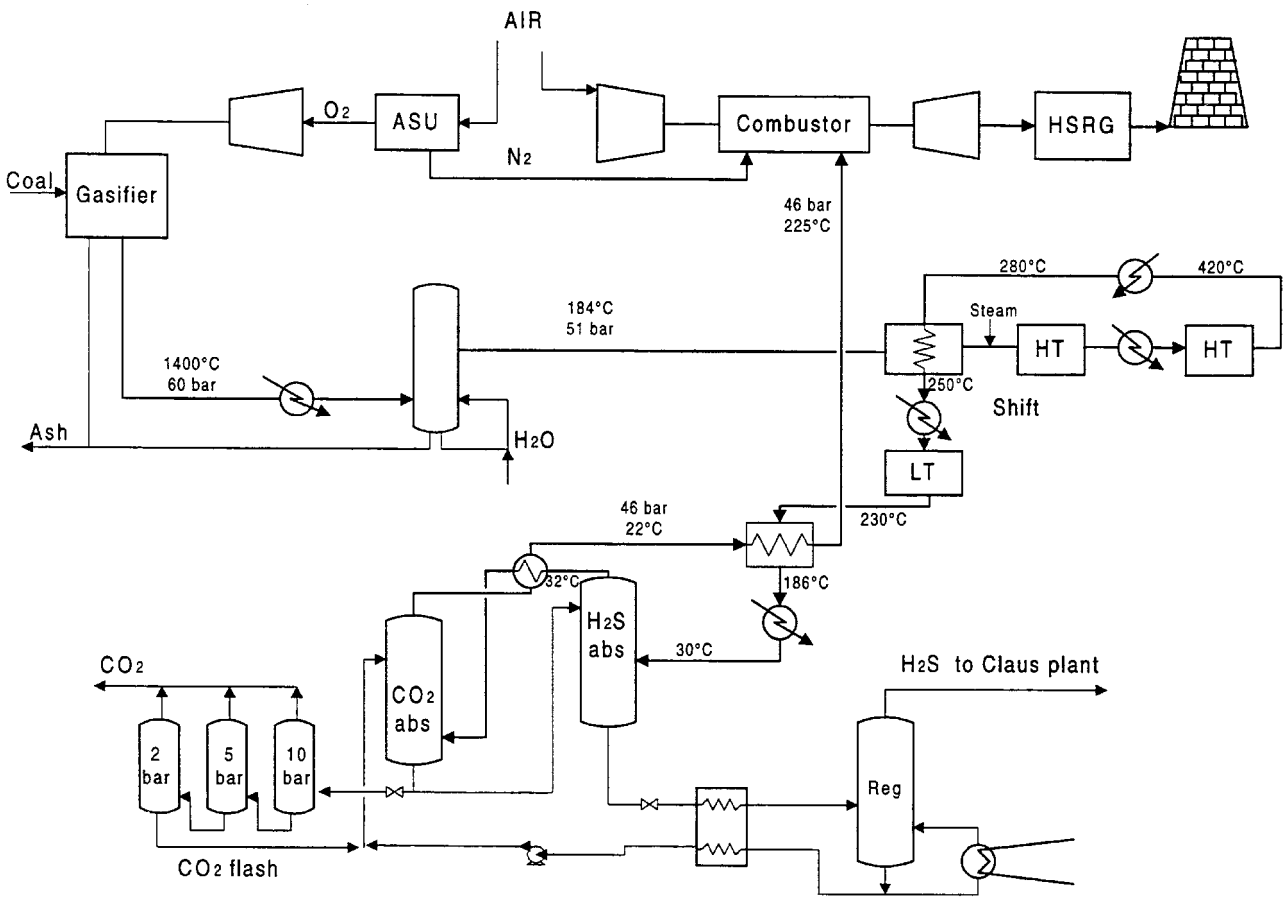


FIG. C-15 Process scheme for a coal-based IGCC (integrated gasification combined cycle) power plant with shift (conversion of CO to CO_2) and CO_2 capture upstream of the gas turbine combustion chamber. (Source: Vattenfall Utveckling AB.)

removal efficiency is assumed to be 90 percent. Hydrogen sulfide is selectively removed before the CO_2 removal. The sulfur-rich gas is transferred to a Claus unit, where elementary sulfur is produced. Regeneration of the absorbent is achieved by temperature increase and flashing. Low-pressure steam for the regeneration is extracted from the steam cycle. The dry isolated CO_2 is pressurized and liquefied.

After CO_2 removal, the hydrogen-rich fuel gas is burned in a gas turbine. A hydrogen-rich gas would most likely be a good gas turbine fuel. The gas turbine combustor must, of course, be designed for this type of fuel gas, since hydrogen has somewhat different combustion characteristics than natural gas. Combustion of hydrogen/steam mixtures for utilization in future advanced gas turbine cycles is investigated by Westinghouse. As in an IEA study, a Siemens V94.4 gas turbine has been assumed.

Carbon dioxide neutral coproduction of methanol, power, and district heating

Carbon dioxide neutral production and utilization of methanol as an automotive fuel for the transport sector integrated with production of electric power and district heat could be achieved with biomass combined with natural gas or coal as a raw material. An amount of CO_2 corresponding to the carbon in the fossil fuel then has to be captured and disposed into, e.g., an aquifer. Examples of a few such options

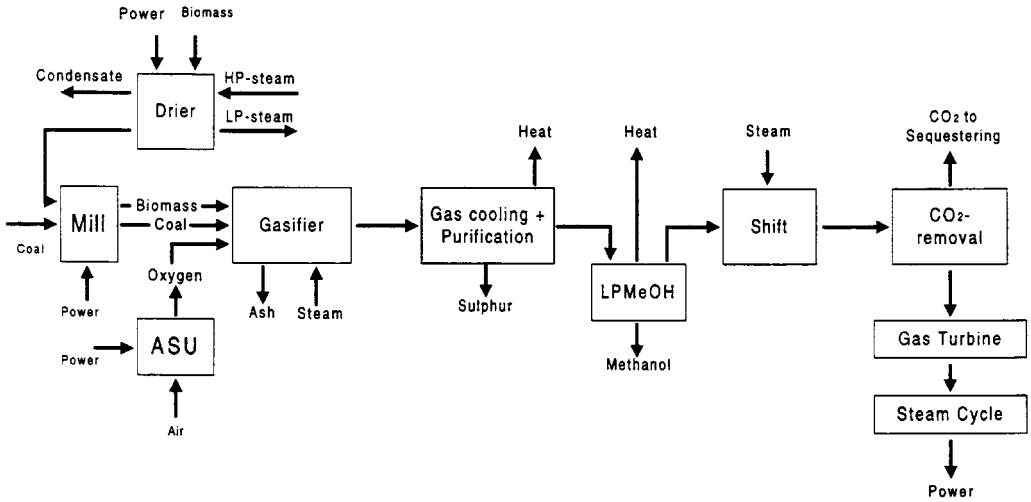


FIG. C-16 CO_2 neutral production of methanol, power and district heat by co-gasification of biomass and coal combined with CO_2 capture. (Source: Vattenfall Utveckling AB.)

have been studied based on IEA studies, other literature, and Vattenfall in-house information.

Co-gasification of biomass and coal. Coal and biomass are gasified in an entrained flow gasifier at 1400°C, 40 bar with oxygen and steam (Fig. C-16). Before being gasified, the biomass is dried in a steam drier, lowering its moisture content from about 50 percent to 10 percent, followed by milling. The air separation unit (ASU) is based on cryogenic separation. The syngas generated in the gasifier is cooled and cleaned from dust and sulfur. Heat is extracted to be used in the steam cycle.

The syngas contains about 28 vol% H_2 and 39 vol% CO at the inlet of the methanol synthesis reactor. Since both biomass and coal have low hydrogen contents, a novel methanol synthesis process under development by Chem. Systems/Air Products in the U.S., called LPMeOH (Liquid Phase Methanol Synthesis), has been selected. This process is less sensitive to the inlet syngas composition—mainly the ratio of $(H_2 - CO_2)/(CO + CO_2)$—than the current commercially available methanol synthesis processes. It has been assumed that 20 percent of the carbon input in the fuel is converted to methanol. Assuming only CO reacts according to $CO + 2H_2 \rightarrow CH_3OH$, the CO conversion is 25 percent on a molar basis. The methanol synthesis reaction is highly exothermic, and the released heat is utilized in the steam cycle.

The unreacted outlet gas from the methanol synthesis reactor contains mainly CO and H_2O. By adding steam, the CO is converted to CO_2 according to $CO + H_2O \Leftrightarrow H_2 + CO_2$ in the shift reactors. Like in an IGCC power plant case, a 95 percent conversion of CO has been assumed. The CO_2 content in the gas then increases from 14 vol% to 44 vol%. CO_2 is captured in a Selexol plant. The removal efficiency has been assumed to be 87 percent, which is close to the assumption for the IGCC power plant case.

After the CO_2 removal, the remaining gas rich in H_2 is burned in the gas turbine combustor. The gas turbine has been scaled to the actual fuel gas capacity from the Siemens V 94.4 gas turbine in the IGCC case, assuming unchanged performance. Heat from the gas turbine exhaust gases is utilized to generate steam for the bottoming cycle in a heat recovery boiler.

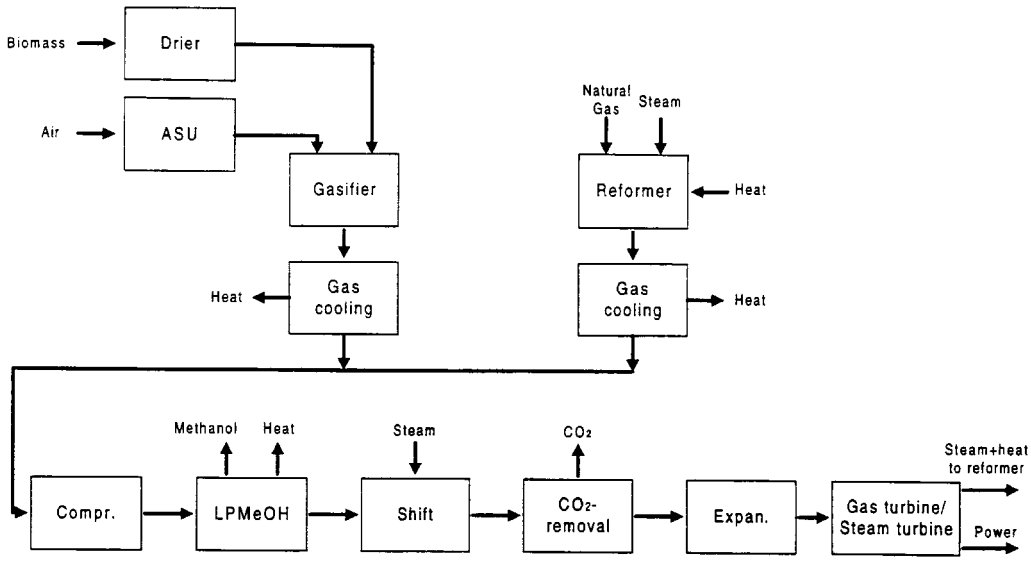


FIG. C-17 CO_2 neutral production of methanol, power, and district heat by gasification of biomass and reforming of natural gas in series combined with CO_2 capture. (Source: Vattenfall Utveckling AB.)

Gasification of biomass and reforming of natural gas in series. In this configuration, syngas is produced from biomass, oxygen, and steam in a fluidized bed gasifier at about 950°C, 20 bar. Before being gasified, the biomass is dried in a steam drier, lowering its moisture content from about 50 percent to 15–20 percent. The syngas from the gasifier is cleaned from dust using a ceramic filter at about 500°C and is then mixed with syngas from a natural gas reformer. Since both the gasifier and the reformer are operating at about 20 bar, the syngas mixture must be compressed before entering the methanol synthesis reactor. Since the gas composition is not optimal for a conventional methanol synthesis process, the LPMeOH process has been selected (Fig. C-17).

The unreacted outlet gas from the methanol synthesis reactor is shifted before the CO_2 removal. The CO_2 content in the gas then increases from 7 to about 18 vol%. Like in the IGCC power plant case, a 90 percent removal has been assumed. The remaining gas, rich in hydrogen (69 vol%), is expanded to the pressure required for the gas turbine combustor. The gas turbine has been scaled to the actual fuel gas capacity from the Siemens V 94.4 gas turbine in the IGCC case, assuming unchanged performance. The heat from the gas turbine exhaust gas is utilized in the steam cycle and for heating the reformer.

Energy efficiencies and costs when capturing carbon dioxide

The calculated efficiencies with and without CO_2 capture for the gas turbine–based power plants and for the described examples of CO_2 neutral coproduction of methanol, electric power, and district heat are summarized in Table C-2. Additional costs, due to the CO_2 capture, were estimated based on data from IEA studies, other literature, and this information source's in-house information. The results are summarized in Table C-2.

CO_2 capture and recovery consumes electricity and energy at high temperatures at the same time as energy at low temperatures can be recovered. This does not

TABLE C-2 Calculated Efficiencies and Carbon Dioxide Capture Costs for Electric Power Plants and for Carbon Dioxide Neutral Coproduction of Methanol, Electric Power, and District Heating

Capital costs:	7 percent real interest rate, 20 years economic lifetime
Fuel costs:	Natural gas 100 SEK/MWh, coal 50 SEK/MWh, biomass 120 SEK/MWh
District heat credit:	150 SEK/MWh
For Methanol and Electricity	
Methanol credit:	230 SEK/MWh (assumed world market price 1 SEK/liter)
Electric power credit:	280 SEK/MWh (calculated production cost from natural gas without CO_2 capture)

	Power Plant 6000 h/year			Methanol + Electricity 8000 h/year	
			Biomass, MW (LHV)	385	385
Fossil Fuel, MW (LHV)	Natural Gas 645	Coal IGCC 870	Fossil Fuel, MW (LHV)	Natural Gas 1245	Coal 870
			Methanol prod., MW	280	365
Net electricity, MW	300	320	Net electricity, MW	505	350
District heating, MW	55	85	District heating, MW	90	45
Net efficiencies			Net efficiencies		
Without CO_2 Capture			*With CO_2 Capture*		
Electric power, %	56	43	Methanol, %	17	29
With CO_2 Capture					
Electric power, %	47	37	Electric power, %	31	28
District heating, %	7	10	District heating, %	6	4
Total, %	54	47	Total, %	54	61
Captured CO_2, tons/h	120	260	Captured CO_2, tons/h	260	315
CO_2 Capture Costs			CO_2 Capture Costs		
SEK/ton CO_2	220	100	SEK/ton CO_2	340	145
SEK/MWh el	150	110	SEK/MWh (el + methanol)	110	65
		190 (IGCC with -PE without)			

mean a total energy loss for a power plant, but since a smaller fraction of the total energy will be available at higher temperatures (exergy loss), and a larger fraction will be available at lower temperatures, the electric efficiencies will be reduced. With the numerous district heating networks in Sweden, this means that it could be possible to compensate for the losses of electric efficiency by recovering energy at low temperatures as district heating. If sufficient quantities of low temperature heat can be sold, the total efficiencies will be nearly the same as for the corresponding power plants without capture of CO_2.

The calculated capture costs per metric ton CO_2 become higher for lower capture capacities than for higher capacities, which shows the scale economy for the capture and recovery process parts.

The estimated costs per ton CO_2 for the methanol and electricity cases are higher than for the power plants with similar capture capacities. This is due mainly to the choices of credits for methanol and electric power. Both these credits reflect natural gas–based production, which is less complex and therefore less costly than when solid fuels—biomass and/or coal—are used. At the same time, the capture costs per MWh (electricity + methanol) become about the same or lower as for the power plants. The main reason for this is that substantial fractions of the total fuel inputs

are biomass and only the CO_2 corresponding to the carbon in the fossil fuel input is captured. The quantities of captured CO_2 then become smaller in relation to the quantities of methanol plus electricity than the quantities of captured CO_2 in relation to the quantities of electricity produced in the power plants.

The most economic power production process for coal without CO_2 capture is PF (pulverized coal combustion) with a supercritical steam cycle. Its electric efficiency is today almost as high as the efficiency for an IGCC. With CO_2 capture, the steam consumption, auxiliary power need and cost for CO_2 capture from a boiler flue gas with low CO_2 concentration (requires chemical absorption) are higher than for CO_2 capture from a pressurized fuel gas with high CO_2 concentration (physical absorption will be sufficient). Consequently, the loss of electric power efficiency and the cost increase for CO_2 capture become substantially lower for an IGCC than for a PF. This results in IGCC being the most economic coal-based power process with CO_2 capture, also achieving the highest electric efficiency.

Transport and disposal of carbon dioxide

Captured CO_2 could be transported into an aquifer using a pipeline, ship, or possibly a train. A pipeline is more economic than ship (or train) for shorter distances, while ship transport seems to be the most feasible option for longer distances. However, for larger CO_2 quantities, pipelines may be the most economic option also for longer distances. The transport costs per ton CO_2 vary due to the distance and the quantities. To give an idea of these costs, estimated transport costs for two cases, a 40-km pipeline and a 700-km ship transport are shown in Table C-3.

The costs for injection of CO_2 into an aquifer have been estimated to 60–70 SEK/ton CO_2 ("SEK" is Swedish kroner) for a few studied examples based mainly on data from Danish studies on aquifer disposal.

Examples of total costs for capture, transport, and disposal of carbon dioxide

The estimated total cost for capture and recovery, transport, and injection into an aquifer of CO_2 consequently vary depending on energy production process, transport distance, and the scale—especially for the CO_2 transport. This is illustrated in Table C-4.

TABLE C-3 Estimated Examples of Transport Costs for Carbon Dioxide (Ekström, 1997)

	Pipeline	Ship
Distance, km	40	700
Quantity of transported CO_2, tons/h	120	90
Transport cost, SEK/ton CO_2	20	110

TABLE C-4 Summary of Estimated Costs for the Capture, Transport, and Disposal of Carbon Dioxide

	Power Plants 300–320 MW el 6000 h/year	Methanol and Electricity Plants 715–785 MW (el + methanol) 8000 h/year
Captured CO_2, tons/h	120–260	260–315
Capture and Recovery of CO_2, SEK/ton CO_2	100–220	145–340
Transport to Aquifer (40–700 km), SEK/ton CO_2	20–110	
Injection into Aquifer, SEK/ton CO_2	60–70	

For most of the studied cases, the estimated total costs are in the same order of magnitude as the current carbon dioxide tax in Sweden (365 SEK/ton CO_2).

If the distance to a suitable aquifer is long, the CO_2 transport costs can be significantly reduced, by collecting the captured CO_2 from more than one energy-producing plant into one high-capacity pipeline, because of the favorable economy of scale for pipeline transport.

Conclusions

Conclusions from the study are:

- Capture and disposal of CO_2 may be considered as one of the opportunities to reduce the total Swedish CO_2 emissions.
- The total estimated costs per ton CO_2 captured and disposed are, with the assumptions used, of the same order of magnitude as the current Swedish CO_2 taxes. Plant owners will have to be credited for the captured and disposed CO_2 in order to make this option economically justifiable and interesting for them.
- The process steps for CO_2 capture and recovery are well-proven and large-scale natural gas or coal-fired gas turbine–based power plants with CO_2 capture could be constructed using commercially available techniques.
- Pipeline transport of CO_2 is practiced. Large-scale ship or train transport systems are not developed for CO_2.
- Injection and disposal of CO_2 into aquifers is now at full-scale operation in the North Sea.
- The costs for CO_2 capture vary depending mainly on energy conversion process and scale. Ongoing process developments are likely to improve the economy.
- It will be important for the total economy to find favorable combinations of energy conversion, CO_2 capture and recovery, transport, and disposal. There is also a need to reduce uncertainties in the available basis for estimation of costs for large-scale transport, injection, and disposal of CO_2 into aquifers.
- For power plants, the losses of electric power efficiencies due to CO_2 capture could be compensated for in terms of maintaining total energy efficiencies, if sufficient quantities of district heat could be sold. The heat quantities produced for the plant capacities chosen in this study are likely to be suitable for a reasonable number of Swedish district heating networks.
- The CO_2 emissions from the transport sector could be reduced by combining CO_2 capture with the production of automotive fuel (methanol), electric power, and heat from biomass combined with natural gas or coal. Optimization work is needed to find the most favorable process configurations.
- Favorably localized aquifers that should be suitable for CO_2 disposal and with a substantial storing capacity exist in South West Skåne-Denmark (eastern part of Zealand).
- Several aspects need to be clarified regarding aquifer conditions, disposal, and large-scale CO_2 transport in order to make more accurate assessments. Swedish–Norwegian–Danish cooperation projects regarding further clarification of techniques and geology for aquifer disposal and large-scale CO_2 transport systems would be beneficial.
- A Swedish implementation of capture and disposal of CO_2 from combustion of fossil fuels will require political decisions and development of legal frameworks. The investigations of these aspects were outside the scope of this study.

Castings (*see* Metallurgy)

Cells

Chemical cells are a specialized, often custom-designed item, common in applications such as metallurgical processes, for example, electroplating. *See* Some Commonly Used Specifications, Codes, Standards, and Texts.

Cement; Portland Cement

Cement is made by several companies around the world to exact standard specifications. When mixed with varying grades of air, aggregate, and soil, concrete of varying strengths is produced. Extracts from American Society for Testing and Materials (ASTM) specification C 150-95 for Portland cement follow.

The manufacture of cement raises significant environmental concerns, and standards have been developed around these considerations. Extracts from a paper that describes cement manufacture in Canada and Canada's emission guidelines with respect to cement manufacturing kilns, also follows. Specifications will vary slightly from country to country. Legislation, such as U.S.–Canada Air Agreements and global conventions on issues such as air quality and global warming (e.g., Kyoto, Rio, Montreal) are helping introduce communality into specifications of products that affect air emissions, such as cement.

Extracts from *Standard Specification for Portland Cement*, ASTM C 150-95[1]

This standard is issued under the fixed designation C 150; the number immediately following the designation indicates the year of original adoption or, in the case of revision, the year of last revision. A number in parentheses indicates the year of last reapproval. A superscript epsilon (ε) indicates an editorial change since the last revision or reapproval.

This standard has been approved for use by agencies of the Department of Defense. Consult the DoD Index of Specifications and Standards for the specific year of issue which has been adopted by the Department of Defense.

1. Scope

1.1 This specification covers eight types of Portland cement, as follows (see Note 1 in section 4 below):

1.1.1 *Type I*—For use when the special properties specified for any other type are not required.

1.1.2 *Type IA*—Air-entraining cement for the same uses as Type I, where air-entrainment is desired.

1.1.3 *Type II*—For general use, more especially when moderate sulfate resistance or moderate heat of hydration is desired.

1.1.4 *Type IIA*—Air-entraining cement for the same uses as Type II, where air-entrainment is desired.

1.1.5 *Type III*—For use when high early strength is desired.

1.1.6 *Type IIIA*—Air-entraining cement for the same use as Type III, where air-entrainment is desired.

---

[1] This specification is under the jurisdiction of ASTM Committee C-1 on Cement and is the direct responsibility of Subcommittee C01.10 on Portland Cement. Current edition approved June 15, 1995. Published August 1995. Originally published as C 150-40 T. Last previous edition C 150-94b.

1.1.7 *Type IV*—For use when a low heat of hydration is desired.

1.1.8 *Type V*—For use when high sulfate resistance is desired.

1.2 When both SI and inch-pound units are present, the SI units are the standard. The inch-pound units are approximations listed for information only.

2. Referenced documents

2.1 *ASTM Standards*:

C 33 Specification for Concrete Aggregates[2]

C 109 Test Method for Compressive Strength of Hydraulic Cement Mortars (Using 2-in. or 50-mm Cube Specimens)[3]

C 114 Test Methods for Chemical Analysis of Hydraulic Cement[3]

C 115 Test Method for Fineness of Portland Cement by the Turbidimeter[3]

C 151 Test Method for Autoclave Expansion of Portland Cement[3]

C 183 Practice for Sampling and the Amount of Testing of Hydraulic Cement[3]

C 185 Test Method for Air Content of Hydraulic Cement Mortar[3]

C 186 Test Method for Heat of Hydration of Hydraulic Cement[3]

C 191 Test Method for Time of Setting of Hydraulic Cement by Vicat Needle[3]

C 204 Test Method for Fineness of Hydraulic Cement by Air Permeability Apparatus[3]

C 226 Specification for Air-Entraining Additions for Use in the Manufacture of Air-Entraining Portland Cement[3]

C 266 Test Method for Time of Setting of Hydraulic Cement Paste by Gillmore Needles[3]

C 451 Test Method for Early Stiffening of Portland Cement (Paste Method)[3]

C 452 Test Method for Potential Expansion of Portland Cement Mortars Exposed to Sulfate[3]

C 465 Specification for Processing Additions for Use in the Manufacture of Hydraulic Cements[3]

C 563 Test Method for Optimum SO_3 in Portland Cement[3]

C 1038 Test Method for Expansion of Portland Cement Mortar Bars Stored in Water[3]

3. Terminology

3.1 *Definitions*:

3.1.1 *Portland cement*—a hydraulic cement produced by pulverizing clinker consisting essentially of hydraulic calcium silicates usually containing one or more of the forms of calcium sulfate as an interground addition.

3.1.2 *Air-entraining portland cement*—a hydraulic cement produced by pulverizing clinker consisting essentially of hydraulic calcium silicates, usually containing one or more of the forms of calcium sulfate as an interground addition, and with which there has been interground an air-entraining addition.

4. Ordering information

4.1 Orders for material under this specification shall include the following:

4.1.1 This specification number and date,

---

[2] *Annual Book of ASTM Standards*, Vol. 04.02.

[3] *Annual Book of ASTM Standards*, Vol. 04.01.

4.1.2 Type or types allowable. If no type is specified, Type I shall be supplied,

4.1.3 Any optional chemical requirements from Table C-6, if desired,

4.1.4 Type of setting-time test required, Vicat or Gillmore. If not specified, the Vicar shall be used,

4.1.5 Any optional physical requirements from Table C-8, if desired.

NOTE 1—Attention is called to the fact that cements conforming to the requirements for all types may not be carried in stock in some areas. In advance of specifying the use of other than Type I cement, it should be determined whether the proposed type of cement is or can be made available.

5. Additions

5.1 The cement covered by this specification shall contain no addition except as follows:

5.1.1 Water or calcium sulfate, or both, may be added in amounts such that the limits shown in Table C-5 for sulfur trioxide and loss-on-ignition shall not be exceeded.

5.1.2 At the option of the manufacturer, processing additions may be used in the manufacture of the cement, provided such materials in the amounts used have been shown to meet the requirements of Specification C 465.

5.1.3 Air-entraining portland cement shall contain an interground addition conforming to the requirements of Specification C 226.

6. Chemical composition

6.1 Portland cement of each of the eight types shown in section 1 above shall conform to the respective standard chemical requirements prescribed in Table C-5. In addition, optional chemical requirements are shown in Table C-6.

7. Physical properties

7.1 Portland cement of each of the eight types shown in section 1 above shall conform to the respective standard physical requirements prescribed in Table C-7. In addition, optional physical requirements are shown in Table C-8.

8. Sampling

8.1 When the purchaser desires that the cement be sampled and tested to verify compliance with this specification, sampling and testing should be performed in accordance with Practice C 183.

8.2 Practice C 183 is not designed for manufacturing quality control and is not required for manufacturer's certification.

9. Test methods

9.1 Determine the applicable properties enumerated in this specification in accordance with the following test methods:

9.1.1 *Air Content of Mortar*—Test Method C 185.

9.1.2 *Chemical Analysis*—Test Methods C 114.

9.1.3 *Strength*—Test Method C 109.

9.1.4 *False Set*—Test Method C 451.

9.1.5 *Fineness by Air Permeability*—Test Method C 204.

9.1.6 *Fineness by Turbidimeter*—Test Method C 115.

9.1.7 *Heat of Hydration*—Test Method C 186.

TABLE C-5 Standard Chemical Requirements

Cement Type[A]	I and IA	II and IIA	III and IIIA	IV	V
Silicon dioxide (SiO_2), min, %	—	20.0	—	—	—
Aluminum oxide (Al_2O_3), max, %	—	6.0	—	—	—
Ferric oxide (Fe_2O_3), max, %	—	6.0	—	6.5	—
Magnesium oxide (MgO), max, %	6.0	6.0	6.0	6.0	6.0
Sulfur trioxide (SO_3),[B] max, %					
When (C_3A)[C] is 8% or less	3.0	3.0	3.5	2.3	2.3
When (C_3A)[C] is more than 8%	3.5	[D]	4.5	[D]	[D]
Loss on ignition, max, %	3.0	3.0	3.0	2.5	3.0
Insoluble residue, max, %	0.75	0.75	0.75	0.75	0.75
Tricalcium silicate (C_3S)[C] max, %	—	—	—	35[F]	—
Dicalcium silicate (C_2S)[C] min, %	—	—	—	40[E]	—
Tricalicum aluminate (C_3A)[C] max, %	—	8	15	7[E]	5[F]
Tetracalcium aluminoferrite plus twice the tricalcium aluminate[C] ($C_4AF + 2(C_3A)$), or solid solution ($C_4AF + C_2F$), as applicable, max, %	—	—	—	—	25[F]

[A] See Note 1 in section 4.

[B] There are cases where optimum SO_3 (using Test Method C 553) for a particular cement is close to or in excess of the limit in this specification. In such cases where properties of a cement can be improved by exceeding the SO_3 limits stated in this table, it is permissible to exceed the values in the table, provided it has been demonstrated by Test Method C 1038 that the cement with the increased SO_3 will not develop expansion in water exceeding 0.020% at 14 days. When the manufacturer supplies cement under this provision, he shall, upon request, supply supporting data to the purchaser.

[C] The expressing of chemical limitations by means of calculated assumed compounds does not necessarily mean that the oxides are actually or entirely present as such compounds.

When expressing compounds, C – CaO, S – SiO_2, A – Al_2O_3, F – Fe_2O_3. For example, C_3A – $3CaO{\cdot}Al_2O_3$.

Titanium dioxide and phosphorus pentoxide (TiO_2 and P_2O_5) shall be included with the Al_2O_3 content. The value historically and traditionally used for Al_2O_3 in calculating potential compounds for specification purposes is the ammonium hydroxide group minus ferric oxide ($R_2O_3 - Fe_2O_3$) as obtained by classical wet chemical methods. This procedure includes as Al_2O_3 the TiO_2, P_2O_5 and other trace oxides which precipitate with the ammonium hydroxide group in the classical wet chemical methods. Many modern instrumental methods of cement analysis determine aluminum or aluminum oxide directly without the minor and trace oxides included by the classical method. Consequently, for consistency and to provide comparability with historic data and among various analytical methods, when calculating potential compounds for specification purposes, those using methods which determine Al or Al_2O_3 directly should add to the determined Al_2O_3 weight quantities of P_2O_5. TiO_2 and any other oxide except Fe_2O_3 which would precipitate with the ammonium hydroxide group when analyzed by the classical method and which is present in an amount of 0.05 weight % or greater. The weight percent of minor or trace oxides to be added to Al_2O_3 by those using direct methods may be obtained by actual analysis of those oxides in the sample being tested or estimated from historical data on those oxides on cements from the same source, provided that the estimated values are identified as such.

When the ratio of percentages of aluminum oxide to ferric oxide is 0.64 or more, the percentages of tricalcium silicate, dicalcium silicate, tricalcium aluminate, and tetracalcium aluminoferrite shall be calculated from the chemical analysis as follows:

$$\text{Tricalcium silicate} = (4.071 \times \%\ CaO) - (7.600 \times \%\ SiO_2) - (6.718 \times \%\ Al_2O_3) - (1.430 \times \%\ Fe_2O_3) - (2.852 \times \%\ SO_3)$$
$$\text{Dicalcium silicate} = (2.867 \times \%\ SiO_2) - (0.7544 \times \%\ C_3S)$$
$$\text{Tricalcium aluminate} = (2.650 \times \%\ Al_2O_3) - (1.692 \times \%\ Fe_2O_3)$$
$$\text{Tetracalcium aluminoferrite} = 3.043 \times \%\ Fe_2O_3$$

When the alumina-ferric oxide ratio is less than 0.64, a calcium aluminoferrite solid solution (expressed as ss($C_4AF + C_2F$)) is formed. Contents of this solid solution and of tricalcium silicate shall be calculated by the following formulas:

$$ss(C_4AF + C_2F) = (2.100 \times \%\ Al_2O_3) + (1.702 \times \%\ Fe_2O_3)$$
$$\text{Tricalcium silicate} = (4.071 \times \%\ CaO) - (7.600 \times \%\ SiO_2) - (4.479 \times \%\ Al_2O_3) - (2.859 \times \%\ Fe_2O_3) - (2.852 \times \%\ SO_3).$$

No tricalcium aluminate will be present in cements of this composition. Dicalcium silicate shall be calculated as previously shown.

In the calculation of all compounds the oxides determined to the nearest 0.1% shall be used.

All values calculated as described in this note shall be reported to the nearest 1%.

[D] Not applicable.

[E] Does not apply when the heat of hydration limit in Table C-8 is specified.

[F] Does not apply when the sulfate resistance limit in Table C-8 specified.

9.1.8 *Autoclave Expansion*—Test Method C 151.

9.1.9 *Time of Setting by Gillmore Needles*—Test Method C 266.

9.1.10 *Time of Setting by Vicat Needles*—Test Method C 191.

9.1.11 *Sulfate Resistance*—Test Method C 452 (sulfate expansion).

9.1.12 *Calcium Sulfate (expansion of) Mortar*—Test Method C 1038.

9.1.13 *Optimum SO_3*—Test Method C 563.

TABLE C-6 Optional Chemical Requirements[A]

Cement Type	I and IA	II and IIA	III and IIIA	IV	V	Remarks
Tricalcium aluminate (C_3A),[B] max, %	—	—	8	—	—	For moderate sulfate resistance
Tricalcium aluminate (C_3A),[B] max, %	—	—	5	—	—	For high sulfate resistance
Sum of tricalcium silicate and tricalcium aluminate,[B] max, %	—	58[C]	—	—	—	For moderate heat of hydration
Equivalent alkalies ($Na_2O + 0.658K_2O$), max, %	0.60[D]	0.60[D]	0.60[D]	0.60[D]	0.60[D]	Low-alkali cement

[A] These optional requirements apply only if specifically requested. Availability should be verified. See Note 1 in section 4.

[B] The expressing of chemical limitations by means of calculated assumed compounds does not necessarily mean that the oxides are actually or entirely present as such compounds.

When expressing compounds, C = CaO, S = SiO_2, A = Al_2O_3, F = Fe_2O_3. For example, $C_3A = 3CaO{\cdot}Al_2O_3$.

Titanium dioxide and phosphorus pentoxide (TiO_2 and P_2O_5) shall be included with the Al_2O_3 content. The value historically and traditionally used for Al_2O_3 in calculating potential compounds for specification purposes is the ammonium hydroxide group minus ferric oxide ($R_2O_3 - Fe_2O_3$) as obtained by classical wet chemical methods. This procedure includes as Al_2O_3 the TiO_2, P_2O_5 and other trace oxides which precipitate with the ammonium hydroxide group in the classical wet chemical methods. Many modern instrumental methods of cement analysis determine aluminum or aluminum oxide directly without the minor and trace oxides included by the classical method. Consequently, for consistency and to provide comparability with historic data and among various analytical methods, when calculating potential compounds for specification purposes, those using methods which determine Al or Al_2O_3 directly should add to the determined Al_2O_3 weight quantities of P_2O_5. TiO_2 and any other oxide except Fe_2O_3 which would precipitate with the ammonium hydroxide group when analyzed by the classical method and which is present in an amount of 0.05 weight % or greater. The weight percent of minor or trace oxides to be added to Al_2O_3 by those using direct methods may be obtained by actual analysis of those oxides in the sample being tested or estimated from historical data on those oxides on cements from the same source, provided that the estimated values are identified as such.

When the ratio of percentages of aluminum oxide to ferric oxide is 0.64 or more, the percentages of tricalcium silicate, dicalcium silicate, tricalcium aluminate and tetracalcium aluminoferrite shall be calculated from the chemical analysis as follows:

$$\text{Tricalcium silicate} = (4.071 \times \%\ \text{CaO}) - (7.600 \times \%\ \text{SiO}_2) - (6.718 \times \%\ \text{Al}_2\text{O}_3) - (1.430 \times \%\ \text{Fe}_2\text{O}_3) - (2.852 \times \%\ \text{SO}_3)$$

$$\text{Dicalcium silicate} = (2.867 \times \%\ \text{SiO}_2) - (0.7544 \times \%\ \text{C}_3\text{S})$$

$$\text{Tricalcium aluminate} = (2.650 \times \%\ \text{Al}_2\text{O}_3) - (1.692 \times \%\ \text{Fe}_2\text{O}_3)$$

$$\text{Tetracalcium aluminoferrite} = 3.043 \times \%\ \text{Fe}_2\text{O}_3$$

When the alumina-ferric oxide ratio is less than 0.64, a calcium aluminoferrite solid solution (expressed as $ss(C_4AF + C_2F)$) is formed. Contents of this solid solution and of tricalcium silicate shall be calculated by the following formulas:

$$ss(C_4AF + C_2F) = (2.100 \times \%\ \text{Al}_2\text{O}_3) + (1.702 \times \%\ \text{Fe}_2\text{O}_3)$$

$$\text{Tricalcium silicate} = (4.071 \times \%\ \text{CaO}) - (7.600 \times \%\ \text{SiO}_2) - (4.479 \times \%\ \text{Al}_2\text{O}_3) - (2.859 \times \%\ \text{Fe}_2\text{O}_3) - (2.852 \times \%\ \text{SO}_3).$$

No tricalcium aluminate will be present in cements of this composition. Dicalcium silicate shall be calculated as previously shown.

In the calculation of all compounds the oxides determined to the nearest 0.1% shall be used.

All values calculated as described in this note shall be reported to the nearest 1%.

[C] The optional limit for heat of hydration in Table C-8 shall not be requested when this optional limit is requested.

[D] This limit may be specified when the cement is to be used in concrete with aggregates that may be deleteriously reactive. Reference should be made to Specification C 33 for suitable criteria of deleterious reactivity.

10. Inspection

10.1 Inspection of the material shall be made as agreed upon by the purchaser and the seller as part of the purchase contract.

11. Rejection

11.1 The cement may be rejected if it fails to meet any of the requirements of this specification.

11.2 Cement remaining in bulk storage at the mill, prior to shipment, for more than 6 months, or cement in bags in local storage in the hands of a vendor for more than 3 months, after completion of tests, may be retested before use and may be rejected if it fails to conform to any of the requirements of this specification.

11.3 Packages shall identify the mass contained as net weight. Packages more than 2 percent below the weight marked thereon may be rejected; and if the average weight of packages in any shipment, as shown by weighing 50 packages taken at random, is less than that marked on the packages, the entire shipment may be rejected.

TABLE C-7 Standard Physical Requirements

Cement Type[A]	I	IA	II	IIA	III	IIIA	IV	V
Air content of mortar,[B] volume %:								
max	12	22	12	22	12	22	12	12
min	—	16	—	16	—	16	—	—
Fineness,[C] specific surface, m^2/kg (alternative methods):								
Turbidimeter test, min	160	160	160	160	—	—	160	160
Air permeability test, min	280	280	280	280	—	—	280	280
Autoclave expansion, max, %	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80
Strength, not less than the values shown for the ages indicated below:[D]								
Compressive strength, MPa (psi):								
1 day	—	—	—	—	12.0 (1,740)	10.0 (1,450)	—	—
3 days	12.0 (1,740)	10.0 (1,450)	10.0 (1,450) 7.0[F] (1,020)[F]	8.0 (1,160) 6.0[F] (870)[F]	24.0 (3,480)	19.0 (2,760)	—	8.0 (1,160)
7 days	19.0 (2,760)	16.0 (2,320)	17.0 (2,470) 12.0[F] (1,740)[F]	14.0 (2,030) 9.0[F] (1,310)[F]	—	—	7.0 (1,020)	15.0 (2,180)
28 days	—	—	—	—	—	—	17.0 (2,470)	21.0 (3,050)
Time of setting (alternative methods):[E]								
Gillmore test:								
Inital set, min, not less than	60	60	60	60	60	60	60	60
Final set, min, not more than	600	600	600	600	600	600	600	600
Vicat test:[G]								
Time of setting, min, not less than	45	45	45	45	45	45	45	45
Time of setting, min, not more than	375	375	375	375	375	375	375	375

[A] See Note 1 in section 4.
[B] Compliance with the requirements of this specification does not necessarily ensure that the desired air content will be obtained in concrete.
[C] Either of the two alternative fineness methods may be used at the option of the testing laboratory. However, when the sample falls to meet the requirements of the air-permeability test, the turbidimeter test shall be used, and the requirements in this table for the turbidimetric method shall govern.
[D] The strength at any specified test age shall be not less than that attained at any previous specified test age.
[E] The purchaser should specify the type of setting-time test required. In case he does not so specify, the requirements of the Vicat test only shall govern.
[F] When the optional heat of hydration or the chemical limit on the sum of the tricalcium silicate and tricalcium aluminate is specified.
[G] The time of setting is that described as initial setting time in Test Method C 191.

12. Manufacturer's statement

12.1 At the request of the purchaser, the manufacturer shall state in writing the nature, amount, and identity of the air-entraining agent used, and of any processing addition that may have been used, and also, if requested, shall supply test data showing compliance of such air-entraining addition with the provisions of Specification C 226, and of any such processing addition with Specification C 465.

13. Packaging and package marking

13.1 When the cement is delivered in packages, the words "Portland Cement," the type of cement, the name and brand of the manufacturer, and the mass of the cement contained therein shall be plainly marked on each package. When the cement is an air-entraining type, the words "air entraining" shall be plainly marked on each package. Similar information shall be provided in the shipping documents

accompanying the shipment of packaged or bulk cement. All packages shall be in good condition at the time of inspection.

NOTE 2—With the change to SI units, it is desirable to establish a standard SI package for portland cements. To that end 40 kg (88.18 lb) provides a convenient even-numbered mass reasonably similar to the traditional 94-lb (42.6384-kg) package.

14. Storage

14.1 The cement shall be stored in such a manner as to permit easy access for proper inspection and identification of each shipment, and in a suitable weather-tight building that will protect the cement from dampness and minimize warehouse set.

15. Manufacturer's certification

15.1 Upon request of the purchaser in the contract or order, a manufacturer's report shall be furnished at the time of shipment stating the results of tests made on samples of the material taken during production or transfer and certifying that the cement conforms to applicable requirements of this specification.

As previously noted, emissions guidelines for cement kilns vary from country to country. The sample country discussed in this case is Canada.

Extracts from "Development of CCME National Emission Guidelines for Cement Kilns"*

The guideline proposes emission limits for new kilns in the cement manufacturing industry, and makes some recommendations on emission reductions from existing plants that are being modified or upgraded. Regional or provincial regulatory authorities could decide to impose stricter standards in response to local air-quality problems.

The guideline was prepared through extensive consultation between industry, governments, and environmental groups. Principles that were considered to be important were those of pollution prevention, energy efficiency, cost-effectiveness, and a comprehensive view toward minimizing various emissions to reduce air pollution and greenhouse gases. A future guideline for lime kilns will also be developed subsequent to further research with the lime industry. Tables C-5 through C-7 describe characteristics of cement.

Cement kilns in the sample country (Canada)

Portland cement is mixed with sand, aggregates, and water to form the basic building material known as concrete. The production of cement, which comprises 10–15 percent of the final concrete mixture, is based on the conversion (pyroprocessing) of a mixture of limestone ($CaCO_3$) and shale or clay, into clinker material consisting of compounds of calcium oxide (CaO), by the addition of large quantities of heat in a coal- or gas-fired rotary kiln (Fig. C-18). The raw cement clinker exits the kiln and is usually mixed with 3 to 6 percent gypsum and then finely ground into powder. Sometimes it can also be mixed with flyash or other cementitious additives if blended cements are used. The characteristics of the raw feed materials, the finished product, and the resulting emissions are very site-

* Source: Environment Canada. Adapted with permission.

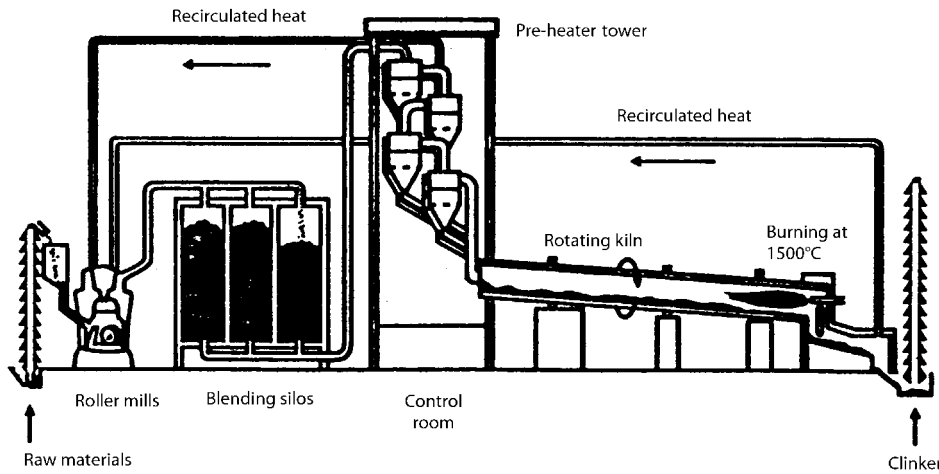


FIG. C-18 Basic schematic of cement production. (Source: Environment Canada.)

TABLE C-8 Optional Physical Requirements[A]

Cement Type	I	IA	II	IIA	III	IIIA	IV	V
False set, final penetration, min, %	50	50	50	50	50	50	50	50
Heat of hydration:								
7 days, max, kJ/kg (cal/g)	—	—	290 (70)[B]	290 (70)[B]	—	—	250 (80)[C]	—
28 days, max, kJ/kg (cal/g)	—	—	—	—	—	—	290 (70)[C]	—
Strength, not less than the values shown:								
Compressive strength, MPa (psi)								
28 days	28.0 (4,060)	22.0 (3,190)	28.0 (4,060) 22.0[B] (3,190)[B]	22.0 (3,190) 18.0[B] (2,610)[B]	—	—	—	—
Sulfate resistance,[D] 14 days, max, % expansion	—	—	—	—	—	—	—	0.040

[A] These optional requirements apply only if specifically requested. Availability should be verified. See Note 1 in section 4.
[B] The optional limit for the sum of the tricalcium silicate and tricalcium aluminate in Table C-6 shall not be requested when this optional limit is requested. These strength requirements apply when either heat of hydration or the sum of tricalcium silicate and tricalcium aluminate requirements are requested.
[C] When the heat of hydration limit is specified, it shall be instead of the limits of C3S, C2S, and C3A listed in Table C-5.
[D] When the sulfate resistance is specified, it shall be instead of the limits of C_3A and $C_4AF + 2C_3A$ listed in Table C-5.

specific depending upon the chemistry of the quarried limestone, the type of cement required, and the type of kiln used. See Tables C-5 through C-8.

The clinkering process begins with the feed of the raw limestone mixture into the higher end of the rotary kiln, where it is exposed to gradually increasing temperatures through an evaporation/preheat zone (100–400°C). In the middle of the kiln temperatures in the 500–900°C range cause calcination, liberating large amounts of carbon dioxide from the limestone. Near the lower burner end the remaining material is sintered at about 1500°C to form clinker pellets, which are subsequently cooled and sent to the grinding mill. Combustion gases flow in the opposite direction toward the stack located at the higher end of the kiln. NO_x formation is dominated by the thermal NO_x mechanism from combustion temperatures approaching 1900°C, in the pyroprocessing stage to sinter the material into a cement clinker, as well as a small quantity from initial limestone calcination.

Clinker production in an older style wet process long kiln, with a slurry raw material feed, uses chain sections at the feed end to contact the feed to evaporate the 30 percent water content. These kilns have high fuel consumption, about 6 GJ/tonne of clinker, as well as higher auxiliary electrical needs. Energy use was reduced by 15–20 percent with the advent of long dry-feed kilns in the mid-1960s, and then again with the use of suspension preheaters at the feed end. These tower preheaters use three to six stages of counterflow contact between the hot exhaust gases and the feed entering a much shorter kiln, resulting in a much improved 3.5 GJ/tonne energy consumption. The 1980s saw the implementation of the precalciner kilns, which are similar to the preheater type, but they move up to half of the fuel burning to the kiln inlet below the preheater tower. This produces calcination outside the kiln, allows for higher capacity shorter kilns, and also produces less NO_x emissions due to the staging aspects of fuel combustion.

The industry is comprised of 18 active cement plants across Canada, with 32 kilns in total, normally producing about ten million tonnes of cement. Kiln capacities range from the 500–1500 tonnes/day for wet and long dry kilns, to 2000–4500 tonnes/day for preheater/precalciner units. Although most of these kilns are of the older wet/long dry design, almost three-quarters of total production is now done by the ten large preheater/precalciner types. As production is shifted toward the new plants, and some of the older kilns such as the Richmond, BC, plant are converted to the higher-efficiency plant, average energy use will improve from 4.5 GJ/tonne to 3.4 GJ/tonne of clinker by year 2010. Fuel use consists mainly of coal and petroleum coke (60 percent), natural gas (25 percent), and oil- and waste-derived fuels (15 percent).

Atmospheric emissions from cement kilns

Air emissions from kilns arise because of the nature and chemistry of the raw material used and from the fuel burned in the kiln. Particulate emissions have traditionally been the main source of concern; PM emissions generally range from 0.3–1.0 kg/tonne from the combined kiln and clinker cooling facilities. Electrostatic precipitators are commonly employed; some humidification may be required to optimize moisture to maintain particle resistivity. Carbon monoxide levels in the ESP must be monitored to avoid explosive conditions. Some plants may also use fabric filter baghouses if flow conditions are appropriate and exhaust temperatures are not too high. Much of the collected kiln dust is often recirculated into the kiln feed.

NO_x emissions from cement kilns are primarily generated from the high combustion temperature at the main burner. They have a wide range of values, depending upon type of kiln and fuel used. See Tables C-9 and C-10. The average emission rate is about 3.7 kg_{NO_x} per tonne of clinker. Total NO_x emissions are in the range of 30–35 kilotonnes, depending on the production split between gas-fired plants (higher NO_x) in Western Canada, and mostly coal-fired plants in Eastern Canada. Table C-9 summarizes findings in the Radian study, based on averages from various research activities and from Canadian and international industry data. In similar kilns, coal firing produces less NO_x than natural gas, since thermal NO_x is the dominant mechanism. Note that this difference is less for precalciner kilns. Pyroprocessing does require high temperature and resulting NO_x emissions in the kiln firing zone, and emission monitoring has been used to optimize this clinker burning zone. Emission prevention potential from combustion can be realized by minimizing fuel use and by transferring the heat input to the feed end of a shorter kiln. There is limited experience with back-end NO_x emission control

TABLE C-9 Average NO_x Emissions from Cement Kilns (kg/tonne)

Fuel	Wet Kiln	Long Dry Kiln	Preheater	Precalciner
Gas	9	7–9.5	5.6	1.7–3
Coal	1.5–4	2.4–4.6	1.5–2.8	1.4–2

TABLE C-10 Comparison of NO_x Control Technologies for Cement Kilns

Control Technology	Technical Feasibility	Potential NO_x Reduction	Annualized Cost Range, 000$/yr	Cost Effectiveness, $/t of NO_x removed	Effect on Clinker Quality	Effect on Other Emissions
COM	High (all types of kilns)	15–30%	—	—	May be either positive or adverse	SO_x, CO, THC may increase
LNB Indirect-fired Direct-fired Kilns	High (all types of kilns)	15–30%	110–160 370–590	340–570 1,280–2,050	May vary with installation; type of fuel may be detrimental	May vary with installation, CO, THC may increase
SAC	High (precalciner, preheater and long kilns	20–50%	120–160 180–220	250–400 660–940	None	CO, THC may increase
SNCR	Medium (precalciner kilns)	40–70%	610–1,250	1,220–1,690	None	Potential for NH_3, PM_{10} emissions
SCR	Low (all types of kilns)	70–90%	3,150–10,050	4,840–7,500	None	Potential for NH_3, PM_{10} emissions; SO_3 may increase

methods. There is evidence that waste-derived fuels, such as tires and solvents, also tend to decrease these emissions.

While the raw material and fuel input of SO_2 into a cement kiln is in the range of 5–12 kg/tonne, the limestone acts as a natural scrubber trapping 90–99 percent of the SO_2 in the clinker product. Emission rates are very site-specific, having even larger variability than NO_x emissions: 1–10 kg_{SO_2}/tonne in wet and long dry kilns and 0.5–2 kg_{SO_2}/tonne in preheater and precalciner units. NO_x and SO_2 emissions are often inversely related in the clinkering process.

Carbon dioxide emissions have also become an important issue, especially since the process of calcination liberates 500 kg_{CO_2}/tonne of clinker. The fuel combustion–related emissions add between 150 and 400 kg/tonne depending on kiln efficiency and fuel used. Total per unit CO_2 emissions will be reduced gradually over the next 20 years through plant efficiency improvements, the use of waste fuels otherwise landfilled, landfill gas as a secondary fuel, and the use of flyash in blended cements to reduce the need for clinker.

Units of emissions reporting vary widely, with the following being common:

- concentration ppmv, 10–11 percent oxygen
- mg per m^3
- kg_{NO_x}/tonne of clinker, or cement product

Each kiln has a characteristic exhaust gas flow rate to allow conversion to a mass-based criterion: older kilns, 2900–3700 m^3/tonne; preheater/precalciners, 2200–2500 m^3/tonne.

NO_x emission control technologies

Several methods of NO_x reduction were assessed, although only some of these have been proven commercially. These include both combustion technologies that reduce or prevent emissions at their point of generation and postcombustion methods that reduce emissions already generated. See Table C-10.

Combustion and operational modifications. Process optimization through combustion modifications (COM) was one of the first options to be considered for reducing emissions. Improvements to be considered include: improved process control and automation systems, reduced excess air and firing temperature, improved raw mix design that optimized clinker cooling and air preheat, and most other modifications that improve thermal efficiency. These low-cost options should be maximized but must be evaluated against degradation of clinker quality and possible increases in other emissions.

Low-NO_x burners. This is one of the main areas of interest in environmental performance of cement kilns. The use of low-NO_x burners (LNBs) is being studied internationally by research organizations and cement companies. Although the concepts of a staged burning zone with exit gas recirculation are commonly used for LNBs in power boilers, the higher temperatures required in kiln pyroprocessing make these methods more difficult to apply for an acceptable quality of clinker. Four types of low-NO_x burners (3 coal, 1 gas) were described in the Radian study, with a wide variety of results (5–30% NO_x reduction) on installed systems, depending upon:

- The initial baseline NO_x emission
- Whether excess air could be kept to design low values to maintain efficiency
- Length of testing time and resulting clinker quality
- Emissions of other pollutants

Operating experience and design will likely improve the concepts. It should be noted that while normal kiln coal burners receive their primary combustion air directly with the fuel, low-NO_x burners require an independent air supply (indirect firing) largely fed from the clinker cooler. On an existing kiln, this represents a significant cost compared to that of the new burner.

Staged air combustion. By breaking the heat input into two locations, short precalciner kilns avoid the need to transfer heat a great distance in a long kiln, thereby reducing emissions and fuel consumption. There is however some thermal and fuel NO_x formed by the fuel-rich precalciner burner. Studies are underway to mitigate this effect by firing some of the precalciner fuel in the initial kiln exhaust to create a reducing condition ($NO_x \rightarrow N_2$), with the remainder fired in the upper calcining section for complete burnout. Staged Air Combustion has the potential for about 20–30 percent emission reduction in new precalciner kilns, but these levels are more difficult to achieve on existing plants. Clinker quality in the upstream portion of the kiln would not be affected.

Selective noncatalytic reduction (SNCR). SNCR using injection of ammonia compounds is also being investigated in Europe as a feasible postcombustion technology. Solid urea or liquid ammonia injection into a high-temperature exhaust stream (900–1000°C) has been done on a trial basis, with a reported 30–60 percent NO_x reduction on precalciner kilns. Issues to be resolved include possible effects on ESP performance, sensitivity to injection location and gas temperature, increased N_2O, CO emissions and NH_3 slip, and the costs and effort in handling ammonia.

Other methods. A related technology, selective catalytic reduction, would operate with an ammonia injection upstream of a catalyst section in a lower temperature range of 300–400°C. This has not yet been employed on high dust and hostile exhaust stream applications such as cement kilns.

NO_x emissions can also be minimized through reductions in clinker/cement ratio, and tests with waste-derived fuels have in some cases shown a positive impact on reducing emissions. The overall environmental impacts of incorporating flyash or slag mixed into the finished cement product to reduce the need for clinker should be considered, as well as the use of substitute waste fuels such as solvents, tires, and landfill gases to supplement traditional fuels. A separate CCME publication was developed entitled "National Guidelines for the Use of Hazardous and Non-Hazardous Wastes as Supplementary Fuels in Cement Kilns." It deals mainly with a more stringent particulate level, as well as with hazardous air pollutants.

The conversion of older long wet or dry process kilns to larger capacity preheater and precalciner kilns will result in fewer emissions and lower fuel consumption per tonne of cement clinker. Note again that experience has shown that NO_x from coal-burning kilns is often lower than from gas-burning kilns, although other emissions may be higher.

General approach for a national emission guideline

The Working Group for the cement and lime industry consultation was established in Canada in March 1993. A total of 15 members were involved representing the federal and several provincial/regional governments, the two major industry sectors, and environmental groups.

The National Emission Guideline states that emissions from large new natural gas- or coal-fired cement kilns (capacity >1500 tonnes/day) should not exceed 2.3 kg of NO_x per tonne of clinker production, based on a monthly average time period. The monthly averaging period was chosen to avoid issues associated with the transient nature of the hourly/daily emissions profile of most kilns. Special permitting could be done for the use of other fuels such as oil or petroleum coke, or for the use of alternative fuel mixes including landfill gas.

For an existing large cement kiln, the New Source Guideline limits stated below shall apply when a modification results in a 25 percent increase in permitted kiln capacity. For modifications associated with one-time lesser increases in permitted capacity, a program to improve performance should take advantage of cost-effective technologies to achieve feasible emission reductions. The resulting emission levels on these modified kilns do not necessarily have to meet those limits stated for new kilns. For kilns smaller than 1500 tonnes/day, or where there are opportunities to improve environmental performance and energy efficiency on kilns for which major modifications are planned, emission rates and control methods can be evaluated in close consultation with the appropriate regulatory authorities.

The reduction of clinker production, or the offset of other emissions, is also recognized as a strategic option for the industry and for regulatory bodies to deal with permitting issues on existing or new plants. Where a plant proposes to reduce

clinker production by the addition of additives such as flyash or slag, a higher NO_x limit could be considered. This would be based on the portion of clinker produced from primary raw materials, excluding the additives. This allowance would serve to increase the kg/tonne of clinker emission level for the kiln, and to recognize a reduction in overall net emissions for a given amount of cement produced.

A similar approach was introduced to credit NO_x emissions if waste heat from the stack exhaust or clinker cooler was used for any other heating processes not normally associated with kiln operation.

Emissions of other pollutants. The cement industry has previously been mostly concerned with other emissions involving particulate matter, addressed in a 1974 regulation. It had been agreed to in the consultation process that a comprehensive evaluation of other emissions would be included to the extent that would be practical. When considering the installation of process modifications to reduce emissions of NO_x, other pollutants should be minimized to avoid adverse environmental impacts, while maintaining acceptable clinker quality. The guideline has a revised particulate limit of 0.2 kg/tonne of clinker (about 90 mg/m^3) from the stack, and 0.1 kg/tonne from the clinker cooling system. New kiln systems should be designed, and raw materials selected, to minimize SO_2 and CO emissions while remaining in compliance with NO_x guidelines.

Measurement. The need for cost-effective pollution prevention, efficiency improvements, and emissions measurement has created a need for new types of integrated control and measurement systems. One of the best methods to improve environmental performance initially is to maximize efficiency through optimization in their manufacturing and energy processes. Many cement kilns now use continuous NO_x measurement at the kiln exit to feed back information to the process control to optimize combustion. Capital costs are usually small, especially if PEM/optimization software systems can be included at the time of a control system upgrade. The increased operator awareness of the plant is a benefit, and fuel savings, quality control, and maintenance planning can quickly pay back initial costs.

New cement kilns should measure NO_x and SO_2 emissions using a continuous emissions monitoring (CEM) system. If the measurement system used for process control at the kiln exit is to be used for emissions reporting, the method should show these emissions to be representative of those exiting the stack. On existing kilns that are to undergo major modifications, measurement should be done with a CEM system, or by a method of comparable effectiveness to continuous monitoring. The averaging period, the reporting requirements, and the type of emissions monitoring are at the discretion of the local permitting authorities.

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.

Centrifuges

Centrifuges are used in applications as diverse as separating sand from liquid (a mixture of oil, water, and caustic soda) in processing oil sands to milk creaming in agriculture.

Centrifugal separators are used in a wide variety of applications, some of them conventional, such as food processing, for instance dairy creamers. More recently, the design demands placed on this technology have increased with mineral

beneficiation applications, such as extracting synthetic crude from oil sands. When the prototypes of this application were first tried in 1977, the centrifuge nozzles had a life of a few hours. The material had to be changed to one that better withstood the highly abrasive sand involved. Nozzle angle also had to be varied.

Basic Working*

Frequently, mixtures of solids and liquids must be separated into their components in order to be effectively utilized. The mixtures may be of different solids or the liquid fraction may contain dissolved solids that are to be removed. Such situations occur in food processing, mineral beneficiation, and chemical conversions.

When the solids and the density difference is small and the flow volume is large, disk nozzle centrifuges are often the best means to accomplish the purification. The separation, which takes place within the rotor of a disk nozzle centrifuge, is effected by the G force, the "rising rate" of the liquid (related to the feed flow) and the separation area provided by a set of conical, close-spaced disks as well as the process factors of fluid viscosity, particle size, shape, and density. In addition, the design of the equipment must allow for the quantity of solids to be handled, the flow characteristics of the slurry, and other practical engineering considerations.

Whereas disk nozzle centrifuges have been in use for concentration purposes for a long time, they are now employed as purifiers; in which instance a large flow of "upflowing" liquid greatly enhances the purity of the products.

Using the elutriating stream concept, we can deduce the beneficial action of displacement washing versus dilution washing. The improved flow pattern enhances the "classification" of particles. Three examples are presented that show typical processes.

Disk nozzle centrifuges have been used for over 60 years for the concentration of fine solids in a stream of slurry feed. Such centrifuges are now in common use around the world for handling food products, chemicals, minerals, biological materials, and waxes. These centrifuges are made in a variety of materials and sizes and in many different countries by various manufacturers with differing design concepts. However, the significant principles are well established.

In addition to simple sedimentation, where the objectives are to obtain a clarified effluent or thickened solids-loaded fraction or the separation of two liquid phases, it is also possible to simultaneously introduce a stream of "wash" into the centrifuge. There may be several purposes served. First, the discharging solids may exit in the "wash" fluid rather than in the mother liquid, or upflow action of the wash stream may flush out a smaller size solids fraction from the larger size solids fraction. In the first case we have purification by washing (solubles removal) and in the second case we have purification by classification (slower-settling solids removal).

Figure C-19 is a photo of an intermediate size disk nozzle centrifuge. This machine is fitted to operate under elevated temperature and pressure conditions and for the purification of terephthalic acid crystals. Note the electric motor, the overhead V-belts, and the flexibly mounted bearing assembly. These power the pendulum-suspended rotor, which has somewhat of a double cone shape with the nozzles located at the largest periphery. The housing material is Hastelloy for extra corrosion resistance and the rotor is similarly special to withstand the severe mechanical and chemical conditions.

---

* Source: Dorr-Oliver Inc., USA. Adapted with permission.

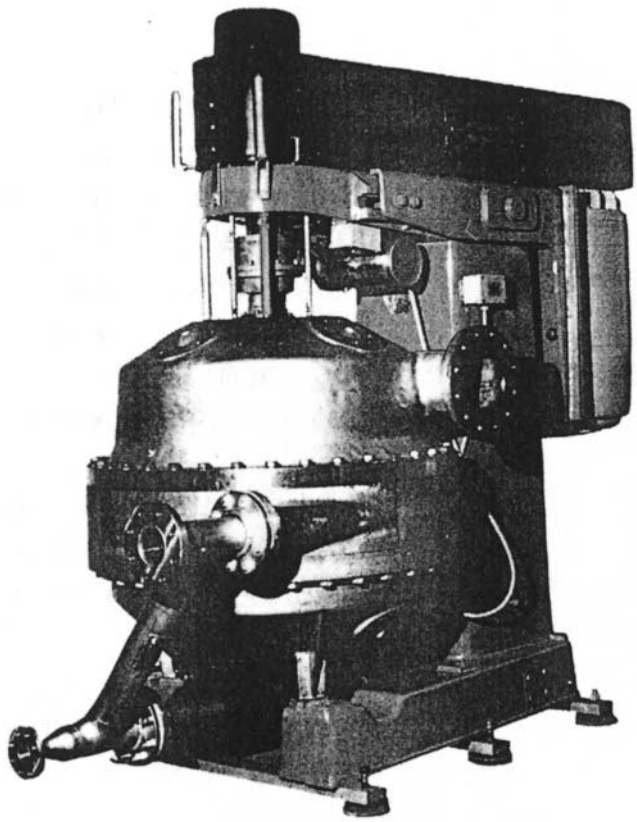


FIG. C-19 Photo of Merco® PCH-30 centrifuge. (Source: Dorr-Oliver Inc.)

Figure C-20 is a cutaway view of a disk nozzle centrifuge that shows the flow pattern. It is easy to follow the path of the feed slurry as it flows continuously down into a central rotating feed distributor and laterally into the main separating chamber. Here the high sedimenting force (of perhaps 5000 g) acts to draw the heavier solids outward where they discharge from the rotor through backwardly reacting nozzles. This slurry is then gathered in a collecting volute and recycles (by means of its velocity head) back to a reinjection port in the bottom of the stationary housing and jets back into the rotor hub where it is reaccelerated.

A major portion of the underflow can be drawn off through a valve located appropriately in the return loop. Meanwhile, the surplus flow (the feed minus the draw-off) moves inwardly through the separating disks, where fine solids are removed, and it discharges from the top of the rotor as clarified overflow. The method of feeding into the disk stack through a set of vertically punched holes and the arrangement of spaces on the disks are well known. They are sized and located in a specific fashion appropriate to the application. Similarly the recycled flow is directed through special tubes back toward the nozzle region.

Figure C-21 is a cutaway view of the centrifuge with a special wash inlet system inserted at the bottom of the housing. This system makes it possible to inject large volumes of wash at the interior of the rotor where the flow must travel inwardly and countercurrently to the outward motion of the solids. This action can accomplish a great increase in the washing capability of a single stage or it can significantly enhance the sharpness of the separation between two classes of solids.

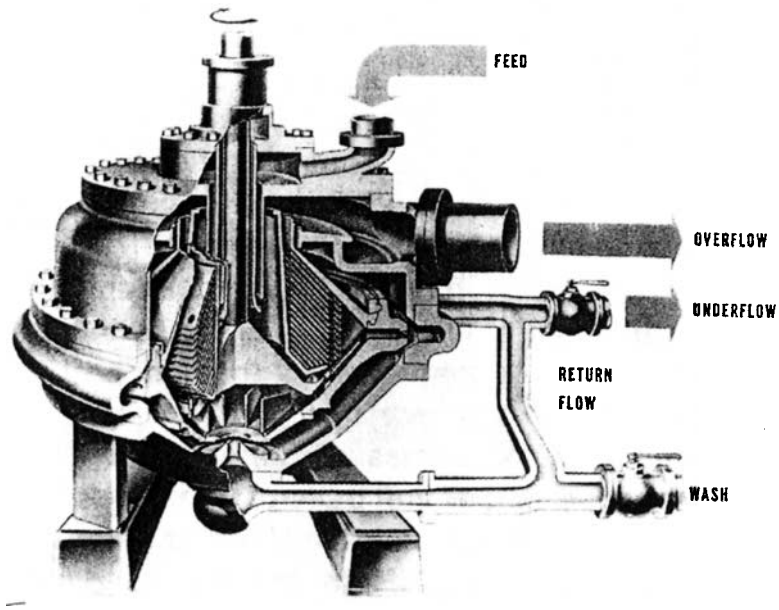


FIG. C-20 Cutaway view of a disk nozzle centrifuge and its flows. (Source: Dorr-Oliver Inc.)

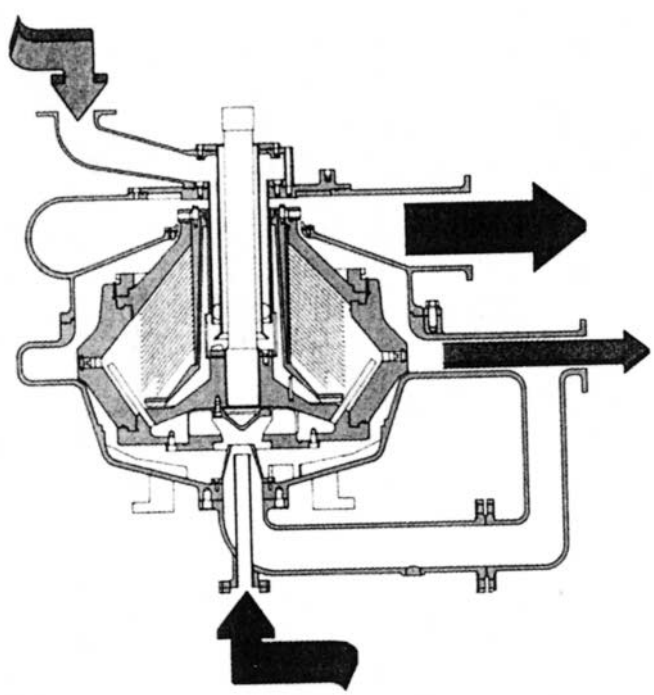


FIG. C-21 Cutaway view of Merco® centrifuge with a special wash inlet. (Source: Dorr-Oliver Inc.)

In the process of washing, the most important thing is to remove the contaminants as completely as possible. Thus, the use of large quantities of wash are generally of benefit. However, we often want to conserve the wash fluid for economic reasons. Accordingly, a balance is struck and the degree of efficiency of washing becomes important.

In the process of classification, the most important thing is to remove all of the slow settling solids but to *not* remove the other solids. Thus, the appropriate quantity of wash has to be sought by testing.

Reference and Additional Reading

1. Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Ceramics

Ceramic is a porcelainlike material that has better corrosion resistance than most metals, so ceramics are used to coat items such as turbine blades. Ceramics used to be prone to cracks due to brittleness, since ceramics lack the ductility of metals. Product improvements in this material have made major strides recently, and operating temperatures for ceramics keep rising. Various manufacturers have their own customized ceramics and their manuals and specifications should be consulted if there is a question or an overtemperature problem suspected.

Chemical Cleaning (*see also* Cleaning; *cleaning information in many other sections*)

The range of chemical cleaners (routine and also used preparatory to most overhaul and repair processes) has grown recently to bewildering proportions. Of prime consideration on the overhaul facility's requirement list is lack of carcinogen content and other increasingly enforced environmental regulations. A facility setting up cleaning facilities should consider not only existing legislation but also potential or proposed changes in environmental law before investment.

The end user's requirement for time between overhauls is greatly reduced with an effective wash system for his or her engine. If the system can be used online, the advantage is still further enhanced. Wash systems are generally designed and manufactured by firms that specialize in this work. Each turbine model's reaction to a system is different. Nozzle size and angle, fluid rate, and so forth have to vary with each model for maximum effectiveness. A manufacturer with highly successful wash systems might find that they fail entirely with a new engine model. Most OEMs generally put their own nameplates of "ownership" on their subsupplier's wash systems.

Chemical Complex; (Petro)Chemical Complex; Chemical Plant

A *petrochemical complex* (a specific kind of chemical complex) is generally a large facility that may encompass more than one company. The complex may process upstream petroleum products (primarily oil and natural gas) into complex downstream chemical and plastic products. As an illustration, a product flowchart (Fig. C-22) from the Petrochemical Company of Singapore (PCS), indicating numerous downstream companies, is included. The large number of companies and products resulting from the complex should be noted. (All acronyms used

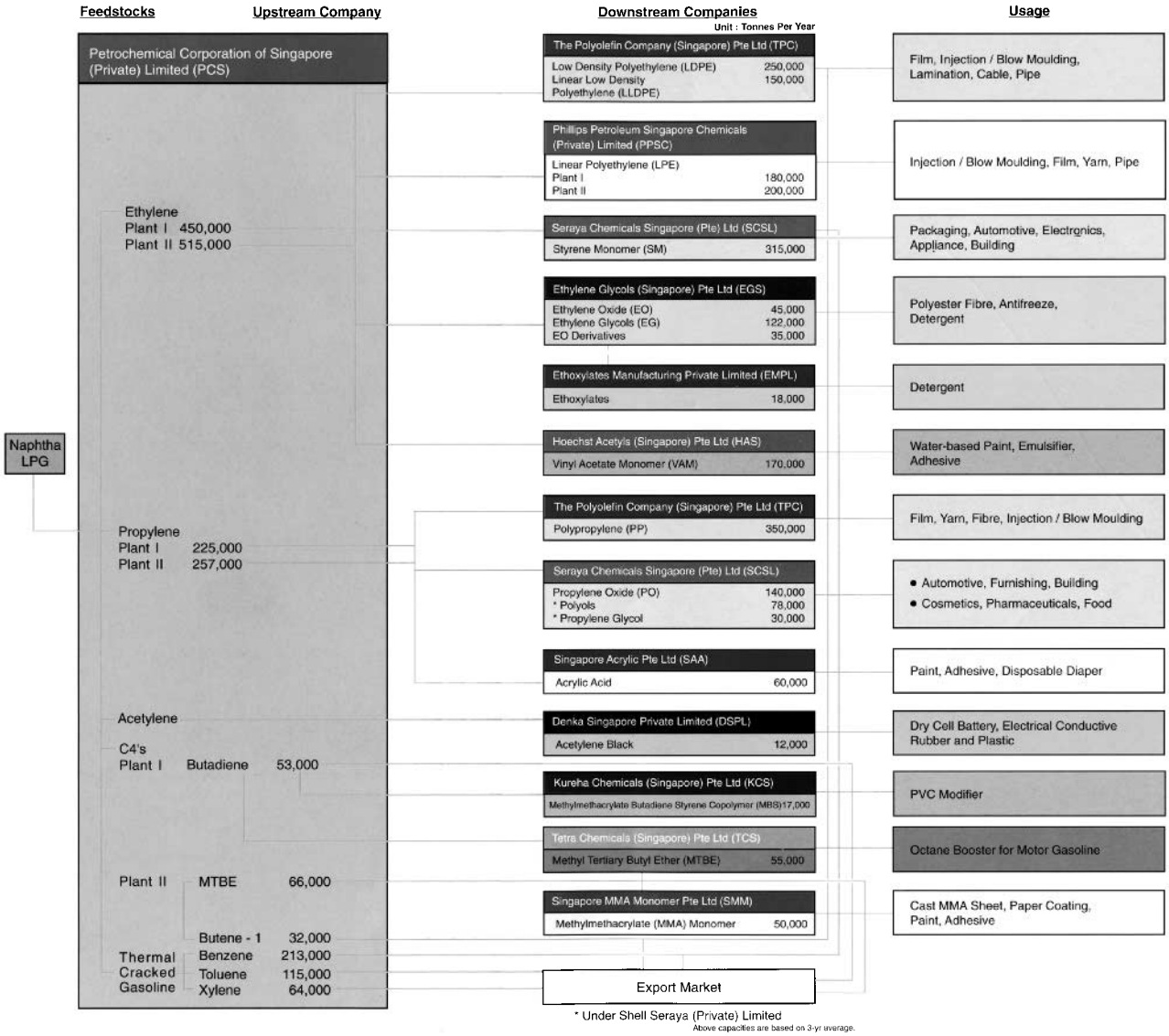


FIG. C-22 Production schematic of overall PCS complex. (Source: Petrochemical Company of Singapore.)

are described in full on the flowchart.) Note that PCS was commissioned in two phases, phase I and phase II. Most chemical complexes evolve or grow in this way, with the simpler products produced initially and more sophisticated ones in later phases. Or the later phase may be constructed for larger volumes of the initial products. A case study at this point is presented to provide insight. PCS phase I and II are described as an example. Figure C-23 is a summary of phase I of the PCS plant itself. Figure C-24 indicates the physical layout of all the various plants within the complex in both phases. Figure C-25 illustrates the ethylene plant in PCS's phase II.

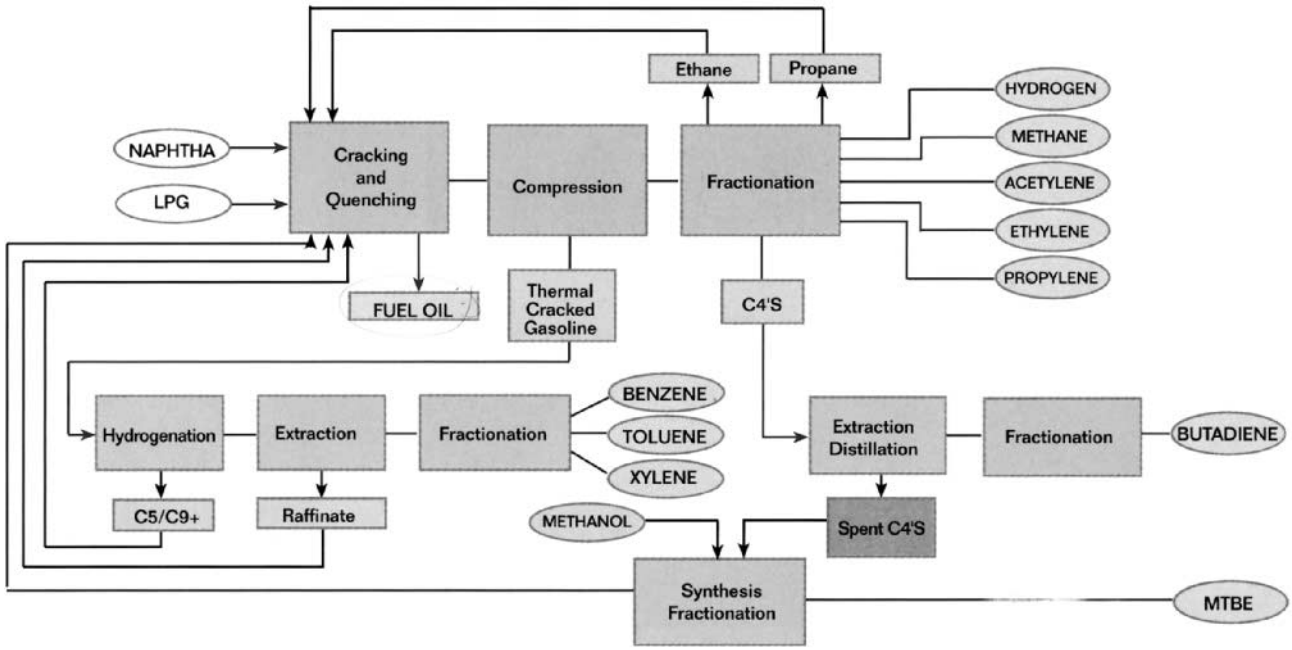


FIG. C-23 Schematic of process flow (PCS Phase I). (Source: Petrochemical Company of Singapore.)

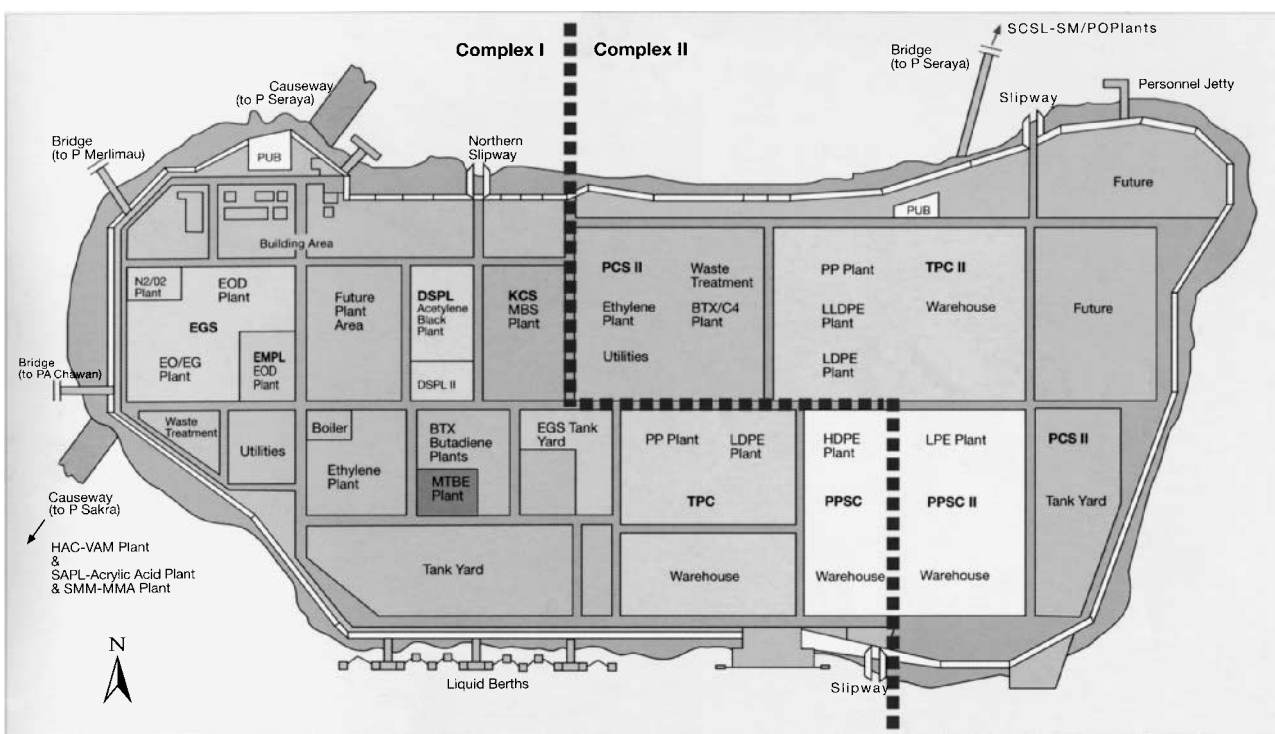


FIG. C-24 Layout of Complex I and II. (Source: Petrochemical Company of Singapore.)

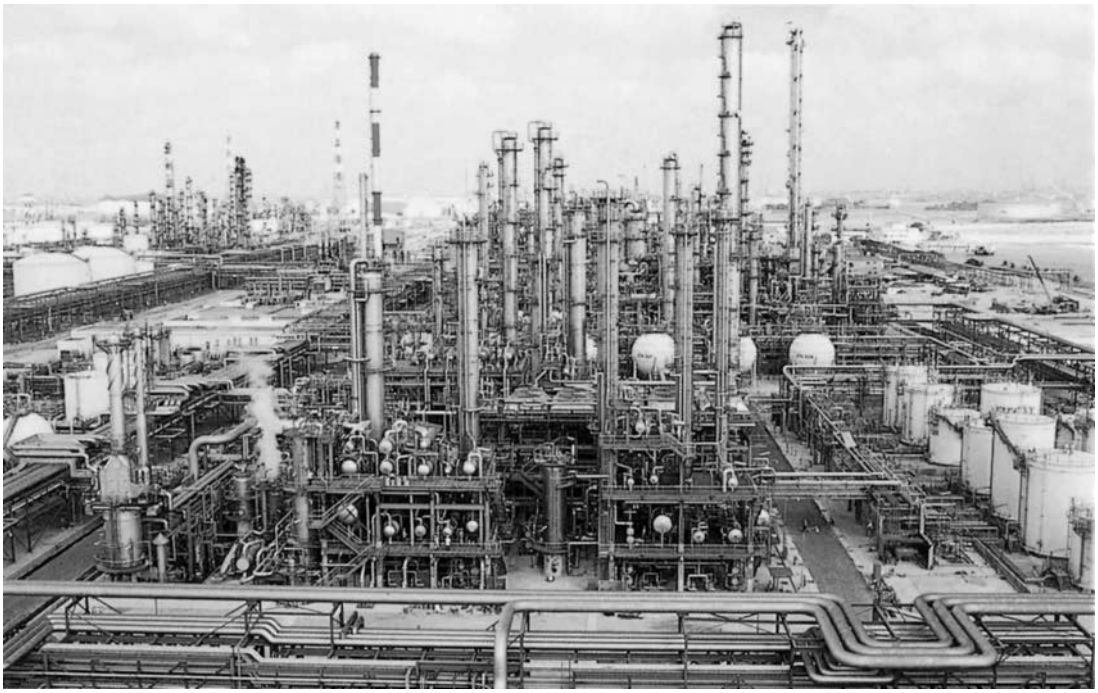


FIG. C-25 PCS II ethylene plant. (Source: Petrochemical Company of Singapore.)

"Complex II" Project Summary

The expansion of the Singapore Petrochemical Complex, known as the Complex II project, was launched on March 1, 1994. This S$3.4 billion project involves:

- The expansion at Pulau Ayer Merbau of the existing plants of PCS, TPC, PPSC, DSPL, EGS, and KCS.
- The establishment of a new downstream Styrene Monomer/Propylene Oxide (SMPO) plant by Seraya Chemicals Singapore (Private) Limited (SCSL) on nearby Pulau Seraya.

With the on-schedule completion and startup of the expansion project in April 1997, PCS's ethylene capacity is now doubled to close to 1 million tonnes/year and it is one of the largest leading ethylene producers in the region.

In the meantime, three more new downstream projects sited on the adjacent Pulau Sakra were added to the expansion. They are:

- A Vinyl Acetate Monomer (VAM) plant by Hoechst Acetyls Singapore Pte Ltd (HAS), which started up in August 1997.
- Sumitomo Chemical's joint venture (with Sumitomo Seika and Toagosei) plants to produce acrylic acid (and its derivatives) as Singapore Acrylic Pte Ltd (SAA).
- Singapore Monomer Pte Ltd (SMM), another Sumitomo Chemical joint venture (with Nippon Shokubai) producing methylmethacrylate (MMA) monomer.

Chemicals

Chemicals can be either organic or inorganic. The range of chemicals and chemical compounds in use today, even if listed by name only, would fill several volumes.

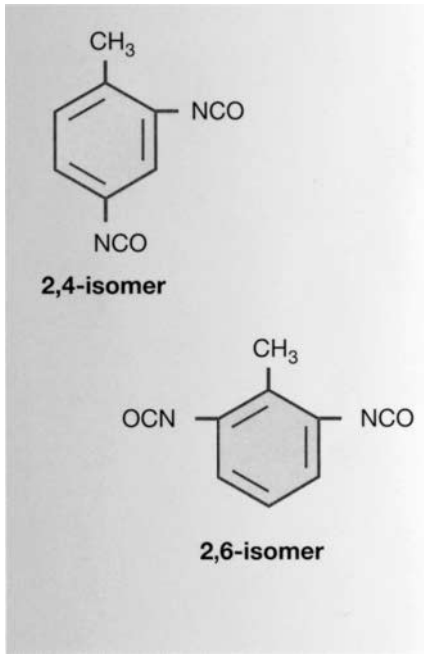


FIG. C-26 Structural formulas of TDI isomers. (Source: ARCO Chemical.)

For illustrative purposes in this handbook, a few specific illustrative examples of applications common to process engineering and some associated technology will be discussed. Toxic chemical handling technology is one such example.

Chemicals (Toxic), Handling

The illustrative example used in this case is an organic chemical that results from hydrocarbon processing. In actual fact, every toxic chemical, inorganic or organic, requires specific precautions. The example that follows indicates the extent of the complexity and caution required with handling a toxic chemical. For liability reasons, manufacturer-provided procedures should be strictly followed.

Some chemicals resulting from hydrocarbon petroleum product processing are toxic and require extreme care during all handling processes. For illustrative purposes, information on handling ARCO Chemical's toluene diisocyanate follows.*

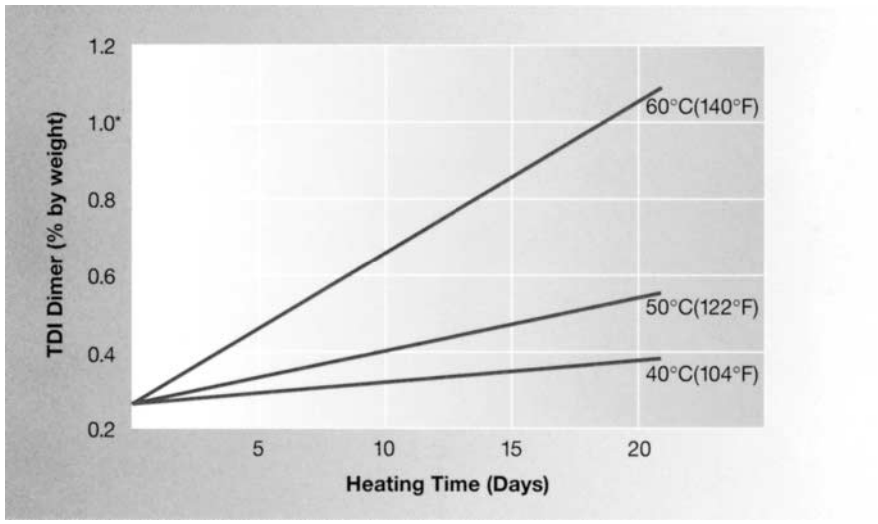
Example: Toluene Diisocyanate (TDI)

ARCO Chemical's toluene diisocyanate is referred to as TDI-80 because it is an 80:20 mixture of the 2,4- and 2,6-isomers of TDI. Structural formulas of these isomers are shown in Fig. C-26. Also see Fig. C-27.

ARCO Chemical produces TDI-80 in two forms, designated Type I and Type II. Both have the 80:20 isomer ratio, but they differ slightly in acidity and hydrolyzable chloride content.

---

* Source: ARCO Chemical, USA. Adapted with permission.



* Dimer crystals precipitate at ambient temperature when dimer concentration approaches 1%.

FIG. C-27 TDI dimer formation over time at various temperatures. (Source: ARCO Chemical.)

Type I is used in foam and nonfoam urethanes. Type II is used in nonfoam urethanes, rebonded flexible foam, and other applications.

Physical properties of TDI-80, Types I and II, are shown in Fig. C-28. Those properties marked by an [a] are ARCO Chemical specifications; other properties are those typical of commercially available TDI.

TDI has a sharp, pungent, sweetish odor. Its vapors are toxic. Certain precautions are necessary when handling or using toluene diisocyanate. Before using TDI, obtain and study ARCO Chemical's *Material Safety Data Sheet* (MSDS) and product literature. For more information, see "TDI Safety and Handling" below.

Reactivity

ARCO Chemical TDI is a clear liquid, water-white to light yellow in color. It yellows on exposure to light.

Chemical: TDI reacts readily with compounds containing active hydrogens, such as acids and alcohols. Contact with bases, such as caustic soda or tertiary amines, might cause uncontrollable polymerization and rapid evolution of heat.

Water: On contact with water, aromatic polysubstituted ureas are formed, and carbon dioxide plus heat are evolved. In time, white aromatic polyurea crystals will precipitate.

Heat: High temperatures can cause formation of dimer and discoloration of the TDI. This phenomenon is time and temperature-related (see Fig. C-27).

When the level of dimer approaches 1 percent by weight, solid dimer forms as needle-like crystals. These crystals cannot be completely filtered out because the solution is supersaturated and new crystals are formed to replace those that are removed.

Temperatures below 15°C (59°F) cause TDI to freeze. Frozen TDI is also white and crystalline. If frozen, TDI may be thawed by heating (see "Thawing TDI" below for methods and proper precautions).

Note: As can be seen from the above discussion, if white crystals are detected in TDI, they may be frozen TDI, aromatic polyurea, or dimer. For suggestions on

Molecular Weight	174.163
Assay[a], min (%)	99.7
Isomer Ratio[a] (%)	
2,4-isomer	80±1
2,6-isomer	20±1
Acidity[a], as HCl (%)	
Type I	0.002-0.004
Type II	0.008-0.010
Hydrolyzable Chlorides[a] (%)	
Type I	0.003-0.008
Type II	0.011-0.014
Chlorine[a], max (%)	0.20
Ash (ppm)	20
Color (APHA)	15-40
Specific Gravity @ 25/25°C [77/77°F]	1.22±0.01
Density (lbs per gal)	
@15.5°C [60°F]	10.23
@20°C [68°F]	10.14
@38°C [100°F]	10.02
@60°C [140°F]	9.86
Viscosity (cs)	
@50°C [122°F]	1.5
@100°C [212°F]	0.8
@135°C [275°F]	0.5
Melting Point Range	
(°C)	11.5-13.5
(°F)	52.7-56.3
Freezing Point	
2,4-isomer (°C)/(°F)	15.0/59.0
2,6-isomer (°C)/(°F)	7.2/45.0
Boiling Point	
@10mm Hg (°C)/(°F)	121/250
@760mm Hg (°C)/(°F)	251[b]/484[b]
Flash Point[c], COC (°C)/(°F)	132/270
Fire Point, COC (°C)/(°F)	142/288
Latent Heat of Evaporation (Btu/lb)	
@120°C [248°F]	131
@180°C [356°F]	121
Vapor Density, air = 1	6
Vapor Pressure, approx. (mm Hg)	
@20°C [68°F]	0.01
@120°C [248°F]	11
@130°C [266°F]	16

[a] ARCO Chemical specification

[b] Decomposes

[c] The flammability properties of this material (or any other material) are not intended to reflect the fire hazards presented by any resultant cellular or foamed plastic product.

FIG. C-28 Physical properties of TDI-80 produced at the Lake Charles, La., plant. (Source: ARCO Chemical.)



FIG. C-29 A drum for TDI containment. (Source: ARCO Chemical.)

dealing with such situations, see "What to Do In Case of . . ." below or call the manufacturer.

TDI Shipments

ARCO Chemical TDI may be obtained in tank cars, tank trucks, cylinders, or drums from this plant or various worldwide distribution centers and terminals. For export, ARCO Chemical has the capacity to ship TDI in bulk and full container lots of drums via ocean vessels.

Cylinders: In the United States, ARCO Chemical provides TDI in carbon steel cylinders that contain 230 U.S. gal and are used at 20–30 psig. Some cylinders with a capacity of 263 U.S. gal and a pressure rating of 275 psig are also available. Intended to be moved with a forklift, the cylinders have two-way-entry metal skids.

Tank Cars: TDI is most frequently shipped in 20,000-gal cars, although other sizes are available upon request. All cars are insulated and have exterior heating coils. All cars are padded with nitrogen.

Specific arrival temperatures with tank car deliveries cannot be guaranteed.

Tank Trucks: TDI is shipped in 4000- to 5000-gal trucks. Shipment weights range from 40,000 to 50,000 lb, depending on the point of origin and road weight regulations.

TDI trucks are equipped for top-unloading only and have compressors and air driers to maintain product integrity. Pumps can be made available upon request. Tanks are constructed of stainless steel; all are insulated and have exterior heating coils.

Drums: TDI is available in 55-gal nonreturnable drums, made of 18-gauge steel (minimum), with phosphatized interiors. Drums contain 551 lb (250 kg) of TDI (see Fig. C-29).

Ocean Vessels: Large chemical companies have the capability to serve world markets with shipments of large quantities in bulk or in drums.

Unloading TDI

Toluene diisocyanate is regulated by the Department of Transportation (DOT) as a Packaging Group II Toxic. Since TDI can cause serious injury to the lungs, eyes, and skin, all persons near the unloading site must wear protective clothing and equipment. They must observe the safe handling procedures and practices



FIG. C-30 A TDI tank car. (Source: ARCO Chemical.)

prescribed in ARCO Chemical's *Material Safety Data Sheet* (MSDS) and product literature. "TDI Safety and Handling" below should be carefully read by, and explained to, all employees. For additional employee training, large chemical companies offer videotapes covering handling procedures.

Customers should give careful consideration to the way that TDI will be received. Adequate facilities must be provided (see "Storage of TDI" below). Ample water should be available at the unloading site, including a shower equipped with a quick-opening deluge head and an eyewash fountain.

The site should also be equipped with an inert gas such as nitrogen or dry air for use in padding the car and purging lines.

Note: While nitrogen is preferred, all future references to "inert gas" should be taken to mean either nitrogen or dry air (–40°C/°F dew point), and all references to nitrogen should be taken to mean that dry air may also be used.

Unloading tank cars

TDI tank cars. Specific manufacturers generally operate a large fleet of dedicated TDI tank cars (see Fig. C-30). Both general-purpose and modified DOT IIIA cars are currently in service. The modified DOT IIIA cars have the following features:

- $^{9}/_{16}$ in tank shell thickness
- Full $^{1}/_{2}$ in protective headshields
- No bottom valve outlet; top unloading only
- All stainless steel fittings
- Safety valves; not safety vents
- All top fittings are mounted on a 20-bolt cover plate, inside a protective housing (bonnet)
- Hot-dipped galvanized steel safety grating
- Two dip legs (eduction pipes), either can be used for unloading (see Fig. C-31)

Most tank cars have a capacity of 20,000 gallons. Figure C-32 shows a typical arrangement of the fittings found under the bonnet on the top of the tank car. In

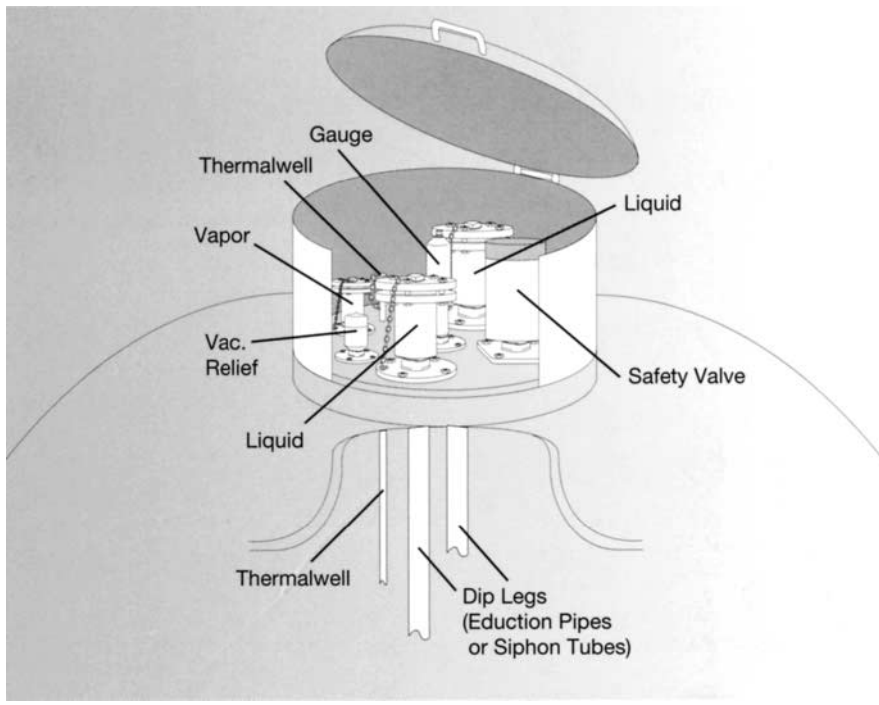


FIG. C-31 Tank car dip legs. (Source: ARCO Chemical.)

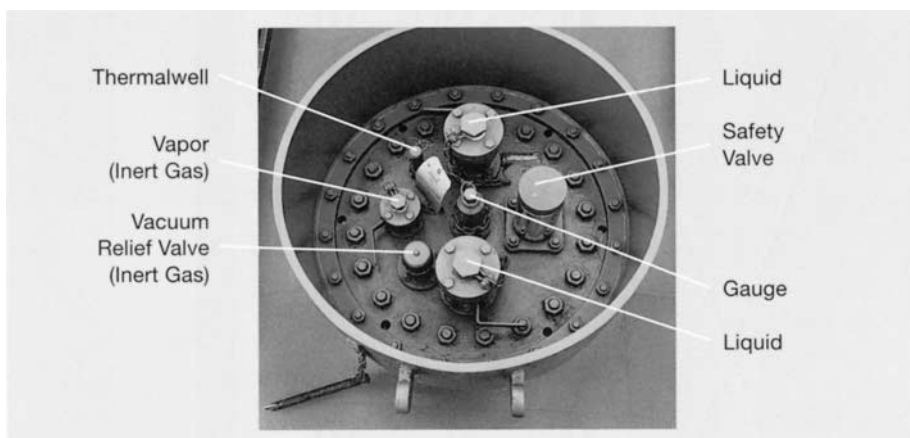


FIG. C-32 Top unloading connections (for modified DOT IIIA cars). (Source: ARCO Chemical.)

addition to these bonnet fittings, every car, regardless of type, has a manway and safety valve. Every car has a thermalwell under the bonnet, which is used in taking the temperature of the car's contents.

All chemical cars made by this information source are designed for top unloading through either eduction pipe. (See Fig. C-32 for typical connections.)

TDI cars are insulated to prevent freezing. However, in the event freezing does occur, all cars have external steam coils for thawing the TDI (see "Thawing TDI" below).

Preliminary procedures. Before tank cars or tank trucks are unloaded, all workers must put on proper protective clothing and equipment. The following three steps should then be taken.

1. *Depressurize the Car:* Open the ball valve on the 1-in inert gas inlet located on top of the car (see Fig. C-32).
2. *Take TDI Temperature:* Temperature is taken through a thermalwell, which is located between the 1-in inert gas inlet and the 2-in eduction pipe. Insert a thermocouple into the thermalwell and read the temperature. (The use of a conventional thermometer may result in an erroneous reading because the ambient temperature is usually lower than the internal TDI temperature.) *If the gas inlet valve is used for taking the temperature, a self-contained breathing apparatus must be worn as protection from TDI vapors*.

   TDI-80 is normally loaded into insulated tank cars or tank trucks at 24–30°C (75–86°F); in winter it is loaded at 38–43°C (100–110°F). Recommended unloading temperature is 21–30°C (70–86°F). If the temperature is between 17°C and 21°C (63–70°F), the TDI can be heated. If the temperature is below 17°C, it is likely that there is some freezing, and the TDI must be thawed.
3. *Sample the Car Contents:* After the car or truck has been depressurized and the TDI temperature measured, a sample should be taken for testing. While this is being done, full protective clothing and a self-contained breathing apparatus must be worn.

   For tank cars, the preferred procedure is to take a sample from the unloading line (through a customer-installed value). This avoids opening the manway cover and losing the nitrogen pad, and thus eliminates a possible source of contamination.

   If a sample is taken through this valve, first flush out 1–5 gal of TDI (for proper disposal procedure, see "Handling Spills and Leaks" below). Flushing ensures that a representative sample is being taken. This is particularly important in determining if aromatic polyurea or dimer (white precipitate) is present.

   If a sample must be taken directly from a pressurized car or truck manway, be sure it is an "all-level" sample at or near atmospheric pressure. Car hatches should be open for as little time as possible. During inclement weather, make provisions to prevent contamination of the product.

   An all-level sample is taken by using a clean, dry, amber-colored glass bottle in a weighted bottle holder. Be certain that workers are wearing proper protective gear before and during sampling.

   To be sure of getting a representative sample, the bottle holder should be lowered to the bottom and then withdrawn at such a rate that the bottle is not quite full when it reaches the surface. (This may take some practice.) Keep the sample out of direct sunlight to prevent yellowing.

   The filled sample bottle should be capped, cleaned, and plainly labeled with product lot numbers, tank car or truck number, compartment number (if more than one), date, and sampler's initials.

What to do in case of . . . *White Precipitates:* There are three causes of white precipitates in TDI: dimer (caused by excessive heat), aromatic polyurea (caused by the presence of water), or frozen TDI. If it is not obvious which of the three is present, heat the crystals. If they melt at 16–21°C (60–70°F), they are frozen TDI. If they melt at 150–160°C (302–320°F), they are dimer. If they do not melt, they are aromatic polyurea.

If the crystals are frozen TDI, the product can be thawed, remixed, and used. If the crystals are aromatic polyurea, they can be filtered out and the remainder of the TDI can be used. However, if the crystals are dimer, they cannot be completely removed (dimer reforms on filtration). The TDI should not be used because the dimer will affect urethane physical properties. It will clog lines and foam heads as well. If dimer is present, contact the manufacturer.

Discoloration: Normal TDI is water-white to pale yellow in color. A darker color means the TDI has been exposed to light or high temperature. A color something other than white to yellow means the TDI has been contaminated and should not be used. Call the manufacturer for assistance.

If the color has merely darkened, assume the cause is high temperature. (The chances of light-induced discoloration are negligible.) Since the high temperature may also cause dimer formation, the TDI should be tested. Simply cool a sample to room temperature. If white crystals precipitate, dimer is present and the TDI should not be used. If no white crystals are present, the TDI may be used. The discoloration will not affect physical properties or foam color.

General unloading regulations and suggestions. Department of Transportation regulations for unloading tank cars are given in Section 174.67 of Title 49, *Code of Federal Regulations, Hazardous Materials Regulations*. The regulations require that all persons responsible for tank car unloading should be familiar with these regulations and that all applicable requirements should be observed.

Below are some of the pertinent federal requirements. Following several of them are related suggestions and recommendations, which this information source believes are also necessary or important to follow, even though they may not be part of the regulations. These are printed in italic type. The most important recommendation that this chemical manufacturer makes is that workers be familiar with the health and safety aspects of TDI, and that they use the proper protective equipment when contact with this product is possible.

1. Unloading operations must be performed only by reliable persons properly instructed in unloading hazardous materials and made responsible for careful compliance with this part.
2. Brakes must be set and wheels blocked on all cars being unloaded. *Tank cars should also be protected during unloading by such means as derails or locked switches*.
3. Caution signs must be so placed on the track or cars to give necessary warning to persons approaching the cars from the open end of a siding. Signs must be left up until after the cars are unloaded and disconnected from the discharge connection.

   The signs must be of metal or other comparable material, at least 12 in high by 15 in wide, and must bear the words, “STOP—Tank Car Connected” or “STOP—Men at Work.” The letters are to be white on a blue background, with the word “STOP” at least 4 in high and the others at least 2 in high.

   If the unloading area has heavy traffic, it should be roped off and passersby warned by posting “Danger—TDI” signs. The contents of tank cars should only be unloaded during daylight hours or when adequate lighting is provided.
4. Unloading connections must be securely attached to unloading pipes on the dome outlet . . . before any discharge valves are opened.

   Tank cars must be depressurized before making any unloading connections.
5. Tank cars may not be allowed to stand with unloading connections attached after unloading is completed. Throughout the entire period of unloading,

and while the car is connected to the unloading device, the car must be attended.

6. If necessary to discontinue unloading a tank car for any reason, all unloading connections must be disconnected. All valves must first be tightly closed, and the closures of all other openings securely applied.

   Before disconnecting—for any reason—all lines should be cleared of liquid material by blowing with nitrogen or dry air.

7. As soon as a tank car is completely unloaded, all valves must be made tight, the unloading connections must be removed, and all other closures made tight, except for heater coil inlet and outlet pipes, which must be left open for drainage. If it has been opened, the manway cover must be reapplied by the use of a bar or wrench, the outlet reducer and outlet valve cap replaced by the use of a wrench having a handle at least 36 in long, and outlet valve cap plug, end plug, and all other closures of openings and their protective housings must be closed by the use of a suitable tool.

TDI tank cars can only be unloaded from the top, through either of the 2-in eduction pipes (dip legs).

Figure C-33 shows how unloading is accomplished using nitrogen. This dry-atmosphere padding is necessary to prevent a reaction between the TDI and any water vapor that might be present. Under no circumstances should a combustible gas be used; it presents an explosion hazard.

All fittings should be inspected for evidence of actual or potential leaks before the tank and piping system are pressurized. An oil trap should be installed on the inert gas supply line.

Tank cars are protected by a safety valve, set to relieve at 35 or 75 psig (depending on the car). The pressure system should be designed so as not to exceed a safe working limit; a pressure of 10–20 psig is recommended.

The steps involved in positioning the car and installing the necessary safety devices must be carried out in accordance with the regulations set forth in Section 174.67, as outlined previously. Before starting to unload, follow the instructions for depressurizing the car, taking the temperature, and sampling, under Preliminary Procedures, also previously. Then:

1. Be sure the tank car manway is secured. Make sure the storage tank is adequately vented.
2. While the 1-in inlet valve is closed, remove the plug and connect the inert gas line. (See Figs. C-32 and C-33.)
3. The temperature of the unloading line should be 21–30°C (70–86°F), the proper temperature for unloading. Check the line temperature and preheat the line, if necessary, before connecting it to the 2-in unloading valve, which leads to the eduction pipe.
4. Open all valves in the unloading line.
5. Open the inert gas supply valve. The pressure on the car will be effectively established by the setting of the inert gas pressure-regulating valve. The flow of TDI can be controlled by a valve in the unloading line.

After unloading is complete (or if unloading must be interrupted):

1. Purge the unloading line with nitrogen before disconnecting. Equalize the line pressure. Close all valves in the line. Disconnect the unloading valve and cap it.

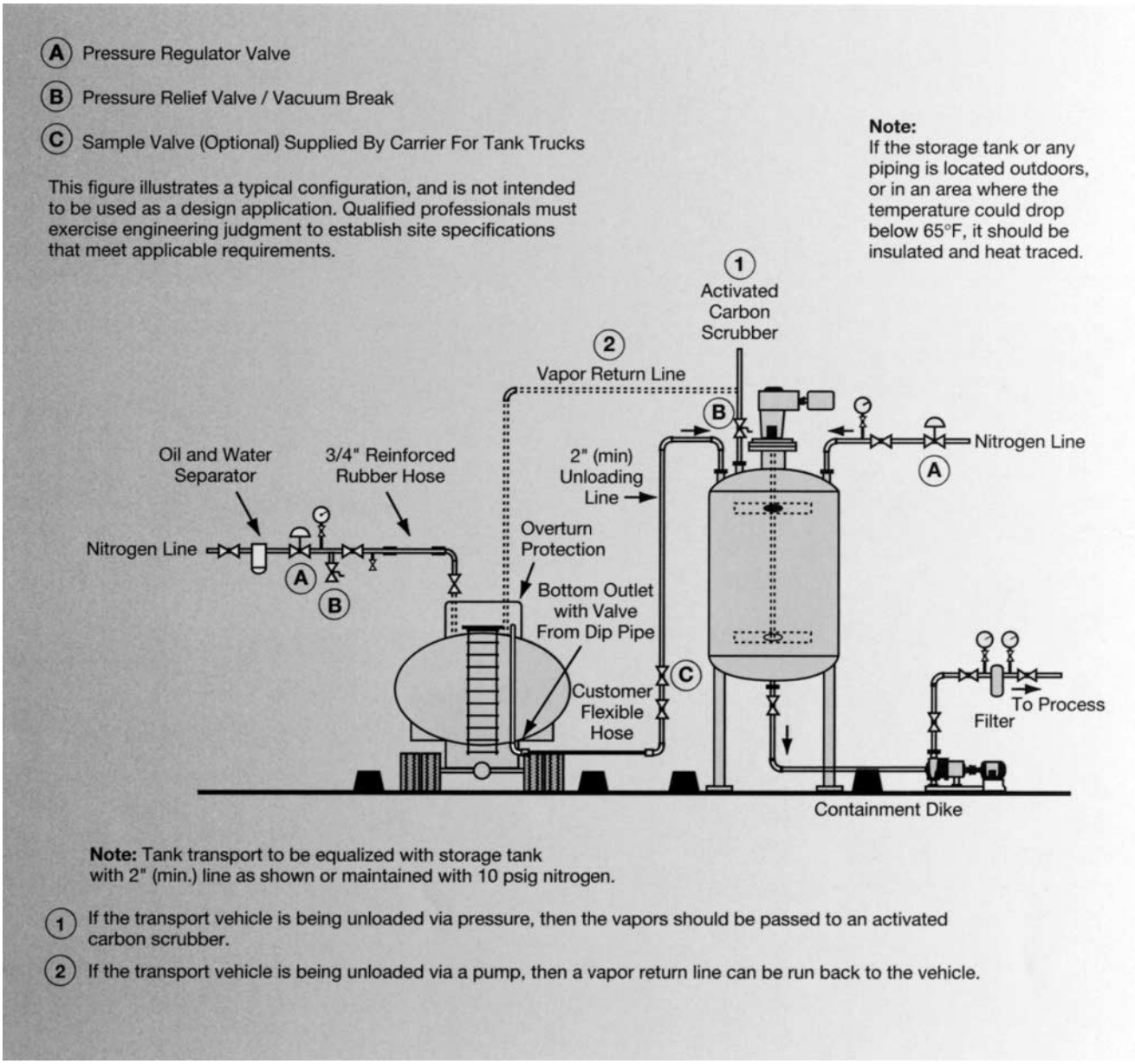


FIG. C-33 Top unloading and storage arrangement. (Source: ARCO Chemical.)

2. Disconnect the steam lines and purge the coil by blowing with nitrogen. Do not replace the caps on the steam line.
3. Repressurize the car with nitrogen to 5–10 psig.
4. Secure the dome bonnet.
5. Be sure all four placards are in place before returning the car by the prescribed routing.

Unloading tank trucks. Prior to unloading, it is the recipient's responsibility to provide competent and knowledgeable supervision, safety equipment, and a properly designed unloading area. Tank trucks are unloaded by the driver of the



FIG. C-34 Tank truck unloading. (Source: ARCO Chemical.)

vehicle, who is responsible for following the proper safety rules, as prescribed by recipient, by the manufacturer, and by government regulations. Trucks are specially equipped for unloading as shown in Figs. C-34 and C-35.

The unloading area must be large enough for easy turning and positioning of the vehicle. It should be level, to ensure complete unloading. It must be covered with an impervious material, such as concrete or steel plate (not asphalt) to prevent ground contamination in the event of a spill. The area also must be contained to prevent a spill from spreading. Safety showers and eyewash stations must be nearby.

The supervisor should make sure the unloading area is clear and that adequate facilities are ready for receiving the shipment. Before unloading begins, the supervisor must check the temperature of the TDI (and adjust it, if necessary). When the temperature is within the proper limits, it is recommended that the supervisor take a sample of the shipment.

After unloading is complete, all lines should be purged with nitrogen. The tank truck should then be padded with nitrogen (3–5 psig).

Unloading TDI cylinders

The cylinders are equipped with the following:

- Primary liquid dip tube fitted with a $1^1/_4$-in Stratoflex fitting liquid dip tube fitted with a $^1/_2$-in Stratoflex fitting
- $^3/_8$-in vent valve with a bleed-down cap assembly
- $^3/_8$-in nitrogen valve with a snaptight fitting and check valve
- Level gauge reading 5–95 percent volume
- 150 psig pressure safety valve

Note: Fitting types and sizes may vary.

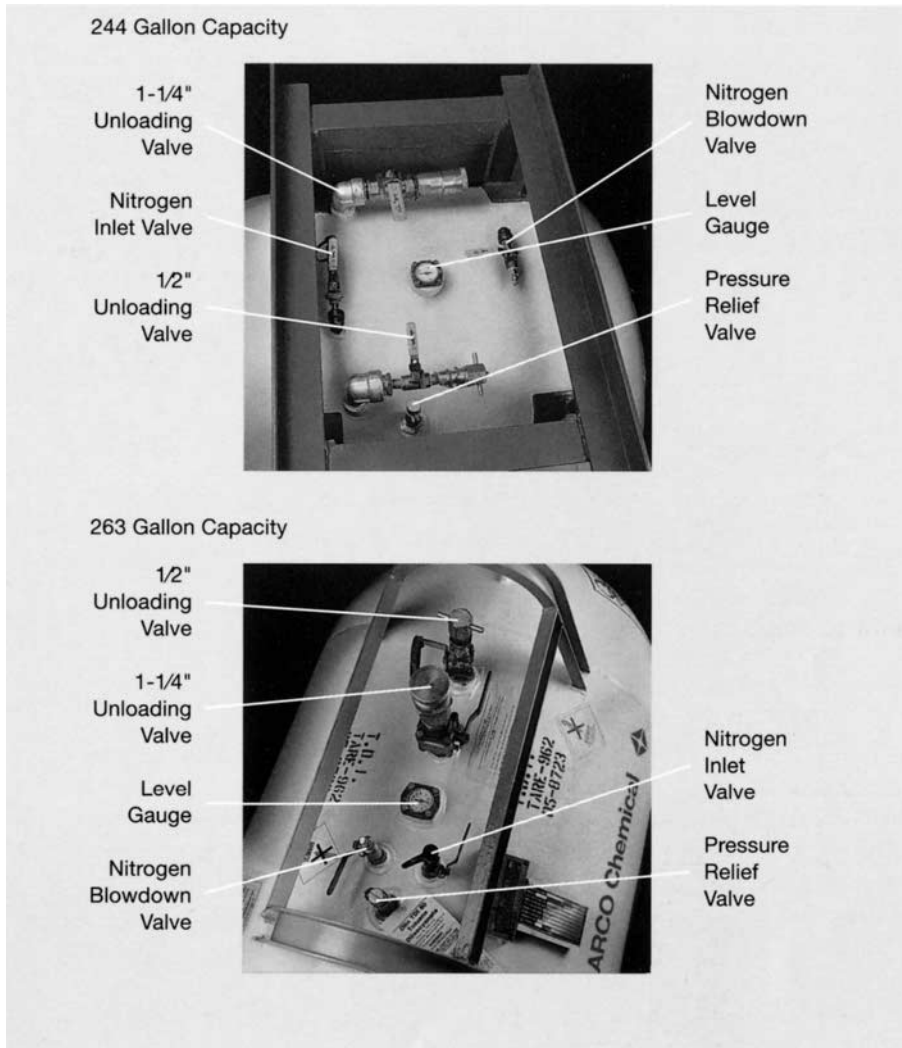


FIG. C-35 Top of cylinders. (Source: ARCO Chemical.)

Receiving cylinders. *Leakage/Damage:* Cylinder exteriors are cleaned and inspected prior to shipping so that damage can be readily seen. Upon receipt of a cylinder, check for any external damage or leakage. As long as there is no leakage, the cylinder can be accepted. Make a note on the carrier's bills and send a copy of the bill of lading and damage report to the manufacturer. If the cylinder is leaking, call the manufacturer and follow the steps in its emergency response guide. Report any dents or damage to skids or cowling to the manufacturer.

Pressure: Cylinders should have a positive nitrogen pad pressure in the range of 5–25 psig. If no pressure is present, call the manufacturer for instructions.

Returning cylinders. *Preparing Empty Cylinders for Return:* Be sure that the dust caps are tightly screwed onto the male and female self-sealing couplings and the nitrogen inlet caps are in place when the tanks are not in use. This is essential to prevent possible contamination and vapor leaks from the connectors. Make sure that the threads and internal body of all fittings are clean.

Before cylinders are transported, reduce internal pressure to 5–25 psig. It is recommended to place a nitrogen pad of less than 25 psig on the cylinder prior to

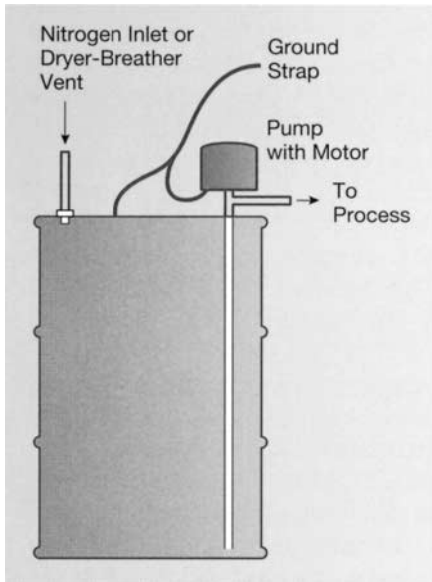


FIG. C-36 Drum unloading system. (Source: ARCO Chemical.)

the return shipment. The shipping regulations permit freight-forwarding and common carriers to charge a rate higher than normal if pressure is above 25 psig, since that places the tank in a "Compressed Gas" category.

Unloading drums

Follow all applicable safety procedures. Be sure full protective clothing is worn (see Fig. C-40) when opening the drum plug (bung), when placing or operating pumps, or when flushing out empty drums. In the event of spillage, see "Handling Spills and Leaks" below.

If the TDI is frozen, or if there is a possibility of freezing because the drums have been exposed to ambient temperatures below 17°C (63°F), then the drums should be heated to 35–43°C (95–110°F) until all TDI is liquid. Do not heat above 43°C (110°F). After the TDI is thawed, the drums should be rolled for at least 30 min to uniformly mix the 2,4- and 2,6-isomers.

During unloading, drums should be kept under a nitrogen pad to prevent contamination by water vapor. However, unloading by pressure is unsafe.

The preferred method is by pump, either manual or electric (see Fig. C-36). If the pump is electrical, be sure the drum is properly grounded. If the drum is to be unloaded by gravity, the faucets should be self-closing. Bungholes should be fitted with a dryer-breather vent device to prevent drum collapse.

Thawing TDI

Thawing TDI in tank cars

TDI is shipped in insulated tank cars. During the winter, it is loaded at temperatures between 38 and 43°C (100–110°F). Despite these precautions, there may be substantial heat loss before the car reaches its final destination. Therefore, during the winter, all incoming tank cars of TDI should be checked for freezing. The 2,4-isomer of TDI-80 freezes at 15°C (59°F), the 2,6-isomer at 7.2°C (45°F). Between these two temperatures, only the 2,4-isomer freezes. If this happens, isomer stratification takes place.

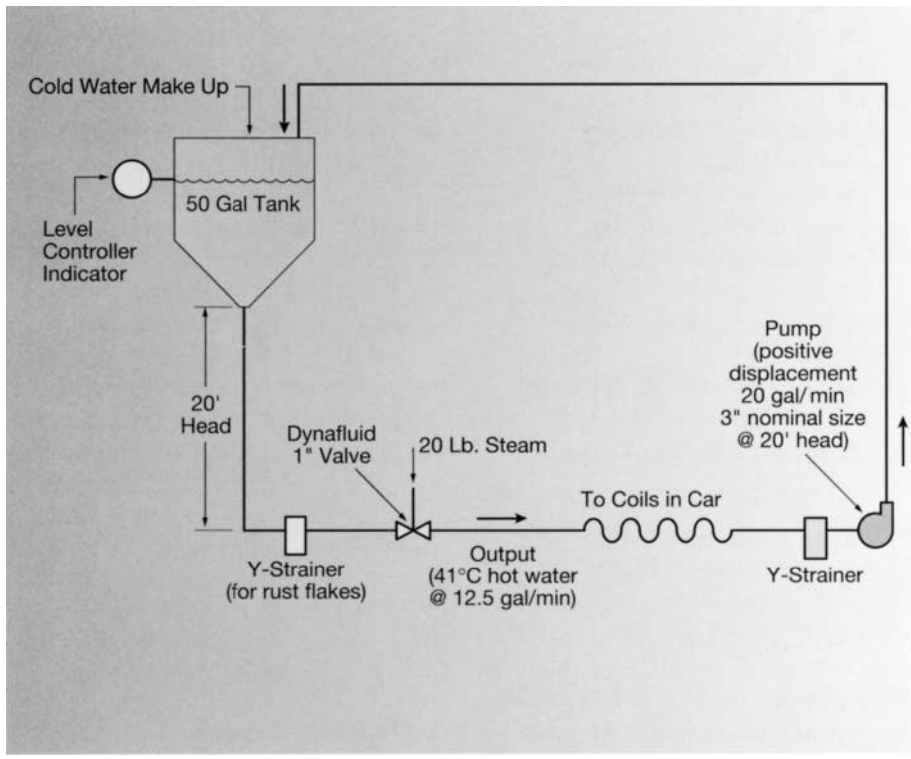


FIG. C-37 Steam/water mixing system. (Source: ARCO Chemical.)

Note: After thawing TDI, the layers remain separated. If they are not mixed, processing problems can be expected. However, if proper care is taken in thawing and remixing TDI, the quality can be maintained and no processing problems should occur.

How to determine if TDI is frozen. The way to tell if TDI is frozen is by taking its temperature while wearing proper protective equipment. Do not open the manway to inspect it visually. Temperature measurement is accurate and will detect frozen TDI, even when it is not visible.

When to heat a TDI tank car. If the TDI temperature is less than 17°C (63°F), the car should be heated before it is unloaded.

Note: If the car is not to be heated immediately, it should be repressurized to 5–10 psig with nitrogen to prevent crystals from forming as the result of contamination of the TDI with water. It should be depressurized before heating and unloading.

How to heat a TDI tank car. The TDI should be heated to 35–43°C (95–110°F) until all the frozen TDI has thawed. Never allow the TDI temperature to exceed 43°C (110°F). If TDI is overheated, dimerization may take place. (See discussion under *Heat* above and graph showing conditions for dimer formation, Fig. C-27.) If dimer forms, the TDI should not be used.

Heat Sources: The best way to thaw frozen TDI is with tempered hot water, thermostatically controlled to 41°C (106°F). Hot water is less likely to cause dimerization than steam. If tempered hot water is not available, an alternate source of heat is 20-lb steam, mixed with cold water. A steam/water mixing system similar to the one shown in Fig. C-37 can be used to obtain the desired temperature.



FIG. C-38 Steam hose connections. (Source: ARCO Chemical.)

Plants that have only steam available should avoid pressures above 20 lb. High-pressure steam, if not watched very carefully, will rapidly overheat the TDI. Even at lower temperatures, careful monitoring must take place.

Heat Source Connections: Tank cars were designed by different tank car manufacturers and put into service at different times. Therefore, cars must be carefully examined to determine the size and location of the external coil inlets and outlets.

In general, the inlet is on one side of the car, away from the handbrake (Fig. C-38). Some cars have two inlet valves. On these cars, the one farthest away from the handbrake side is for the left-side coils; the one nearest the handbrake side is for the right-side coils.

After TDI is thawed. After the TDI has been heated to 35–43°C (95–110°F), it must be completely mixed to eliminate isomer separation. Unload the entire contents into a bulk storage tank and circulate for 2–3 h before use.

Thawing TDI in cylinders

TDI will freeze at temperatures below 60°F. It is therefore imperative that during winter, cylinders be stored in a temperature-controlled environment. Recommended storage temperature is 70°F.

However, if the product does freeze, each cylinder must be placed in a heated room. The material should be completely thawed prior to use.

During this time period, daily movement of the cylinder will be necessary to allow the TDI isomers to thoroughly mix inside the cylinder. Short, jerking motions while moving with a forklift will provide sufficient agitation. To avoid product damage, never apply steam or an open-air flame to the exterior of the cylinder. A nitrogen pad of 20–25 psig should be maintained while the cylinder is being stored or heated.

Storage of TDI

TDI may be stored indoors or outdoors.

If TDI is stored indoors, the building should have sprinklers, good ventilation, and adequate heat to maintain storage temperature of 21°C (70°F). Constant monitoring of TDI temperature is required. If TDI is stored outdoors, or if indoor



FIG. C-39 Typical TDI storage tank. (Source: ARCO Chemical.)

temperature may drop below 21°C, provisions must be made for warming and thawing the TDI. These include adequate tank and line insulation, external heating coils or jackets, and steam-traced or electrically heated lines.

If thawing is necessary, never heat the TDI above 43°C (110°F). Prolonged overheating will cause dimer formation (see *Heat* above). After thawing, mix the TDI to eliminate isomer separation. Use a tank agitator or a circulating pump.

Whether indoors or outdoors, bulk storage tanks should be blanketed with nitrogen. Without this dry atmosphere, water vapor will react with the TDI to form solid aromatic polyurea, which can plug lines and foam machine heads.

A pneumatic bubbler gauge[1] that operates with nitrogen is recommended. This gauge measures the pressure required to displace TDI from a vertical tube in the tank.

Storage tank design

Vertical, cylindrical steel tanks (Fig. C-39) are normally preferred for storing TDI, although limited indoor headroom may dictate the use of horizontal tanks.

Storage tanks may be field-erected on a concrete foundation, and there is no practical limitation to size. Recommended capacity is 30,000 gal for tank car deliveries and 6–8000 gal for tank trucks. In other words, capacity should be sufficient to accept the entire contents of a tank car or truck, even when half-filled. The storage tank vent should be routed to an approved emission control system.

Materials of construction

TDI tanks can be made from carbon steel (ASTM A 285 Grade C) or from stainless steel (Type 304 or 316). API Code 650 specifies $^1/_4$-in steel for the bottom and $^3/_{16}$-in for the shell and roof.

Stainless steel tanks require no lining and are recommended. Carbon steel may also be used provided it is rust-free, sandblasted, and "pickled" with an initial TDI charge prior to use, or has a baked phenolic lining. Recommended are: Heresite P 403,[2] Lithcote LC 73,[3] Amercote 75,[4] or Plascite 3,070.[5] The inside surface should

[1] Petrometer Corp. or Varec Div., Emerson Electric Co.

[2] Heresite-Saekaphen, Inc.

[3] Lithcote Company.

[4] Amercon Corporation.

[5] Wisconsin Protective Coatings.

be sandblasted to a commercial finish and cleaned prior to the application of the lining.

Hose and piping to receive TDI

From Tank Cars: TDI is discharged by nitrogen pressure supplied by the customer through flexible hose into piping to the storage tank. Both the hose and the piping are provided by the customer. The hose should be a polypropylene-lined flexible hose.

When unloading, it is also necessary to repressurize the car. Use a $^3/_4$-in reinforced rubber hose attached to the 1-in inert gas inlet fitting.

From Tank Trucks: TDI is usually discharged from a built-in compressor or pump on the truck, through flexible polypropylene-lined hose provided by the trucker, into piping supplied by the customer. The length of the hose is specified by the customer with the first order. The piping should be Schedule 316 stainless steel. An oil-and-water separator and pressure regulator are also suggested as an assembly in the pressure line off the compressor.

Auxiliary equipment

Valves: Ball valves should be stainless steel with nonvirgin TFE seals. Plug valves and gate valves are not acceptable. Valves may be threaded or they may be flanged (150-lb ASA or MSS).

Liquid Filter and Pressure Gauges: A filter should be placed in the piping between the tank car or tank truck and the storage tank. A cartridge with a 20- or 30-micron glass fiber element is recommended.

Pressure gauges should be installed on either side of the filter to measure the drop. This will indicate when the filter must be cleaned or replaced.

Sampling Valves: If delivery is by tank car, an in-line sampling valve is recommended.

Pumps: Sealless magnetic drive pumps are recommended for TDI transfer.

TDI Safety and Handling

The following contains information as of December 1997. The health and safety information is partial. For complete, up-to-date information, obtain and read the current *Material Safety Data Sheet* (MSDS). (To order an MSDS, call the chemical company's nearest office.)

TDI is a toxic and highly reactive compound. It should be kept in closed, isolated systems and transferred with care. However, TDI is not a difficult material to handle. If proper procedures are followed, there is relatively little chance of danger.

The sections below briefly discuss some possible hazards and describe what to do in an emergency. Plant personnel should be thoroughly familiar with these procedures.

Reactivity hazards

TDI is a stable compound with a relatively high flash point. However, it will react with water, acids, bases, and other organic and inorganic compounds. TDI is also affected by heat and, like any organic compound, will burn.

Water: When TDI comes in contact with water, aromatic polyurea is formed, heat is generated, and carbon dioxide is evolved. Pressure buildup from the carbon dioxide will occur. This pressure could rupture a storage vessel. To help prevent reactions with water, the TDI should be kept under a nitrogen pad.

Chemical: Contact between TDI and acids should be avoided. Contact with bases, such as caustic soda and primary and secondary amines, might produce a violent

reaction. The heat given off causes pressure buildup and risk of rupture of the storage vessel. Contact with tertiary amines (commonly used as urethane catalysts) may cause uncontrollable polymerization, with a similar result. High temperatures may also cause dimerization.

TDI should be kept away from certain rubber and plastics. These materials will rapidly become embrittled; cracks may develop and their strength may be weakened.

Fire hazards

TDI has a flash point of 132°C (270°F) and therefore does not constitute a severe fire hazard. However, TDI is an organic material and will burn when exposed to fire. In addition, the flash point of TDI does not reflect the hazards presented by any cellular or foam plastic product that contains TDI.

Health hazards

TDI is highly toxic through inhalation and if inhaled in significant quantities can produce serious health effects. TDI is an animal carcinogen and is considered to be a possible human carcinogen. TDI has a characteristically pungent odor. However, TDI is considered to have poor warning properties; if you can smell it, the concentration of TDI would be in excess of the occupational exposure limit of 0.005 ppm (0.04 mg/m^3) as an 8-h time-weighted average.

Inhalation: Repeated overexposure and/or a high one-time accidental exposure to TDI may cause allergic lung sensitization similar to asthma. Symptoms may include wheezing, choking, tightness in the chest, and shortness of breath. Any individual exposed to TDI above the occupational exposure limit may develop these symptoms; however, for sensitized persons, these symptoms may occur at or below the occupational exposure limit. Repeated overexposure to TDI may also produce a cumulative decrease in lung function.

Dermal and Oral Exposure: The liquid and vapor of TDI can cause moderate to severe irritation to the eyes, skin, and mucous membranes. If not rinsed off immediately (within 5 min), burns to the eyes and skin may occur with the possibility of producing visual impairment. While the oral toxicity of TDI is low, ingestion of TDI can result in severe irritation to the gastrointestinal tract and produce nausea and vomiting.

Protective clothing

Because of the health hazards associated with TDI, full protective clothing and equipment (see Fig. C-40) must be worn whenever there is a possibility of contact. Such occasions include, but are not limited to:

- Opening tank car hatches, truck manway covers or drum plugs
- Connecting and disconnecting hoses and pipes
- Placing and operating pumps
- Breaking TDI piping, including piping previously decontaminated
- Flushing (cleaning) TDI drums
- Pouring foams, in operations where ventilation may not be adequate

Where liquid TDI spills can occur, butyl rubber clothing should be worn. If any article of clothing becomes contaminated, it should be removed immediately and discarded promptly.

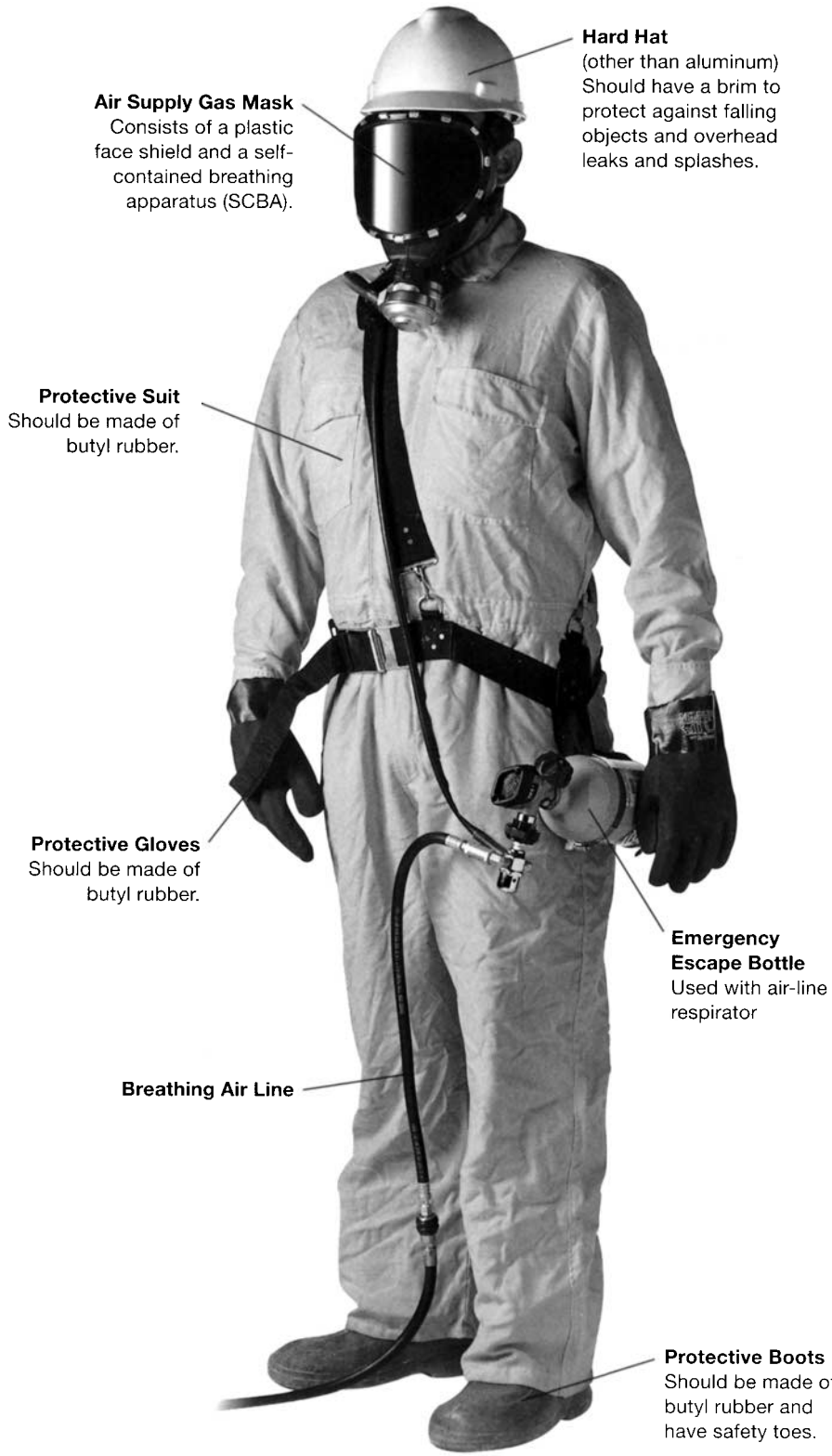


FIG. C-40 Protective clothing and equipment. (Source: ARCO Chemical.)

The odor warning of TDI is insufficient to be used as a method for detecting the presence of hazardous concentrations. Whenever there is a chance that airborne levels of TDI vapors could exceed the recommended Threshold Limit Value (0.005 ppm as an 8-h time-weighted average or 0.02 ppm as a ceiling value), a NIOSH/MSHA positive-pressure, supplied-air respirator should be worn. When issuing respirators to employees, follow all OSHA respirator requirements (29 *Code of Federal Regulations* 1910.134).

The equipment necessary to properly protect any individual who may come into contact with liquid TDI is shown in Fig. C-40.

Emergency Actions

The following section contains basic information on what to do in the event of an accident.

In addition, the Chemical Manufacturers Association (CMA) has established CHEMTREC to give advice on spill, leak or fire emergencies involving transportation or transport equipment. The current CHEMTREC number for the United States and Canada is 800-424-9300.

In the District of Columbia or from outside the U.S., call 703-527-3887.

Note: If the spill is greater than 100 lb, U.S. federal law requires it to be reported to the National Response Center (NRC). The number is 800-424-8802.

First aid

If there is known contact with TID, take the following steps:

Eye Contact: Flush the eyes with clean, lukewarm water; then periodically flush for 20–30 min. Prompt medical attention should be sought.

Skin Contact: Immediately flush thoroughly with water for 15 min. Seek medical attention if ill effect or irritation develops.

Inhalation: Immediately move victim to fresh air. Symptoms of exposure to TDI vapors include: tightness in the chest, watering eyes, dry throat, nausea, dizziness, and headaches. The onset of symptoms may be delayed, so a doctor should monitor exposed personnel.

Handling spills and leaks

Wear a NIOSH/MSHA-approved, positive-pressure, supplied-air respirator. Follow OSHA regulations for respirator use (see 29 *Code of Federal Regulations* 1910.134). Wear recommended personal protective equipment: clothing, gloves, and boots made of butyl rubber.

Spill and leak cleanup:

1. Stop the source of spill. Stop the spread of spill by surrounding it with dry noncombustible absorbent.
2. Apply additional dry noncombustible absorbent to the spill. Add approximately 10 parts decontamination solution to every one part spilled TDI.

Suggested Formulation for Decontamination Solution

	% by Weight
Water	75
Nonionic Surfactant[a]	20
n-propanol	5

[a] e.g., Poly-Tergent® SL-62 (Olin).

3. Sweep up material and place in proper DOT-approved container. Use more decontamination solution to clean remaining surfaces and also place this residue in container.
4. Loosely apply lid. Do not seal for 48 h, since gas generation may occur during neutralization. Isolate container in a well-ventilated place.
5. Discard all contaminated clothing. Decontaminate personnel and equipment using approved procedures.

Decontamination of empty containers:

1. Spray or pour 1–5 gal of decontamination solution into the container. Ensure that the walls are triple rinsed.
2. Leave container standing unsealed for a minimum of 48 h to allow for a complete neutralization of TDI.

Disposal:

1. Care should be taken to prevent environmental contamination from the use of this material.
2. Dispose of contaminated product, empty containers and materials used in cleaning up leaks, spills, or containers in a manner approved for this material.
3. The user of this material has the responsibility to dispose of unused materials, residues, and containers in compliance with all relevant federal, state, and local laws and regulations regarding treatment, storage, and disposal for hazardous and nonhazardous wastes.
4. Ensure that drums are labeled with correct hazardous waste code. Waste code U223.

Chillers; Crystallizers; Chemical Separation Method; Alternative to Distillation/Fractional Distillation*

Crystallization: An Alternative to Distillation

Many organic mixtures may be separated by cooling crystallization. In simple terms, cooling crystallization means that a mixture of organic chemicals is partially crystallized by reduction in temperature, without removal of any of the components by evaporation. An example of a crystallizer is illustrated in Fig. C-41.

Crystallization is a one-way process: the heat is removed, crystals are formed, and the mixture of crystals and solids are then separated. Many crystallizations take place at near ambient temperature so there is little heatup or cooldown required to get the right conditions for the separation to start.

Distillation, on the other hand, is a refluxing operation, where products are repeatedly evaporated and recondensed. Most distillations take place at elevated temperatures, which means that the materials being processed must be heated up and cooled back down again, usually with energy losses both ways. Also many distillations are run under vacuum to achieve better separation, which is energy intensive.

The latent heat of fusion in crystallization is generally much lower than the latent heat of vaporization. Since the latent heat must be removed only once, instead of many times as in distillation, the energy requirements are drastically lower for crystallization.

In the great majority of crystallizations, the crystals that form are 100 percent pure material, as opposed to something only slightly richer than the feed material

* Source: Armstrong Engineering Associates, USA. Adapted with permission.

3. Sweep up material and place in proper DOT-approved container. Use more decontamination solution to clean remaining surfaces and also place this residue in container.
4. Loosely apply lid. Do not seal for 48 h, since gas generation may occur during neutralization. Isolate container in a well-ventilated place.
5. Discard all contaminated clothing. Decontaminate personnel and equipment using approved procedures.

Decontamination of empty containers:

1. Spray or pour 1–5 gal of decontamination solution into the container. Ensure that the walls are triple rinsed.
2. Leave container standing unsealed for a minimum of 48 h to allow for a complete neutralization of TDI.

Disposal:

1. Care should be taken to prevent environmental contamination from the use of this material.
2. Dispose of contaminated product, empty containers and materials used in cleaning up leaks, spills, or containers in a manner approved for this material.
3. The user of this material has the responsibility to dispose of unused materials, residues, and containers in compliance with all relevant federal, state, and local laws and regulations regarding treatment, storage, and disposal for hazardous and nonhazardous wastes.
4. Ensure that drums are labeled with correct hazardous waste code. Waste code U223.

Chillers; Crystallizers; Chemical Separation Method; Alternative to Distillation/Fractional Distillation*

Crystallization: An Alternative to Distillation

Many organic mixtures may be separated by cooling crystallization. In simple terms, cooling crystallization means that a mixture of organic chemicals is partially crystallized by reduction in temperature, without removal of any of the components by evaporation. An example of a crystallizer is illustrated in Fig. C-41.

Crystallization is a one-way process: the heat is removed, crystals are formed, and the mixture of crystals and solids are then separated. Many crystallizations take place at near ambient temperature so there is little heatup or cooldown required to get the right conditions for the separation to start.

Distillation, on the other hand, is a refluxing operation, where products are repeatedly evaporated and recondensed. Most distillations take place at elevated temperatures, which means that the materials being processed must be heated up and cooled back down again, usually with energy losses both ways. Also many distillations are run under vacuum to achieve better separation, which is energy intensive.

The latent heat of fusion in crystallization is generally much lower than the latent heat of vaporization. Since the latent heat must be removed only once, instead of many times as in distillation, the energy requirements are drastically lower for crystallization.

In the great majority of crystallizations, the crystals that form are 100 percent pure material, as opposed to something only slightly richer than the feed material

* Source: Armstrong Engineering Associates, USA. Adapted with permission.



FIG. C-41 Fatty chemical crystallizer with both brine and boiling refrigerant cooling. (Source: Armstrong Engineering Associates.)

as in distillation. With crystallization it is not necessary to repeatedly melt and refreeze to obtain high purity. The pure crystals may have some impure mother liquor on the surfaces and sometimes contained within the crystals as occlusions. However, the purity increase is extremely rapid and normally one or perhaps two crystallizations can give very high purities.

In addition to much lower energy costs as compared to distillation, crystallization has other significant benefits, such as:

- Low-temperature operation, which means low corrosion rates, and often the use of less costly alloys compared to evaporation-based separations. The low-temperature operation also means little or no product degradation, which for heat-sensitive materials may be crucial. There is no formation of tars, which represent a yield loss, a severe waste disposal problem, and usually requires additional separation equipment and energy for the tar removal in order to give the desired product color.
- Enclosed systems with little or no chance of leakage of dangerous or noxious fluids. The systems are normally simple and require few pieces of equipment and little instrumentation.
- Favorable equilibrium; often the freezing points of organic chemicals are widely spread enabling easy separation by crystallization, where separation by distillation may be extremely difficult.
- High purity; the crystals that form in a great majority of cases are 100 percent pure material. While impurities may adhere to crystal surfaces, or be included inside the crystal, recrystallization usually produces very high purities with relative



FIG. C-42 Stainless steel process side crystallizer for fatty chemicals—shell side has stainless steel for corrosive coolant in lower section. (Source: Armstrong Engineering Associates.)

ease. The normal product purity range is 95 to 99.5 percent, although higher figures are often reached. One large plant produces 99.9+ percent pure product.

- The scraped surface crystallizer makes crystallization continuous. Generally, the only reason to work with batch crystallization is very low design capacity. If design capacity is above 500,000 lb annually, the scraped surface continuous crystallizer will save time, energy, and manpower.

Many crystallizations are performed using batch cooling in stainless steel or glass-lined kettles (Fig. C-42). By and large this represents continued growth from specialty chemical to commodity, with little engineering attention paid to the crystallization part of the process.

This method offers significant advantages over batch crystallization, such as:

- Smaller equipment, which generally means less expensive installations, less floor space needed, less operator labor, and no duplication of instrumentation, piping, etc.
- Better process control, less upsets of hazardous or expensive materials, and less peak utility demand

Many continuous crystallizations are done in evaporative crystallizers based on designs typically used for inorganic chemicals. With inorganics there is usually a very flat solubility curve, which means that a change in mixture temperature produces relatively few crystals. Other continuous crystallizations are sometimes performed by cooling and partially crystallizing in shell and tube exchangers, which can foul, requiring them to be taken out of service for cleaning.



FIG. C-43 Crystallizer for very viscous medium with individual drive gear motors. (Source: Armstrong Engineering Associates.)

The scraped surface continuous crystallizers offer many advantages over these other methods of continuous crystallization, such as:

- Modular design allows for easy expansion with growth in demand.
- Simple, self-contained construction with minimum instrumentation and auxiliaries, such as: condensers, vacuum systems, etc.
- May be run for extended periods between hot washings where many shell and tube exchangers would plug up in minutes.
- May be run at much higher temperature differences between process fluid and coolant than could ever be attempted with shell and tube equipment without serious fouling or plugging.
- May be used over an extremely wide temperature range, from –75 to +100°C. It is usually very difficult to run vacuum crystallization equipment over a broad range of temperatures.
- May be used with high percent solids. Vacuum crystallizers are normally limited to about 25 percent by weight or less solids. This equipment has worked in a range of 65 percent by weight solids as slurry.
- High viscosities are not a problem, with several crystallizations being carried out from mother liquor with viscosities of 10,000 cp or higher (see Fig. C-43).



FIG. C-44 Crystallizer for separation of aromatic isomers. (Source: Armstrong Engineering Associates.)

- Flow pattern in once-through operation closely resembles plug flow so conversion of batch to continuous processes is easy, and virtually any desired time/temperature pattern is possible.
- In small-capacity cases, a scraped surface crystallizer will be very inexpensive. This is also true in cases where, for much larger installations, vacuum crystallization may seem most attractive.

Wide range of capacities

Scraped surface crystallizers have been used over a wide range of capacities, from the smallest continuous operations (typically about 1000 tons/year) up to 250,000 tons/year. There is no practical upper limit to capacity.

Good solubility curve

Cooling crystallizations are obviously most advantageous where the solubility curve will produce good yields with simple cooling of the mixture. This is true of a wide variety of organic mixtures. See Figs. C-44 and C-45.

Low-temperature crystallizations

Scraped surface continuous crystallizers offer the best approach for low-temperature crystallizations such as: the separation of meta- and paraxylenes or oleic and linoleic acids.



FIG. C-45 Stainless steel process side crystallizer for oligomers formed in fiber processing—three separate process duties are included. (Source: Armstrong Engineering Associates.)

Products with high boiling point rise

Some mixtures of inorganic chemicals in water show very high boiling point rises as concentration proceeds, reducing the vapor pressure, and dramatically increasing the vacuum requirements. Many such mixtures produce abundant crystal growth on cooling. Often a scraped surface continuous unit may be used in conjunction with a vacuum unit, with the vacuum unit doing the high-temperature part of the crystallization and the scraped surface unit doing the low-temperature part.

Products with similar vapor pressures

Many aromatic chemicals, particularly isomers, have nearly identical vapor pressure characteristics, which makes distillation very difficult. However, these same mixtures often have widely varying freezing points, which makes crystallization simple and effective.

High viscosity fluids

High viscosity, due either to high mother liquor viscosity or high percent solids, does not present problems to the scraped surface continuous crystallizer but may make other types of crystallizers totally inoperable.

Severe fouling

The fouling tendencies of many slurries are overcome because the deposits on the heat transfer surfaces are continuously removed.

The following list of compounds is incomplete because in some cases manufacturers are not made aware of the material they are working with, and in other cases, manufacturers are bound by secrecy agreement not to discuss the use of equipment with a specific product.

- Anthracene
- Anthraquinone
- Benzene Hexachloride
- Benzoic Acid
- Bisphenol A
- Butyl Cresol
- Butyric Acid
- Caffeine
- Calcium Nitrate
- Caprolactam
- Cyanoacetamide
- Dibutyl Cresol
- Diglycerides
- Dimethyl Hydantoin
- Dimethyl Terephthalate
- Fatty Acids
- Lactose
- Laurolactam
- Levulinic Acid
- Menthol
- Methionine
- Monoglycerides
- Naphthalene
- Nitrochlorobenzene
- Oligomers
- Palm/Palm Kernel Fats
- Paracresol
- Paradichlorobenzene
- Paraxylene
- Pentaerythritol
- Potassium Chloride
- Potassium Nitrate
- Sebacic Acid
- Silver Nitrate
- Sodium Carbonate
- Sodium Lauryl Sulfate
- Sodium Sulfate
- Sorbic Acid
- Sterols
- Tall Oil Fatty Acids
- Tallow Fatty Acids
- Tetrachlorobenzene
- Tetramethyl Benzene
- Vitamins
- Waxes

As mentioned earlier, there are substantial differences between processes. Crystallizers are designed to handle a specific duty. What might be right for one application may not be appropriate for another.

The following are examples of applications that require different approaches to achieve the separation of materials by cooling crystallization.

Separation of chlorobenzenes

Para- and orthodichlorobenzene, which are used in the example on solubility thermodynamics, represent two important chemical products that lend themselves to separation by cooling crystallization. The paraisomer crystallizes at temperatures far above the point where either ortho crystals, or the eutectic is reached. Paradichlorobenzene forms extremely tough crystals, which adhere readily to any cooled surface, requiring vigorous scraping to remove them. These tough crystals can stand a certain amount of abuse without degradation in size.

Normally the mixture produces a very thick slurry. Great care must be exercised to handle it. The extremely steep solubility curve presents many opportunities for good crystal growth. However, there is a danger of uncontrolled crystallization, which must be handled carefully or the entire unit may freeze solid.

Strong equipment, and ingenious slurry handling, often with staged operations, are the basics of this process and similar separations of xylene isomers, cresols, and other disubstituted benzenes. (See Fig. C-46.)

Separation of fatty materials

Fatty acids from tallow or tall oil, mono-, di-, and triglycerides, fatty alcohols, and related compounds all may be separated by crystallization when other separation

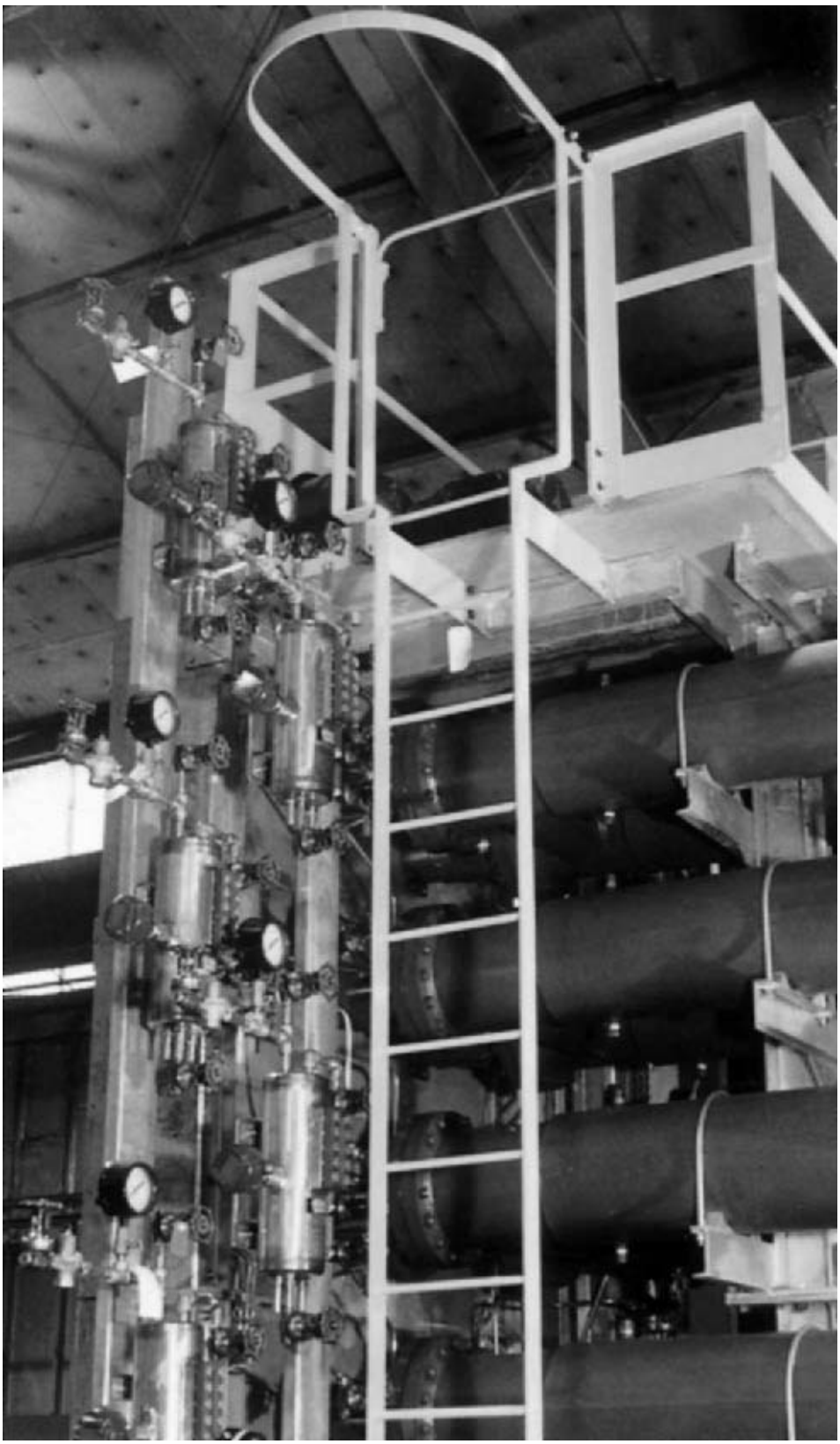


FIG. C-46 Drive end of a special unit which includes mechanical seal systems. (Source: Armstrong Engineering Associates.)



FIG. C-47 Large installation of wax crystallizers in a petroleum refinery. (Source: Armstrong Engineering Associates.)

methods will not work. The extremely delicate nature of the crystal and the sensitivity to shear, which can rapidly produce an inseparable crystal, must be taken into account when separating these materials.

The time/temperature relationship is also of extreme importance, sometimes requiring sophisticated cooling arrangements on the shell sides of the equipment.

Solvents are sometimes used to obtain optimal separations, although solvent-free separations using detergents to separate saturated and unsaturated compounds have also been frequently used.

With this process, crystal growth is relatively slow. Care must be exercised to allow time to grow a decent crystal, which may be easily separated. Reducing shear is more important than producing a rugged machine for handling these delicate materials.

Dewaxing lubricating oil represents the largest use of scraped surface continuous crystallizers (Fig. C-47). Wax has the same boiling point range as lubricating oil fractions, but has a much higher freezing point. Therefore, cooling crystallization is a very effective way to separate the two materials.

Many of the processing plants are quite large and require many scraped surface continuous crystallizers, often with a number of units in a series. Larger plants usually require several parallel trains of crystallizers.

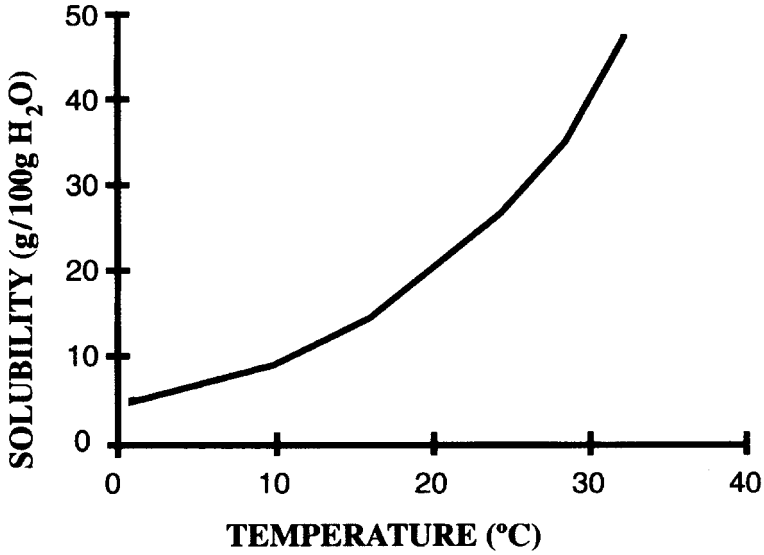


FIG. C-48 Solubility of Na_2SO_4 in H_2O. (Source: Armstrong Engineering Associates.)

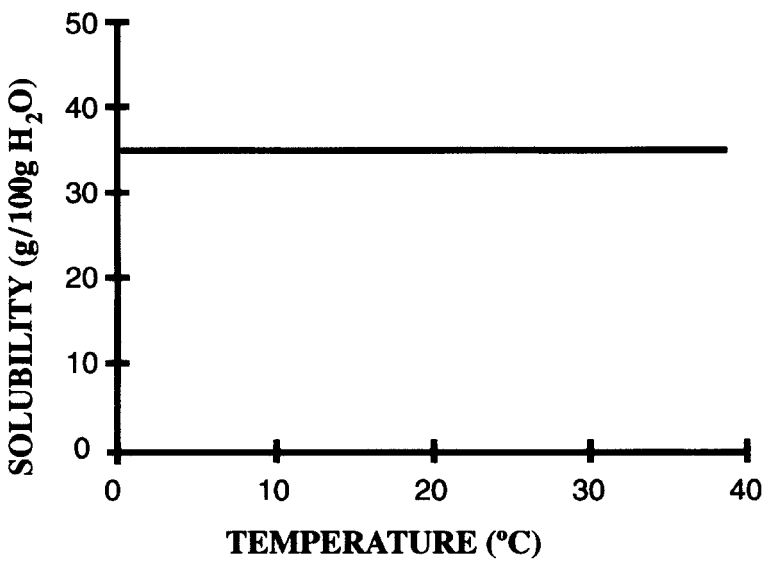


FIG. C-49 Solubility of NaCl in H_2O. (Source: Armstrong Engineering Associates.)

The basic goal of designing scraped surface continuous crystallizers for dewaxing is to ensure longer time on stream between turnarounds.

Some inorganic chemicals have a steep solubility curve with temperature, i.e., a small amount of cooling produces a substantial crystal yield. Such materials are well suited for cooling crystallizations. A typical such solubility curve is shown in Fig. C-48.

Many inorganic compounds have relatively flat solubility curves as shown in Fig. C-49. These compounds are not well suited to cooling crystallization. Vacuum crystallization is the best method of separation for these mixtures.

However, there are some important cases with good characteristics for continuous cooling operations, using scraped surface crystallizers. Some examples of where scraped surface continuous crystallizers offer advantages include: sodium sulfate, potassium nitrate, sodium carbonate, nickel sulfate, ammonium thiosulfate, calcium nitrate, as well as many other inorganic compounds.

Many such processes have been relatively small scale, however some extremely large facilities have also been built. There is no practical upper limit to equipment capacity. The starting cost is modest, and expansion on an incremental basis is simple and often attractive.

The method of cooling can be either direct jacket side boiling refrigerant or brine cooling, depending on the temperature requirements.

Solubility Thermodynamics

In order for cooling crystallization to be an attractive method of separation, it is necessary that one component come out of a solution as the temperature changes. This can be determined by solubility thermodynamics. Understanding these relationships is fundamental to the equipment design.

The ideal case for crystallization

There are a number of frequently encountered cases where the ideal liquid mixture assumptions are applicable. In such cases the solubility, and therefore the ease of separation, can be easily calculated. Many of these cases are reaction mixtures that do not lend themselves to conventional methods of separation. Some frequently encountered examples are:

- Mixed xylenes
- Mixed chlorobenzenes
- Paraffins
- Many multisubstituted benzenes

The nonideal case for crystallization

There are a number of cases where the ideal liquid mixture assumptions are not true.

These include:

- Polar solutes in polar solvents, such as fatty acids in acetone
- Polarized solutes in polar solvents, such as naphthalene in methanol
- Dimerization or hydrogen bonding, such as many organic acids

Prediction of solubility in the ideal case

Under those conditions where the ideal liquid mixture assumptions can be considered to hold, the solubility relationship is quite simple. In the ideal case, the solubility curves and eutectics can be fairly accurately predicted using the Van T'Hoff equation:

$$L_n X_a = \frac{\lambda_a}{R}\left(\frac{1}{T_o} - \frac{1}{T}\right)$$

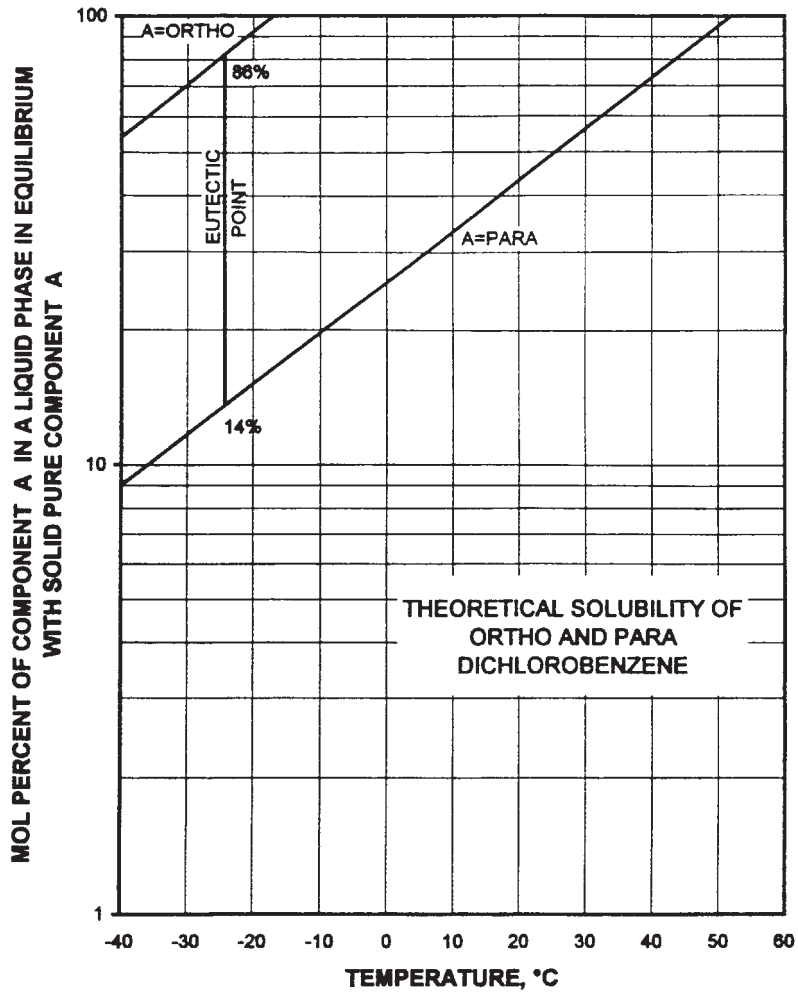


FIG. C-50 Theoretical solubility of ortho- and paradichlorobenzene. (Source: Armstrong Engineering Associates.)

where

X_a is the mole fraction in solution of component a
λ_a is the molar heat of fusion of component a
R is the gas constant
T_o is the melting temperature of component a at the system pressure
T is the system temperature

Therefore given the melting point of a substance and its molar heat of fusion, it is possible to predict its solubility in an ideal mixture and, by judicious use of these results, predict the eutectic temperature and composition.

Numerical example

Figure C-50 illustrates a direct plot of the Van T'Hoff equation, relating the mole fraction of both ortho- and paradichlorobenzene in solution of an ideal mixture at the temperatures shown. This means that in the case of an ideal mixture of para in a solvent, the composition of the saturated liquid phase is as indicated by the

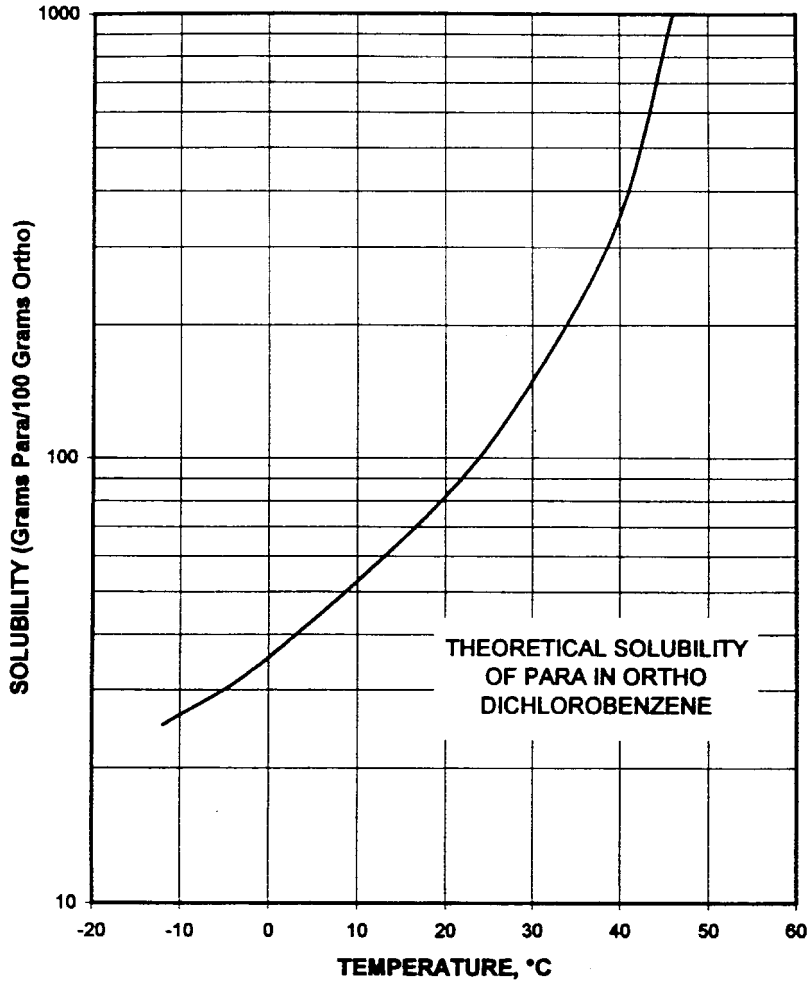


FIG. C-51 Theoretical solubility of para in orthodichlorobenzene. (Source: Armstrong Engineering Associates.)

para curve. For instance, this ideal liquid mixture would contain about 26 percent para at 0°C regardless of solvent, as long as the system is ideal. The ortho curve has the same significance.

A mixture of ortho and paraisomers is a frequently encountered case, and closely approximates the ideal predictions. In this case, mole and weight percents are identical because the two components are isomers. In this binary case, the solubility of para is read from its curve, and the ortho is the balance needed to sum up to 100 percent. The balance, which is the ortho percent, must be less than or equal to the percents indicated by the ortho curve, otherwise an impossible case of more than 100 percent could arise.

At the point where the sum of the ortho and para curves equal 100 percent, the eutectic occurs. Further change of composition for that system is impossible. There is a unique temperature where this will occur. At this point the system shows equilibrium between the liquid phase mixture of the two isomers, and solid pure para and also pure ortho as solid. The phase rule indicates this is a determinant point, and if the temperature is further lowered, solidification will take place,

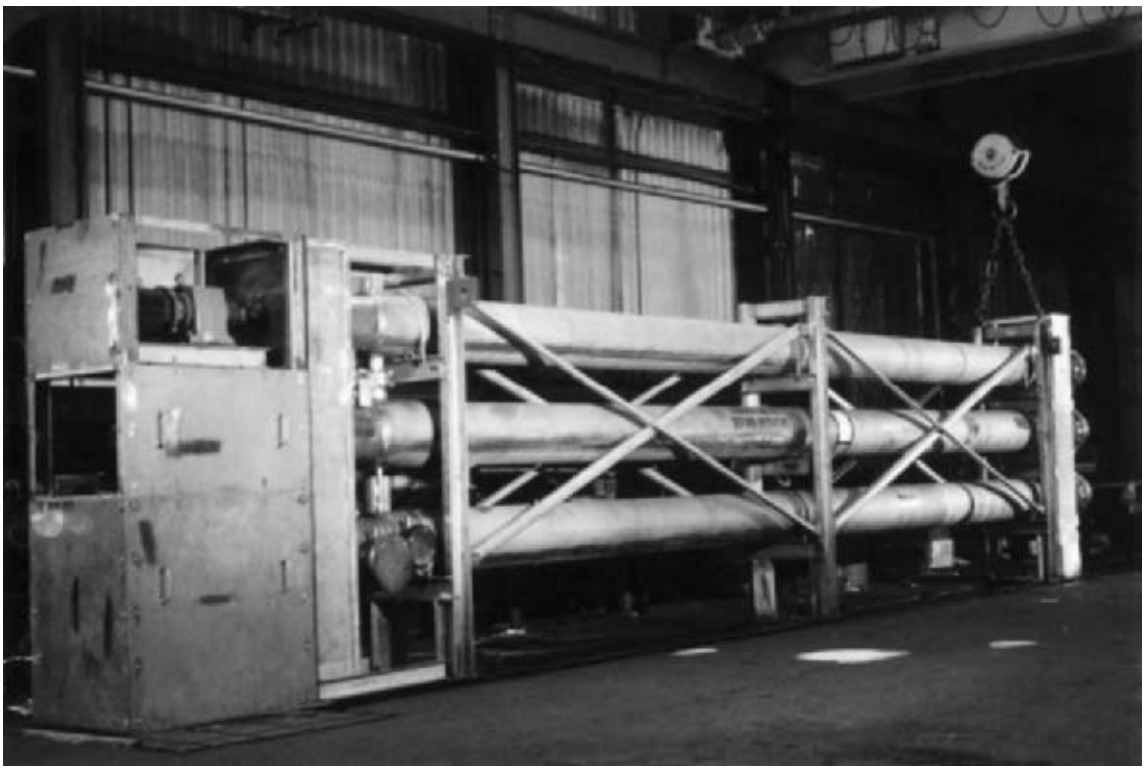


FIG. C-52 Special crystallizer made of all stainless steel, including structures and drive cabinet. (Source: Armstrong Engineering Associates.)

producing the homogeneous eutectic. As nearly as the plot can be read, this turns out to be at –24°C, with a composition of 14 percent para and 87 percent ortho. The International Critical Tables show the eutectic to be at –23.4°C, with a composition of 86.7 percent ortho, and 13.3 percent para, which is certainly in excellent agreement with published data.

Plotting the figure in another way, solubility expressed as grams para per 100 grams ortho will give the results most often used to calculate the recoveries, as illustrated in Fig. C-51.

Here it is clear that the best way to separate a mixture of these two components is to cool the mixture, driving the para out of the solution; as the cooling approaches the eutectic point nearly total recovery is attained.

A simple mass balance will show that a mixture of 65 percent para, 35 percent ortho can be reduced to the eutectic point with a theoretical recovery of around 90 percent of the para. At that point the mixture, ortho rich mother liquor can be sold, or perhaps the eutectic can be split, using solvents to produce pure substances.

While the calculations shown are theoretical in nature, the results are almost identical to published solubility data.

These calculations can be quite useful for basic feasibility work to evaluate whether crystallization is a viable alternative to other separation methods.

If this is the case, verification of the theoretical predictions is usually advisable. Verification is easy to carry out using simple laboratory equipment.

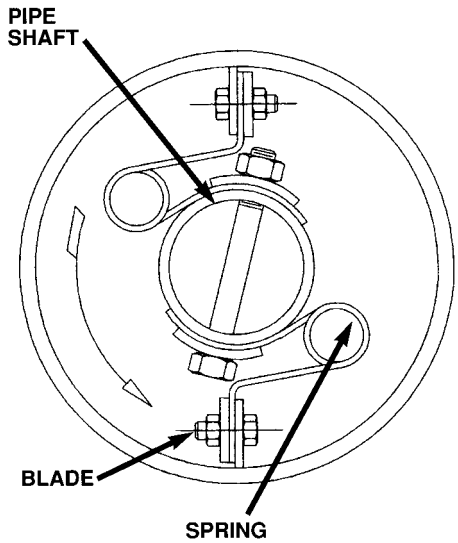


FIG. C-53 Internal arrangement in crystallizer. (Source: Armstrong Engineering Associates.)

Scraped Surface Continuous Crystallizer Specifically Designed for the Growth of Easily Separable Crystals

A scraped surface continuous crystallizer is built as a double pipe element, generally with a 6-, 8-, 10-, or 12-in diameter inner pipe and a larger diameter outer pipe. The area between the inner pipe and outer pipe is filled with cooling fluid. The inner pipe contains a rotating scraper blade element, which mixes the process fluid flowing through the inner pipe and removes deposits that form on the inner pipe wall as cooling occurs. (See special crystallizer in Fig. C-52 and internal arrangement in Fig. C-53.)

In effect, the scraped surface continuous crystallizer is a heat exchanger, but quite an unusual one, because it generally performs as a cooling crystallizer. Heat transfer occurs across the wall in the inner pipe, with cold fluid outside, and process fluid inside. As cooling occurs, crystals tend to form and foul the inner pipe wall. The scraper blades rotate on the inner pipe wall and remove the deposits that would inhibit heat transfer. The majority of the crystallization takes place in the bulk of the fluid, as opposed to the wall, thus allowing growth of easily separable crystals.

Generally, the scraper blades rotate at moderate speeds of 15 to 30 rpm. This speed promotes adequate heat transfer, consistent with good crystal growth and easy separation. A scraped surface crystallizer is sometimes confused with a margarine or ice cream machine since the principle is similar. The final result, however, is very different. Margarine and ice cream machines are also designed to crystallize materials, but the scraper blades rotate at much higher speeds, which produce higher heat transfer and tiny, inseparable crystals. This method is used to produce a homogenized product, but not for separating crystals.

Cooling is carried out by whatever coolant is selected and run through the annulus between the inner and outer pipe. Coolants may be cooling tower water, chilled brines, evaporating refrigerants, mother liquor stripped of crystals, or a combination of these.

A typical commercial-sized installation will consist of several double pipe elements, each with a length of 20 to 40 feet, connected in series. This provides a

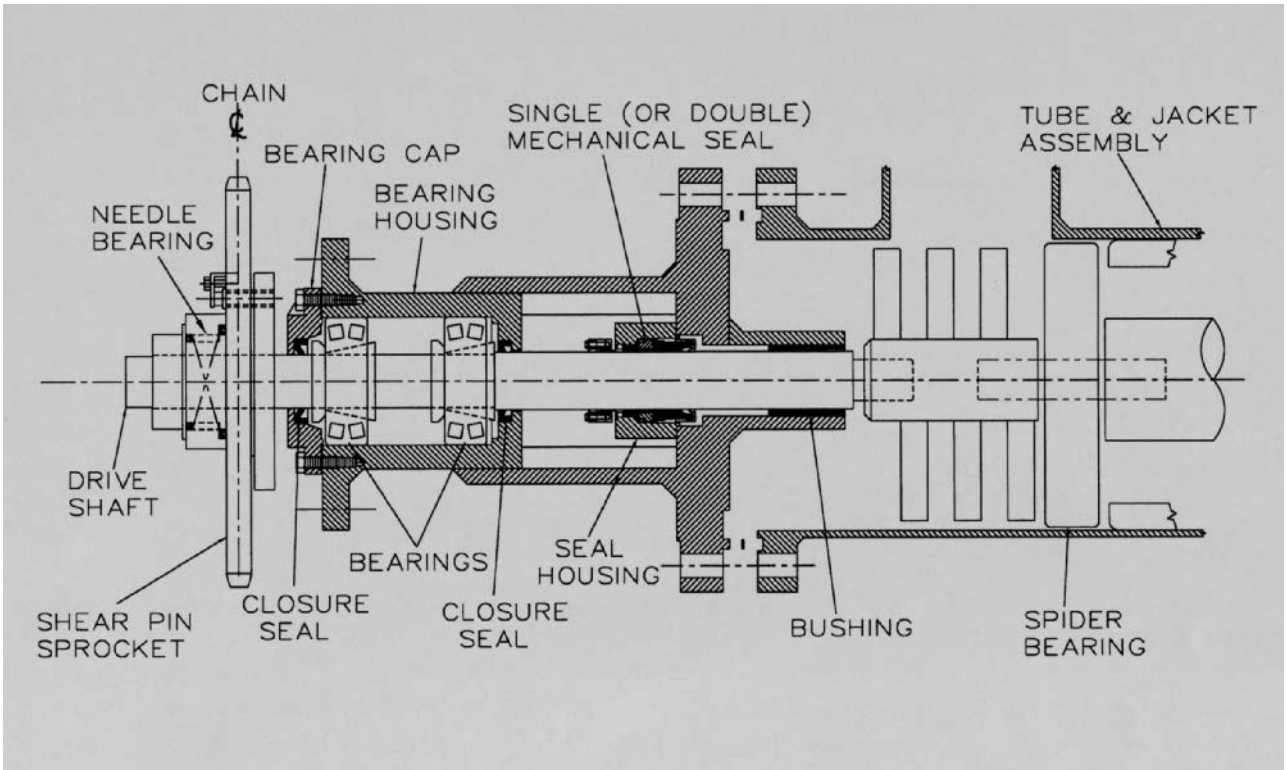


FIG. C-54 Bearing and seal assembly (crystallizer). (Source: Armstrong Engineering Associates.)

long thin flow path that promotes a close approach to plug flow, which is very important in many crystallizations.

Drive system

The drive system on a scraped surface continuous crystallizer must allow free turning of the driven shaft. The shaft must also be sealed against leakage of the process fluids. The drive system used is illustrated in Fig. C-54.

The drive housing is flanged and bolted to the inner pipe. Frequently, mechanical seals are used to avoid leakage around the driven shaft. It is imperative that the drive shaft is completely stabilized to avoid any bending, which would shorten the life of the seal. The use of two heavy-duty self-aligning bearings locks the drive shaft into place, and prevents any unwanted movement, which could cause leakage. This system has proven to be completely reliable in numerous installations around the world.

In concept this arrangement is quite similar to the drive system in many pumps, so maintenance personnel should be familiar with the system.

The use of square end matching parts between the drive shaft and the internal parts allows removal of the scraping element without any need to disturb the drive system. Conversely, the drive system can be maintained without disturbing the scraping mechanism.

Chimneys (*see* Stacks)

Cleaning

If chemical compounds need to be removed, many potential acidic, alkaline, or organic cleaners are available on the market. Environmental regulations should be carefully heeded, as there have been many recent changes with respect to carcinogenic content. These changes can be stated in both federal and state or provincial laws. The strictest of the different laws should be selected for compliance to minimize the risk of lawsuits and other potential problems.

Cleaning a surface might require just a water wash, a hot-water wash, or a steam jet. Temperature and purity of the water or steam used will depend on the material to be cleaned and the process. Cleaning may require soap or detergent of some kind.

Manufacturers' precautions need to be observed for reasons of safety and environmental considerations. For instance, many detergent companies now advertise their phosphate-free content to people supporting the environment. Although domestic detergents are not classified as poisonous, many large laundry facilities, such as those in large hotels, ask that clients conserve the use of linens and soap to reduce the effects of detergent solution in sewer effluent. Phosphates in detergents are among the "nutrients" in ocean environments, higher levels of which are attributed to accelerating the growth of "red tide" algae (given the right sunlight, salinity, and water temperature). Plankton species involved in "red tide" include *Ptychodiscus brevis*, which produces a poison called brevetoxin. In 1989, NOAA (National Oceanic and Atmospheric Administration, USA) stated that this poison had caused the death of dolphins on an unprecedented scale or sufficient weakening of their immune systems that they died from other less serious causes. This has consequences with respect to human health as dolphins are mammals very high on the food chain. The only mammals higher than dolphins are humans.

For additional material on air cleaning, *see* Air Filtration; Air Pollution Control; Air Purification.

For gas-turbine cleaning, *see* Turbines.

Specialized cleaning processes, and the corresponding customized equipment, are frequently required in industries such as agriculture and pharmaceuticals. Consult manufacturers of items, such as centrifuges and filters, to discuss specific options. Also consult Some Commonly Used Specifications, Codes, Standards, and Texts for cleanliness requirements for different applications.

Clutches (*see* Power Transmission)

Coatings (*see* Ceramics; Metallurgy)

Cogeneration

The term *cogeneration* means recovery of waste heat. Frequently, the term is used to describe modifications of turbine cycles and some form of recovery of waste heat from the exhaust gases of combustion.

The exhaust gases may be circulated through a heat-recovery steam generator (HRSG) and the resultant steam used to power a steam turbine. This is called a *combined cycle* application (gas turbine + HRSG + steam turbine). *See also* Turbines.

The exhaust gases may be led directly into a greenhouse to provide heat to growing flora. This is termed a *waste heat recovery* (WHR) cycle.

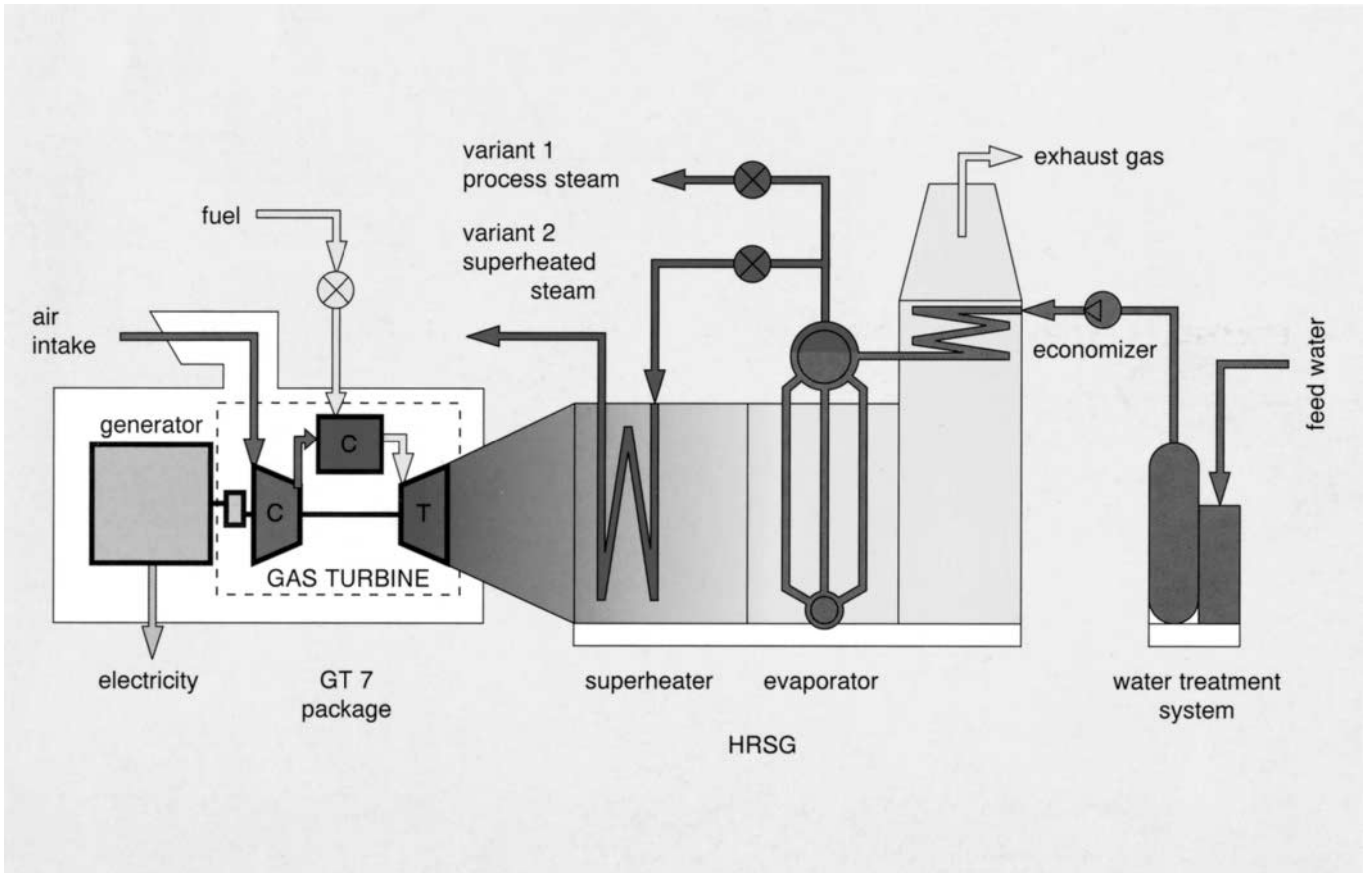


FIG. C-55 Cogeneration based on a GT 7 gas turbine. (Source: Alstom.)

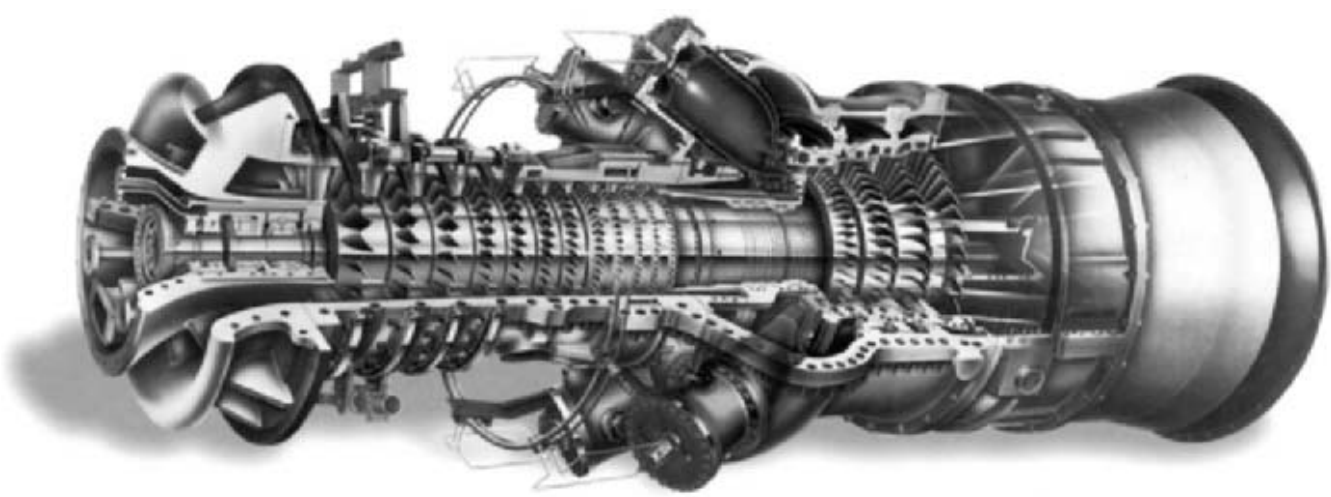


FIG. C-56 The GT 7 gas turbine. (Source: Alstom.)

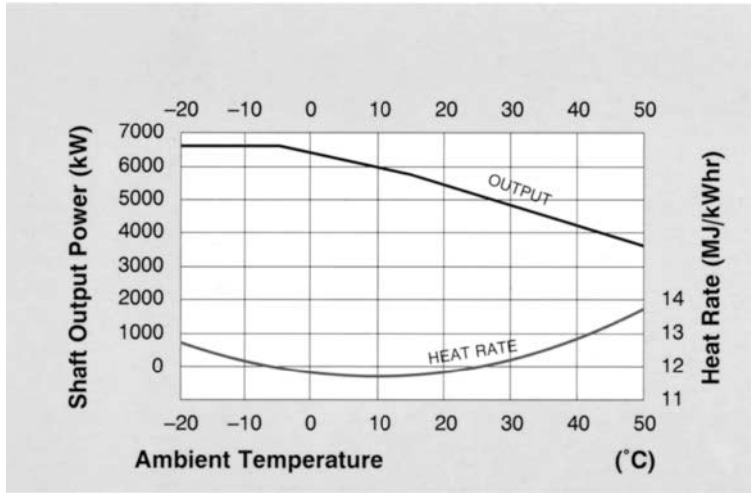
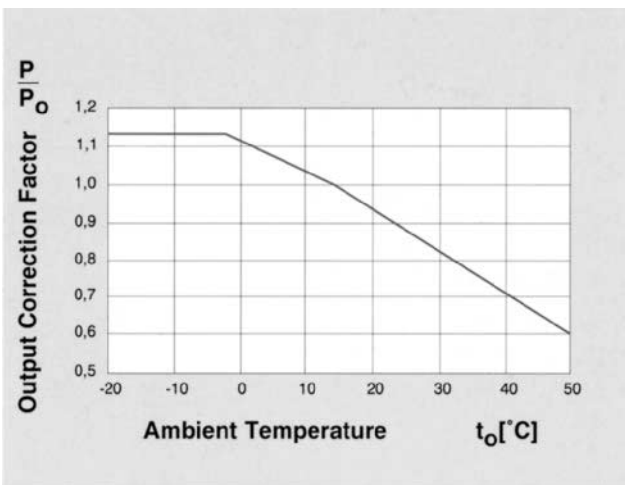
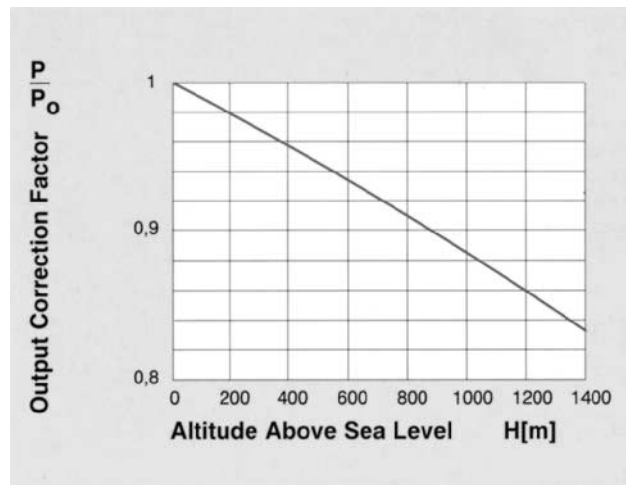


FIG. C-57 Nominal performance curve at gearbox output shaft for a GT 7. (Source: Alstom.)

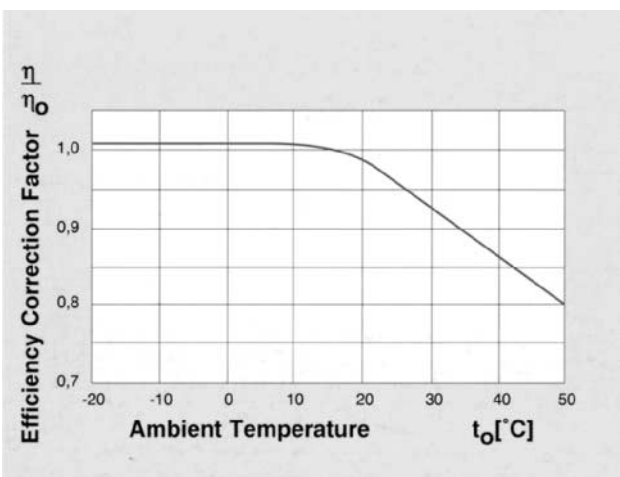
(a)



(b)



(c)



(d)

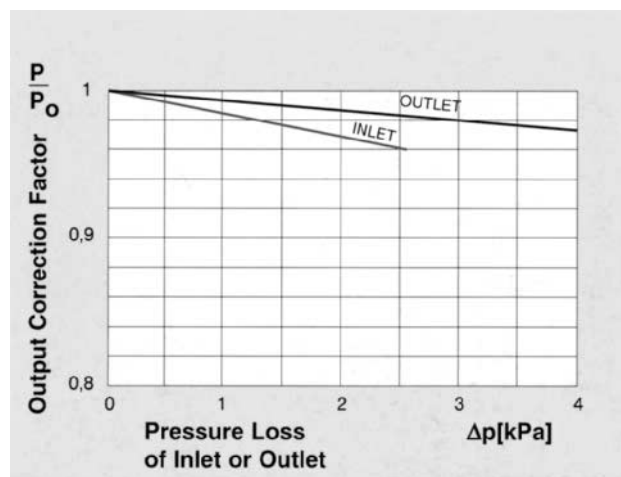


FIG. C-58 GT 7 gas turbine correction curves at gearbox output shift. (Source: Alstom.)

The exhaust gases may be circulated through a heat exchanger and the resultant heat used to provide household or industrial heating needs. This is a *heat and power* (H&P) application.

Figures C-55 through C-61 illustrate a typical H&P application using an Alstom GT 7 turbine. (*Note:* Alstom's GT 7 cogeneration plant is based on Kawasaki Heavy Industries gas turbine type M7A-01.)

The cogeneration unit can operate on either natural gas or liquid fuel with more efficient fuel utilization than separate generation of heat and power. The gas turbine exhaust is coupled to a heat recovery steam generator and turbine exhaust waste heat is used for heating purposes or process steam production.

The high turbine exhaust gas temperature—around 550°C—enables production of steam at high temperatures. Such steam may be required as process steam in the gas and oil industry and paper and pulp mills, as well as in the textile and food stuff industries.

The cogeneration unit based on a gas turbine is environmentally friendly—emissions in exhaust gas from the HRSG's stack meet current environmental regulations.

Modular configuration of the gas turboset, heat recovery steam generator, and accessories enables a quick assembly of the whole cogeneration plant at the site. The unit is designed for continuous base load service and suitable material ensures the long life cycle of hot gas path components.

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.

Coker (*see* Oil Sands)

Color Coding

Color coding has been argued about among plant operators. It may occur in some plants, while others prefer not to use it. Basically, pipes are coded with a colored band and text on the colored band, according to "fire protection" (white letters on red), "dangerous" (black letters on yellow), "safe" (black letters on green), and "protective" (white letters on blue) potential. The width of the band will vary according to pipe size. Colored arrows indicate flow direction. See the safety specifications included in various standards (*see* Some Commonly Used Specifications, Codes, Standards, and Texts).

Columns (*see* Towers and Columns)

Combustor(s); Low NO_x Combustor (*see* Turbines)

Compressors

The two main subdivisions for compressors are turbocompressors (rotating) and reciprocating compressors. These two types fulfill different purposes in process plants. A rough rule of thumb is that *reciprocating compressors* are used in high-pressure ratio (pressure ratio = discharge pressure/suction pressure), low-volume applications. *Centrifugal compressors* were thought of as being high-volume, low-pressure ratio applications. In the 1970s, the commissioning of the Cooper RBB series barrel compressor design at the south Kaybob plant in northern Alberta then meant centrifugals were also suitable for high-volume, moderately high-pressure applications. Those particular compressors were used in

GT 7 Package for Cogeneration Application

Gas Turbine Engine Kawasaki M7A–01

Gearbox
Type: parallel
Output Speed: 1,500 rpm or 1,800 rpm

Generator
Cooling: air cooled
Power Factor: 0.8
Frequency: 50/60 Hz
Capacity: natural gas – 7 059 kVA
liquid fuel – 6 800 kVA
Insulation Class: F

Starter: hydraulic starting system

Gas Turbine Air System

Air Intake Unit
Filter
Silencer

Enclosure Ventilation
Ventilation Fan
Silencer
Filter

Enclosure
Type: outdoor type pre-fabricated
Steel Sound Attenuating Enclosure
Noise Level: 80 dB(A) at 1 m

Baseframe
Welded Frame: common to turbine, gearbox, generator and auxiliaries

Electrical/Control Panels
Gas Turbine Control Panel
Generator Panel
Synchronizing and Protection Panel
MV Measuring Panel
Generator Terminal Box
DC System

GT 7 Cogeneration Power Plant Performance

(Parameters valid for altitude 0 m, inlet duct loss 1.2 kPa, exhaust duct loss 3.0 kPa. Feed water temperature 60 °C, saturated steam pressure 1.8 MPa.)

Natural Gas

Inlet Air Temperature °C	Output at Generator Teminals kW	Fuel Consumption kW	Exhaust Gas Flow kg/s	Gas Turbine Exhaust Gas Temperature °C	Saturated Steam Production t/hr	Electrical Efficiency %
0	6,130	20,275	22.61	550	15.0	30.22
15	5,577	18,962	21.58	560	14.8	29.47
30	4,652	16,904	19.52	581	14.4	27.58
40	4,023	15,522	18.04	598	14.1	25.92

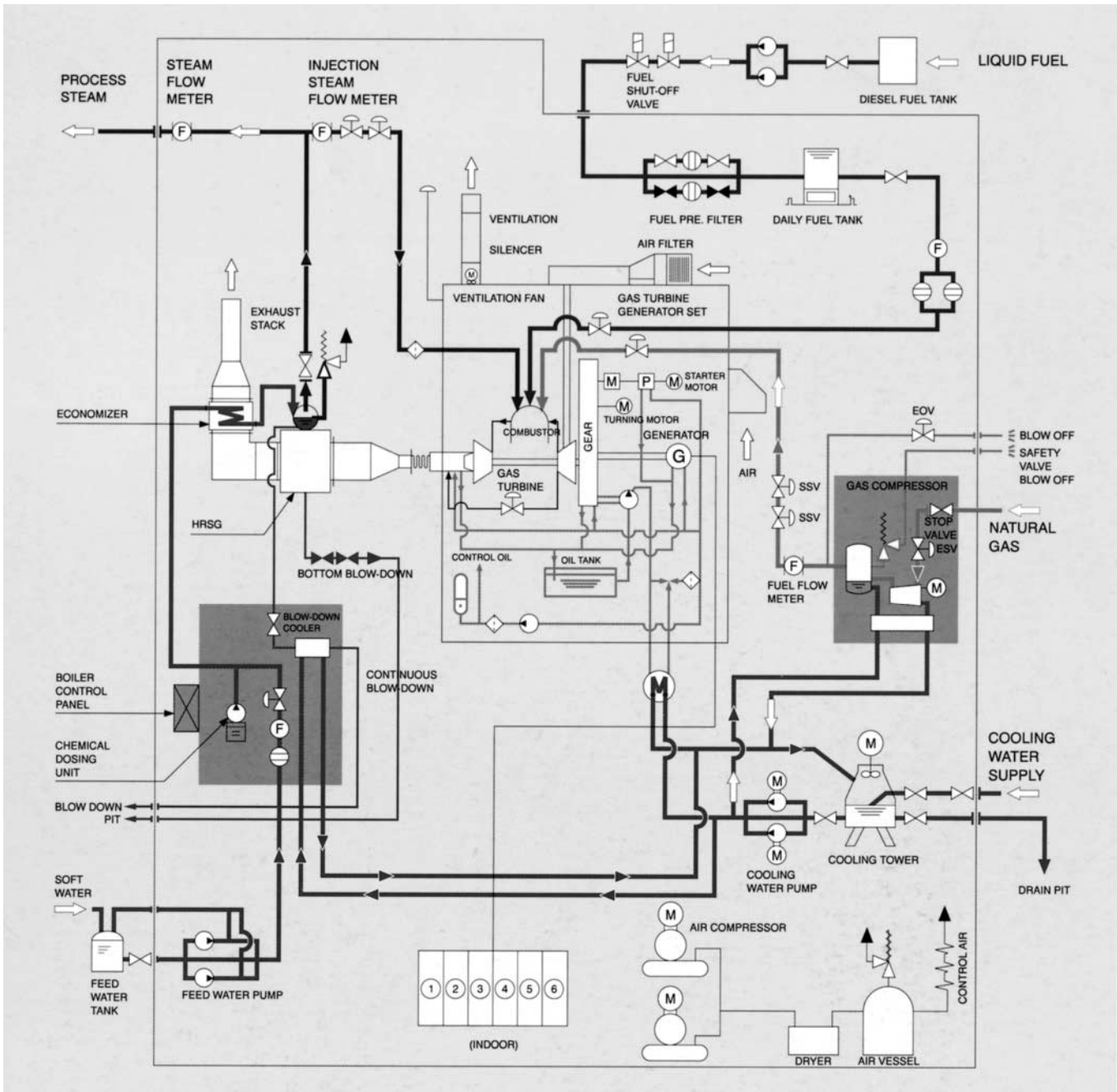
Liquid Fuel

Inlet Air Temperature °C	Output at Generator Teminals kW	Fuel Consumption kW	Exhaust Gas Flow kg/s	Gas Turbine Exhaust Gas Temperature °C	Saturated Steam Production t/hr	Electrical Efficiency %
0	5,990	20,065	22.67	551	15.2	29.9
15	5,450	18,775	21.65	561	15.0	29.0
30	4,540	16,731	19.58	581	14.5	27.1
40	3,930	15,377	18.09	598	14.2	25.6

Weights and Dimensions

Module	Weight (t)	Dimension (m)
Complete unit with GT 7 including gearbox and generator	67	11×3×7
HRSG	~22	~10×3.3×7

FIG. C-59 Data for the GT 7 package for cogeneration application. (Source: Alstom.)



1. Gas Turbine Control Panel
2. Generator Panel
3. Synchronizing and Protection Panel
4. MV Measuring Panel
5. Generator Terminal Box
6. DC System

FIG. C-60 System flowchart: GT 7 gas turbine in cogeneration application. (Source: Alstom.)

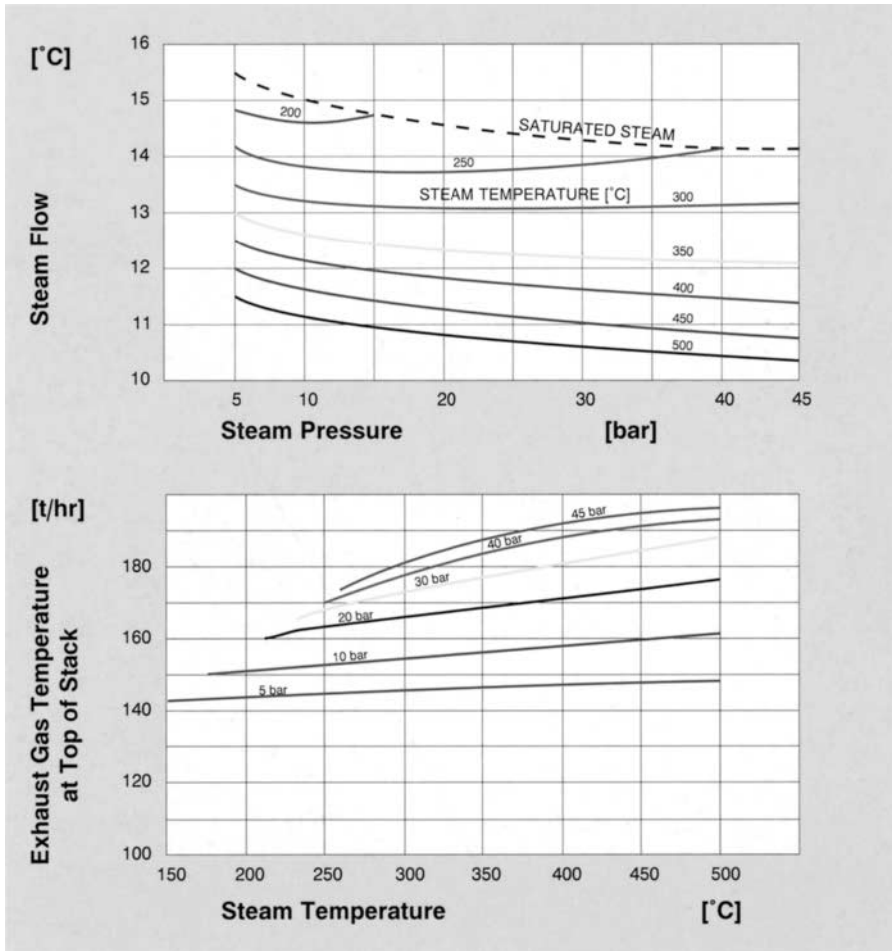


FIG. C-61 Heat recovery steam generator performance curves (GT 7 cogeneration package). (Source: Alstom.)

reinjection application (pushing oil closer to the surface by reinjecting gas in a mixed field).

Where smaller volumes of gas for reinjection purposes are available, a reciprocating design may be used.

Reciprocating*

The most sophisticated design and demanding application among models of reciprocating compressors may be the "laby"™ compressor, which achieves high-pressure ratios without lubrication. This design will be discussed in some detail. An indication of operating ranges for different types of reciprocating compressors may be observed in Fig. C-62.

The "laby" compressor is of sufficient importance to many process applications that the following adapted extracts of a detailed paper are included.

*Source: Sulzer-Burckhardt, Switzerland. Adapted with permission.

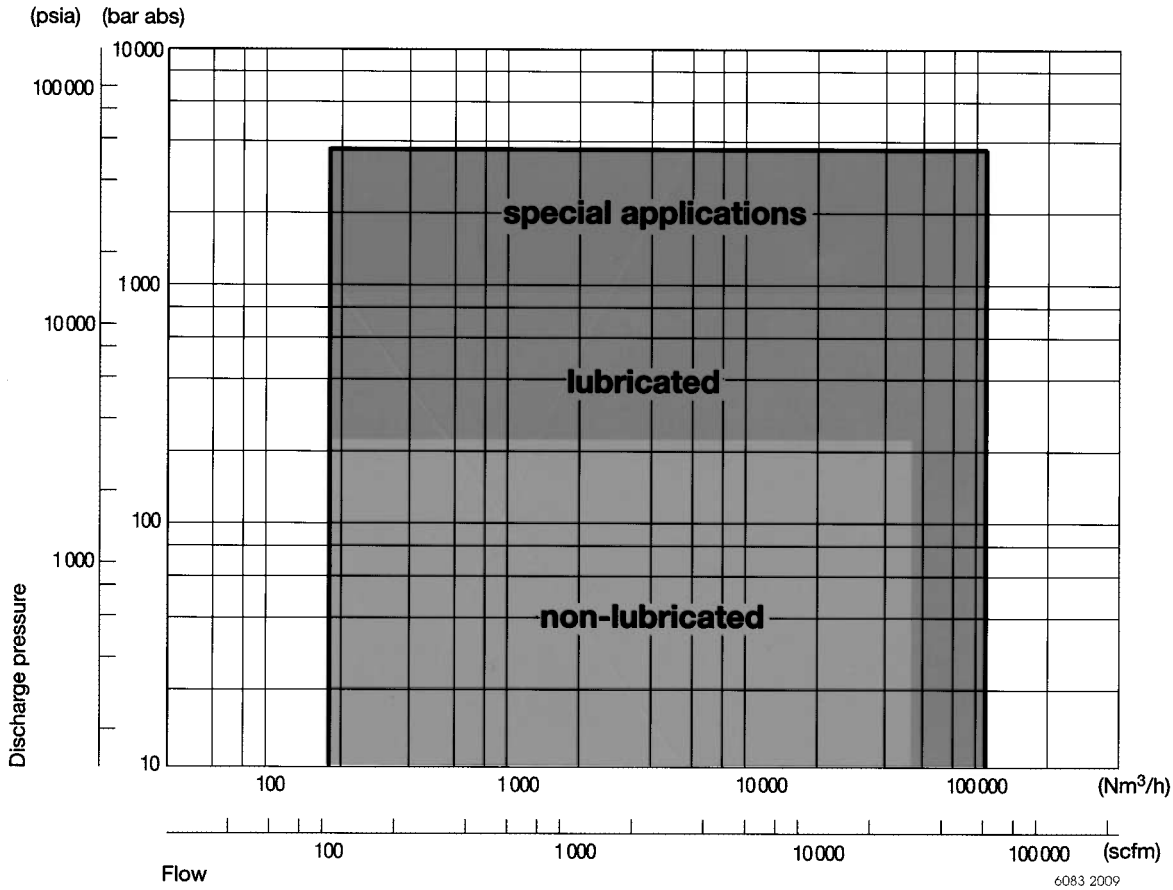


FIG. C-62 Typical performance range of reciprocating process compressors. (Source: Sulzer-Burckhardt.)

Typical applications for "laby" compressors include:

- Petrochemical plants
- Chemical industry
- Natural gas handling
- Polyethylene plants
- Environment protection
- Offshore installations
- Power generation
- Wet oxidation
- Seismic exploration
- Research and development

Typical gases compressed and process conditions:

- Hydrogen
- Hydrocarbon gases

- Hydrogen sulfide
- Hydrogen chloride
- Chlorine
- Highly corrosive gases
- Toxic gases
- Bone-dry gases
- Low-temperature service
- High pressures

Major design and maintenance features

Crankcase and bearings. Rugged compressor frames in cast iron are used for all sizes. This allows an optimized transfer of cylinder forces and rod loads to the frame walls and main bearings. The wall thicknesses are designed to transmit the loads properly and lead to a very stiff frame, thus avoiding uneven elastic deformation between crankcase and cylinders. See Figs. C-63 and C-64. This results in considerable noise reduction and a long life for the crankgear. See also Figs. C-65 through C-68.

Trimetal precision bearings, consisting of a steel shell with lead-bronze layer, babbitt-lined are used. Main bearings and big end bearings are identical and interchangeable. They can be replaced without any manual adjustment. The bearing arrangement ensures that the partition of the two bearing halves is perpendicular to the main load, so that the lubricating film remains unbroken. Oil enters the bearings at points where there is no load and the oil supply bores to the bearings are arranged to maintain an uninterrupted oil flow throughout each complete revolution.

The bearings of heavy-duty process compressors are not considered as wearing parts. The moderate specific load guarantees extremely long bearing life. See Fig. C-69.

Crosshead fastening. The prestressed connection between piston rod and crosshead contains only one connecting element comprising an antifatigue shaft and a screw thread at the piston rod end. The optimized prestress transmits low alternating loads to the thread and can be adjusted entirely without any measuring devices. A simple distance ring is all that is required to properly join the two components. Prestressing is done hydraulically for most compressor sizes requiring no special hydraulic tools. This crosshead fastening element has been developed and is safe. This method by far exceeds the requirements of API 618. See Fig. C-70.

Piston rod runout and crosshead design. A unique procedure is used by this manufacturer to verify the piston rod alignment of horizontal compressors and to reduce the rod runout to a minimum. The system allows the rod runout to be minimized by adjusting the piston rod in relation to the piston by means of a patented eccentric device for the connection of piston and piston rod. It does not need any dismantling of the driving mechanism. This solution is superior to the method recommended in API 618, which advocates crossheads incorporating shimmed shoes. See Fig. C-71.

The easy and quick resetting of the piston rod to its theoretical centerline improves the lifetime of the packing, piston, and guide rings. The need for shoes to



FIG. C-63A Vertical, in-line, reciprocating process compressor, lubricated or oil-free. (Source: Sulzer-Burckhardt.)

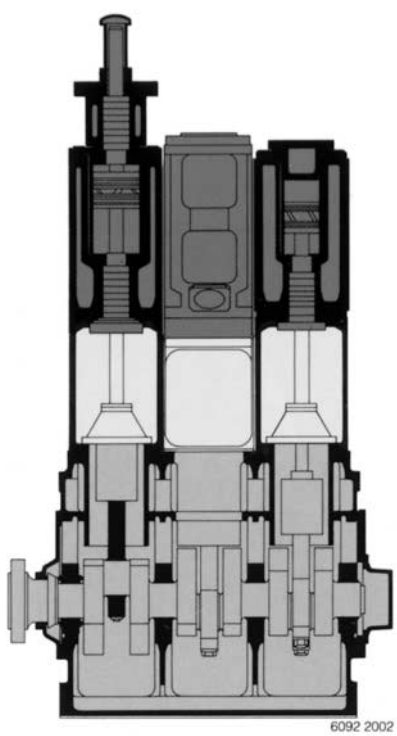


FIG. C-63B Side elevation of Fig. C-63A. (Source: Sulzer-Burckhardt.)

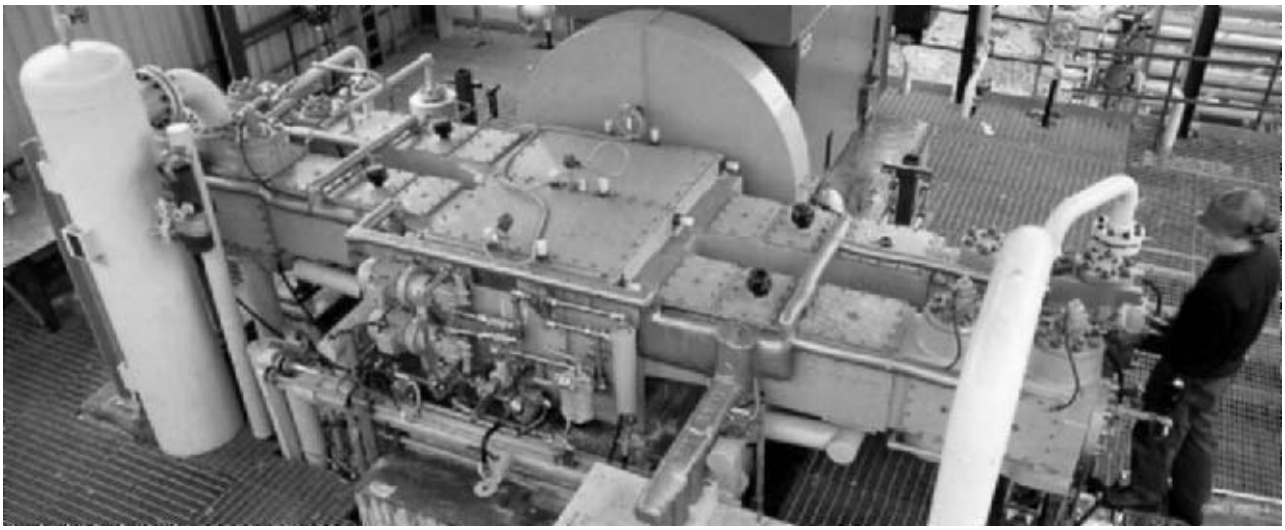


FIG. C-64A Balanced-opposed reciprocating process compressor, lubricated or oil-free. (Source: Sulzer-Burckhardt.)

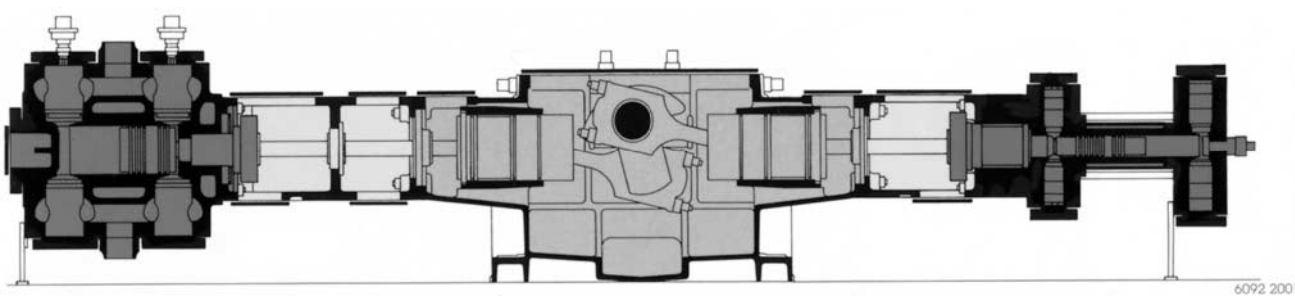


FIG. C-64B Plan view of Fig. C-64A. (Source: Sulzer-Burckhardt.)

be shimmed is rendered obsolete by our patented rod alignment. Therefore we use crossheads with white metal-lined surfaces.

The crosshead is pressure-lubricated. The small end connecting rod bearing runs on a hardened and superfinished full floating crosshead pin. Its specially designed oil feed guarantees a constant unbroken lubricating film during operation.

Piston. A wide variety of piston constructions and materials is available. Depending on the operating condition, we select pistons, sealing rings, wear bands, and piston rods accordingly. See Fig. C-72.

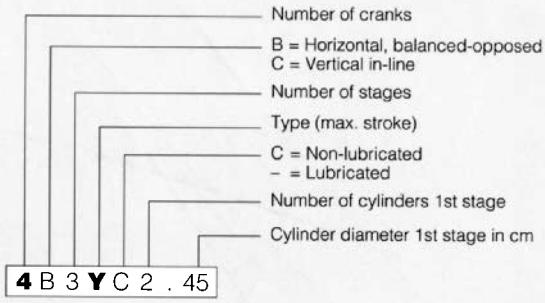
The piston and piston rod connection has a unique and safe locking device. After tightening the piston rod nut to the required prestress, the toothed locking ring can be installed and positively locked in any angular position.

Hollow pistons are continuously self-venting. Nevertheless, they are designed to withstand a differential pressure equal to the maximum cylinder operating pressure.

"Laby" process compressor with benefits for operation and maintenance

Compressor valves. The distillation of many years' experience has guided the design and optimization of compressor plate valves. They contain valve seats and

Key to type designation



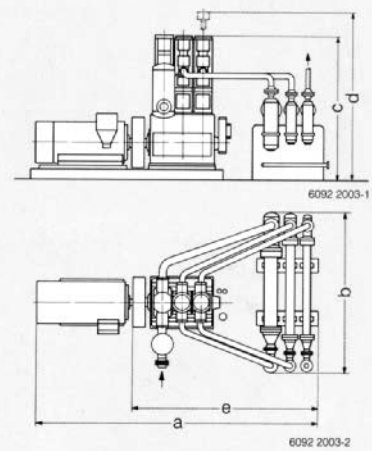
Design data

Type	Max. stroke (mm)	Max. power (kW)	Max. Speed (rpm)
Q	125	300	1 000
Y	160	800	850
S	200	1 600	750
X	270	2 200	600
A	320	3 400	500
C	450	9 500	375
E	560	18 000	300

Main dimensions in millimeters and for estimating purposes only

		a	b	c	d	e
1 crank	**1CQ**	2300	2900	2400	3200	1200
	1CY	3600	4000	3000	4200	1800
	1CS	4200	4600	3400	4600	2000
2 cranks	**2CQ**	3100	2900	2400	3200	1700
	2CY	4800	4000	3000	4200	2900
	2CS	5200	4600	3400	4600	2900
	2CX	7600	6000	3600	5000	5000
	2CA	8100	6500	4300	6000	5200
3 cranks	**3CQ**	4200	2900	2400	3200	2400
	3CY	5900	4000	3000	4200	3900
	3CS	6200	4600	3400	4600	3800
	3CX	8300	6000	3600	5000	5700
	3CA	9000	6500	4300	6000	6100

Vertical



		a	b	c	d	e
2 cranks	**2BQ**	2600	2900	2200	4000	1500
	2BY	3800	4000	2400	6300	1800
	2BS	4800	4600	2600	7500	2600
	2BX	5600	5300	2800	8800	3100
	2BA	6500	5800	3000	10000	3500
	2BC	8300	9000	3600	13000	4800
4 cranks	**4BQ**	3800	2900	2200	4000	2400
	4BY	4700	4000	2400	6300	2700
	4BS	5700	4600	2600	7500	3300
	4BX	6900	5300	2800	8800	4200
	4BA	7700	5800	3000	10000	4700
	4BC	10000	9000	3600	13000	6000
	4BE	12000	11000	4500	15700	8000
6 cranks	**6BS**	6800	4600	2600	7500	4100
	6BX	7800	5300	2800	8800	5000
	6BA	8800	5800	3000	10000	5600
	6BC	11500	9000	3600	13000	7500
	6BE	14000	11000	4500	15700	9700

Horizontal

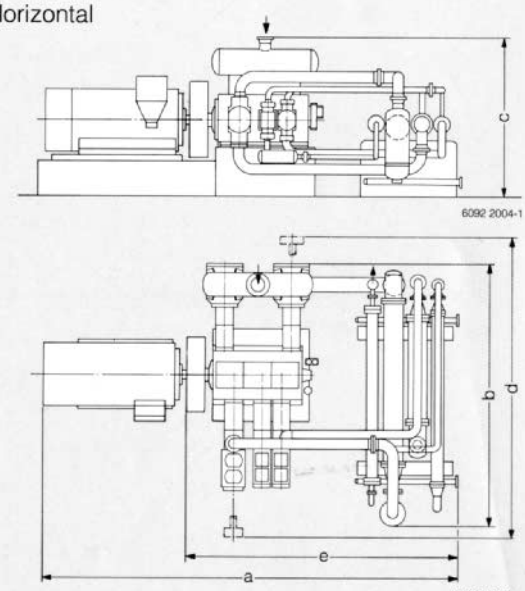


FIG. C-65 Typical reciprocating process compressor design dimensions and corresponding performance range. (Source: Sulzer-Burckhardt.)

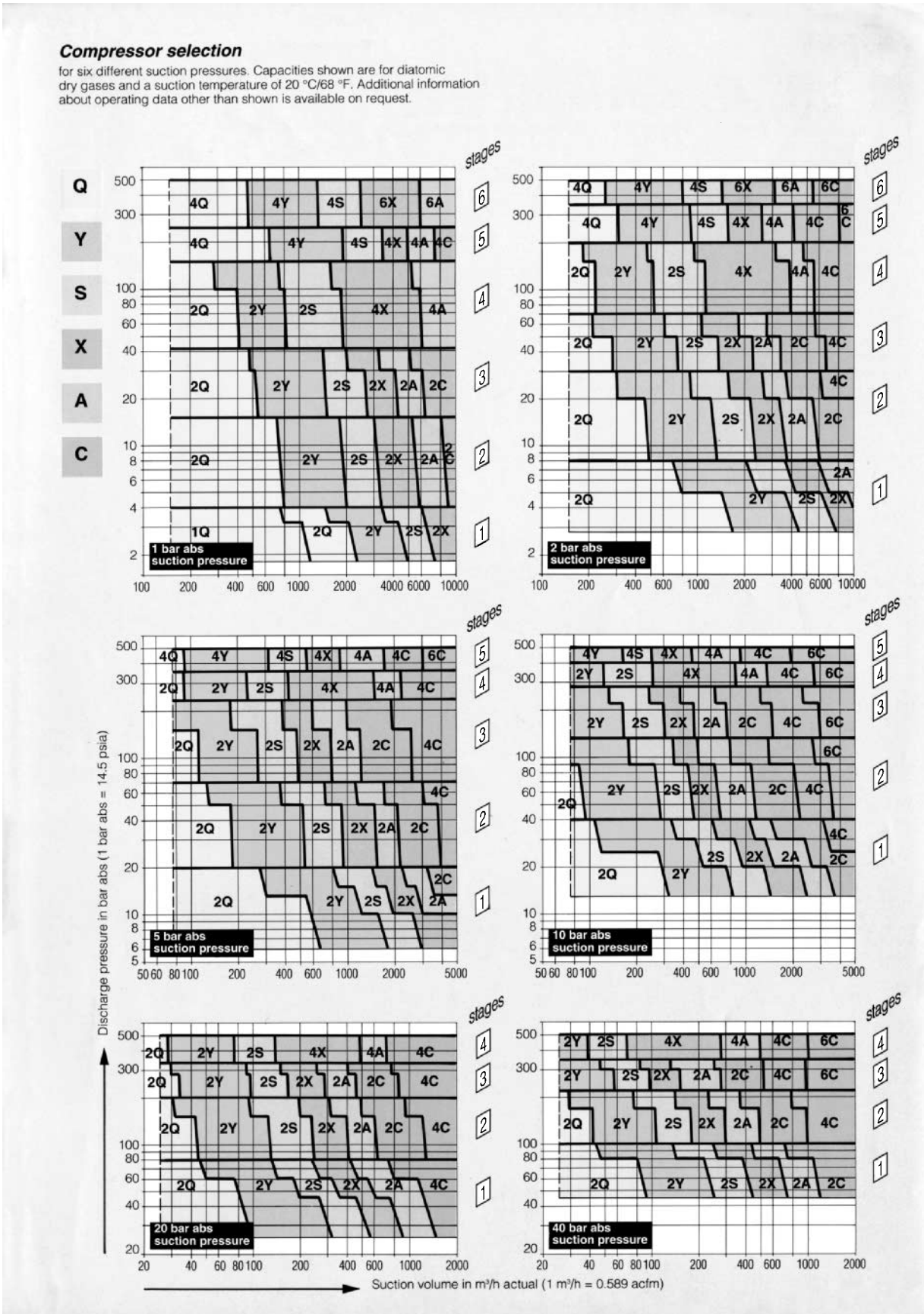


FIG. C-66 Typical compressor selection charts. (Source: Sulzer-Burckhardt.)

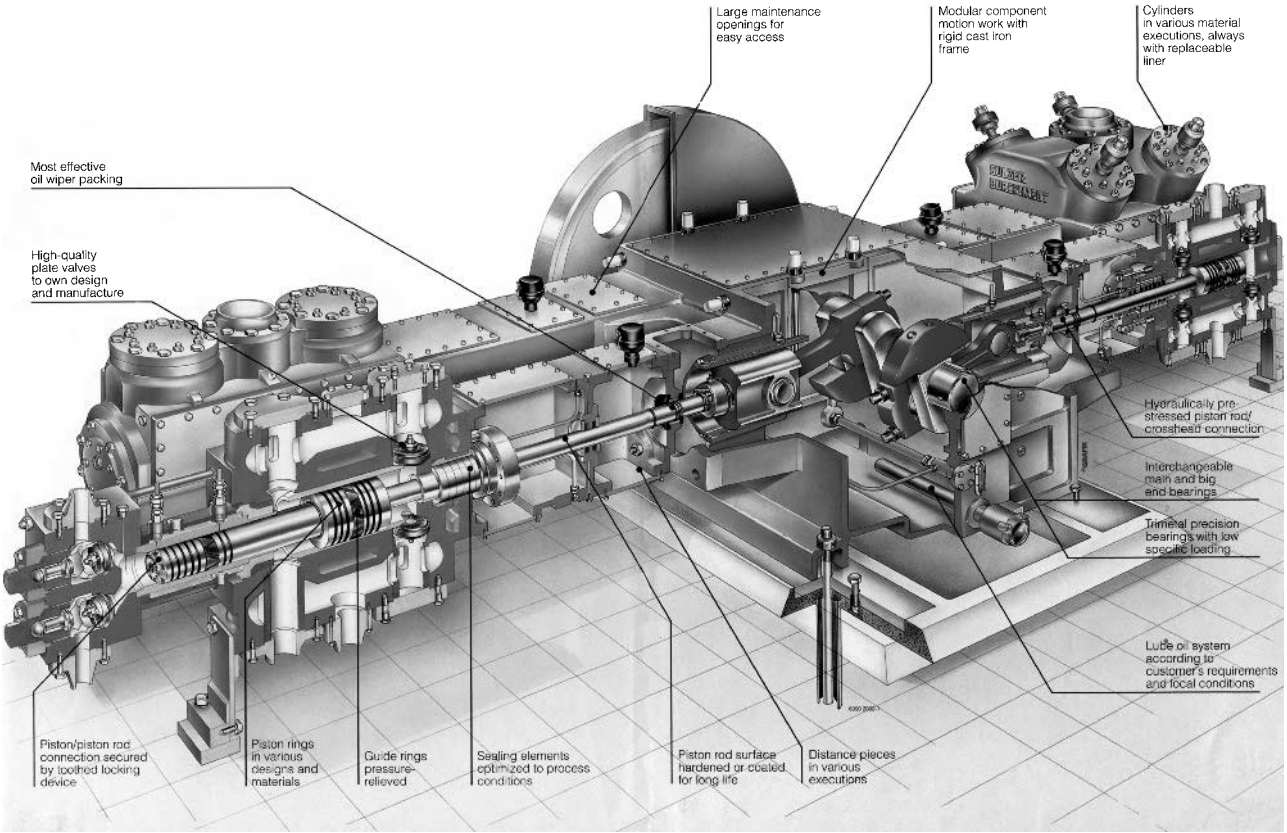


FIG. C-67 Typical features in an API 618 reciprocating compressor. (Source: Sulzer-Burckhardt.)

stroke limiters that are not cast but machined, as well as frictionless guided plates with low lift and good fatigue properties. This OEM has its own compressor valve production facility. This ensures high reliability and availability of the valves and compressors. The valves fulfill the requirements of ISO 8012, API 618, and the European Standard on compressor safety. They are designed to prevent inadvertent wrong assembly into the cylinder. See Fig. C-73.

Suction valves may be equipped with unloading devices for continuous or stepped capacity control.

Special tools. Together with each compressor, the OEM supplies a set of special tools to simplify routine service, like the device to conveniently assemble and install valves for certain sizes of horizontal compressors, or the hydraulic tools, allowing the proper fastening of vital connections. See Fig. C-74.

Piston rod packing. The packing to seal piston rod and cylinder consists of several stacked rings, designed as a cartridge for ease of maintenance. Packing rings are made of PTFE or other newly developed compounds as well as sintered metal. They are subject to stringent quality requirements and feature minimum gas leakage, low friction losses, and minimum wear. See Fig. C-75.

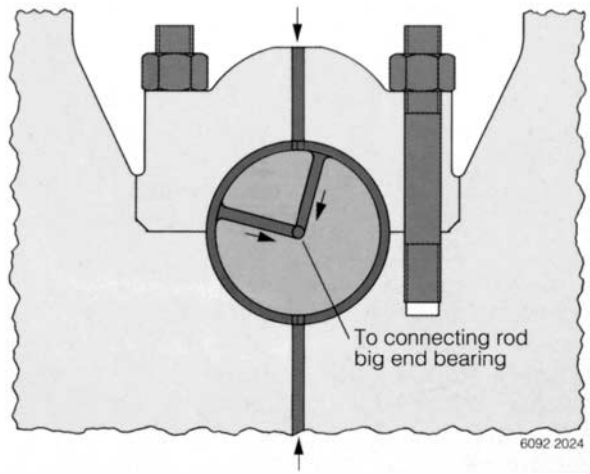




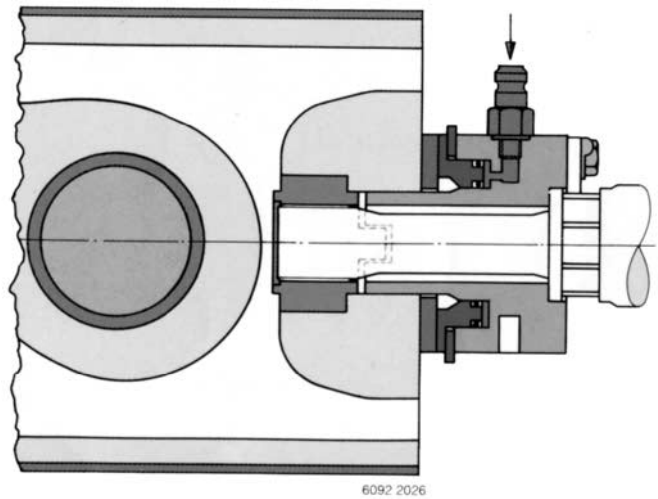
FIG. C-68 Typical crankcase. (Source: Sulzer-Burckhardt.)



FIG. C-69 Typical bearing. (Source: Sulzer-Burckhardt.)

To provide efficient heat removal, direct or indirect cooling is used, depending on the application. Special designs for high-pressure duties or particular process requirements are available.

Piston rod packings are generally well suited to meet the increasing demands imposed by legislation to protect the environment.



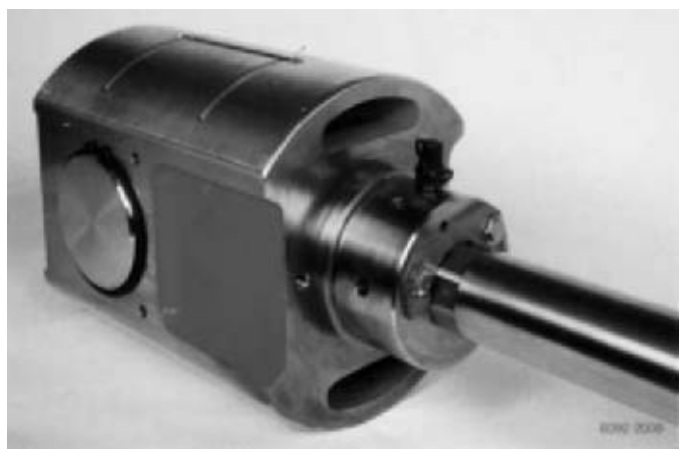
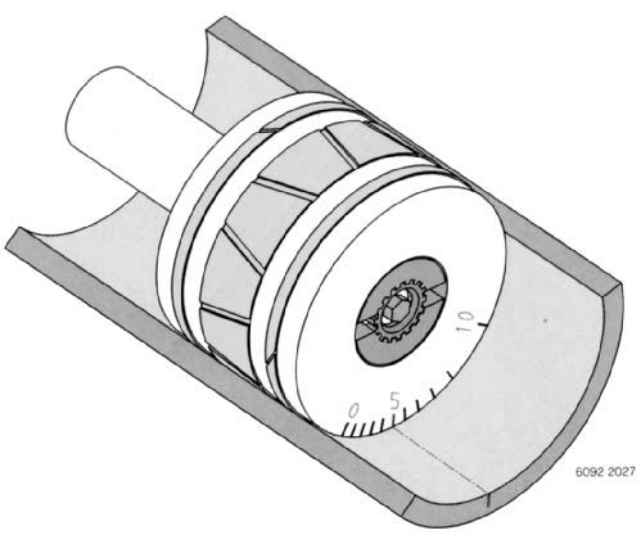


FIG. C-70 Typical crosshead fastening. (Source: Sulzer-Burckhardt.)



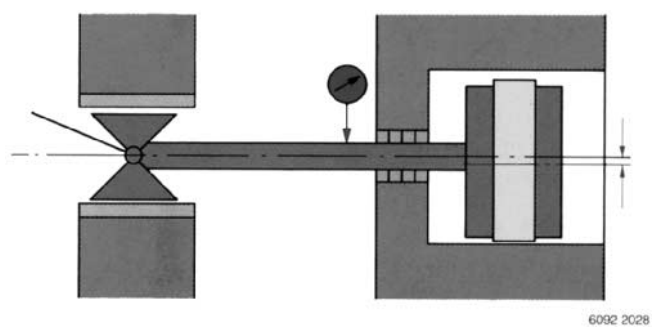


FIG. C-71 Typical piston nod runout and crosshead design. (Source: Sulzer-Burckhardt.)

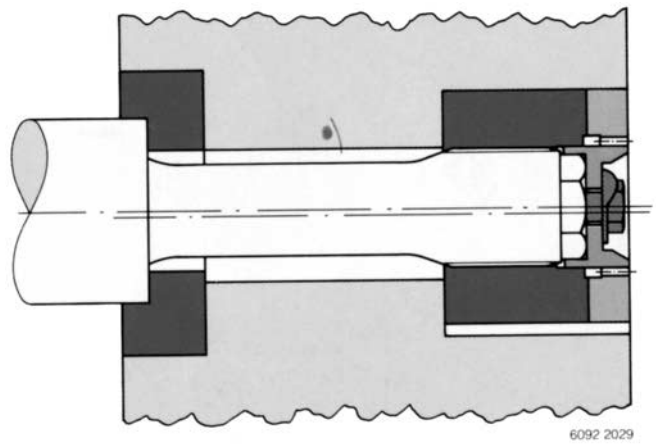




FIG. C-72 Typical piston design. (Source: Sulzer-Burckhardt.)

Oil wiper packing. For oil-free compression no trace of oil may be allowed to creep from the crankcase along the rod into the cylinder. Triple wiper rings—equipped with garter springs—are well known and also used in labyrinth-type compressors for oxygen service. Each of the three rings is provided with an oil wiper lip. The collected oil is led back into the crankcase. See Fig. C-76.

Scraper performance is further improved by providing large communicating passages around the crosshead to avoid crosshead pumping effects.

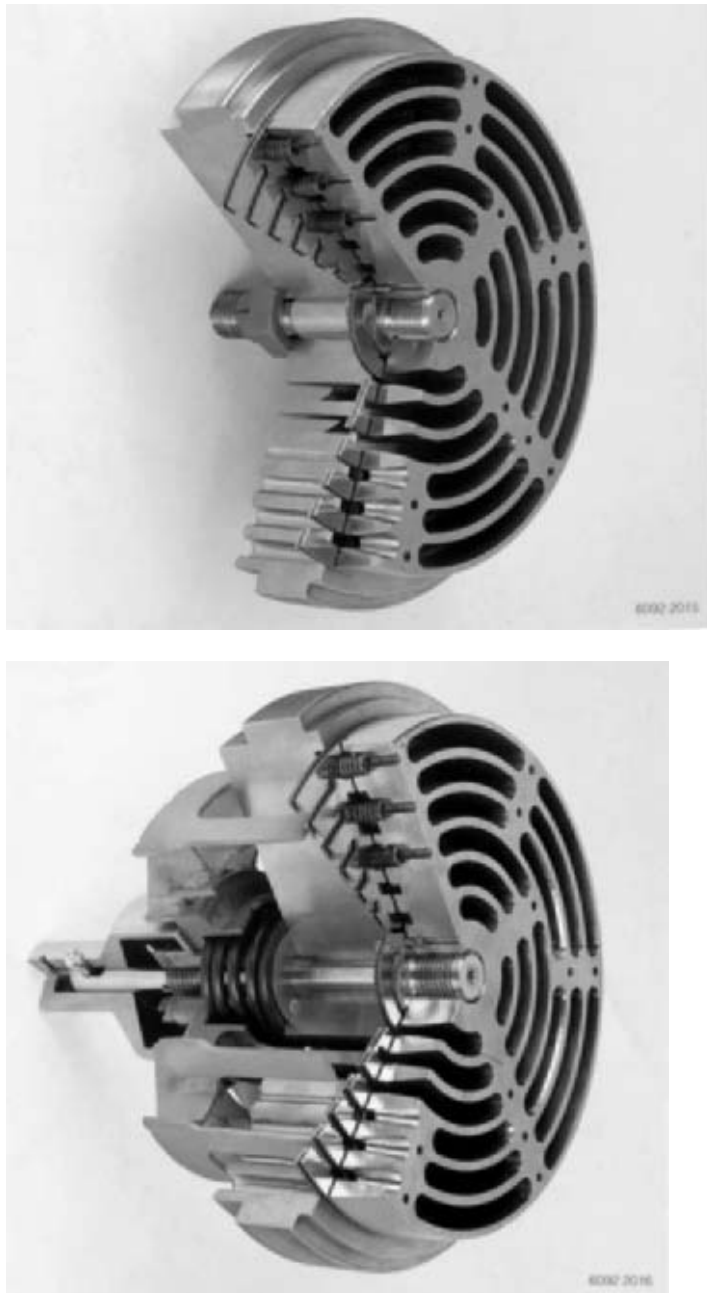


FIG. C-73 Typical compressor valve. (Source: Sulzer-Burckhardt.)

Typical application for reciprocating compressors: water-cooled, lubricated trunk-piston compressor

Typical application processes include: (gases) H_2, He, natural gas, C_nH_m, N_2, Ar, CO_2, air, and other noncorrosive gases and gas mixtures in the following plant systems:

- Bottling plants
- Electrolysis plants
- Air separation plants
- Oil and gas industry

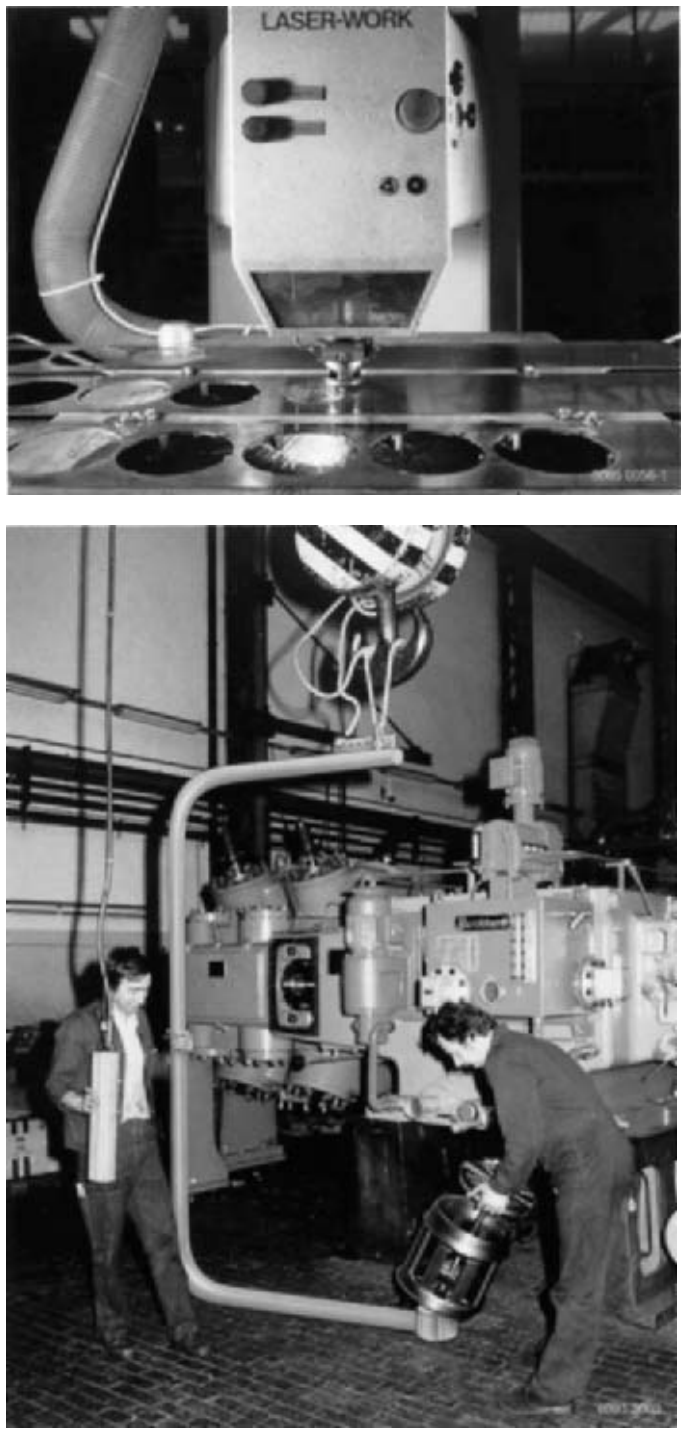
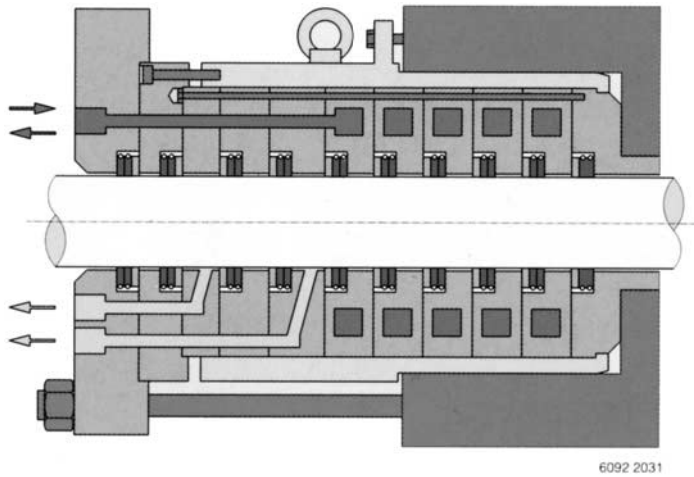
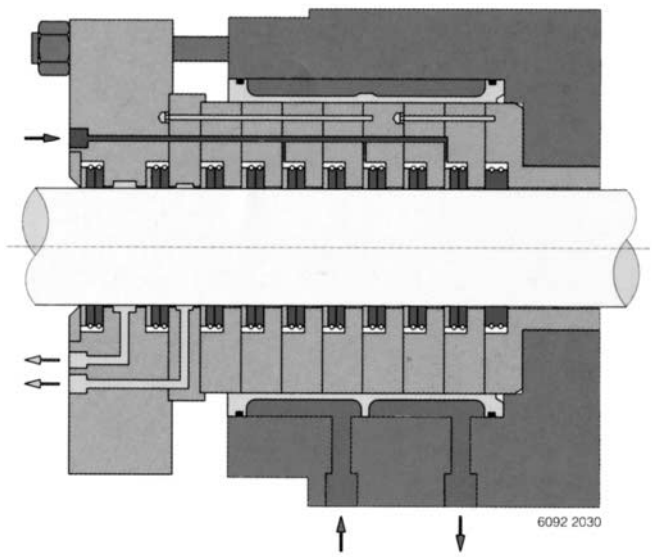


FIG. C-74 Typical cooling for valve installation. (Source: Sulzer-Burckhardt.)





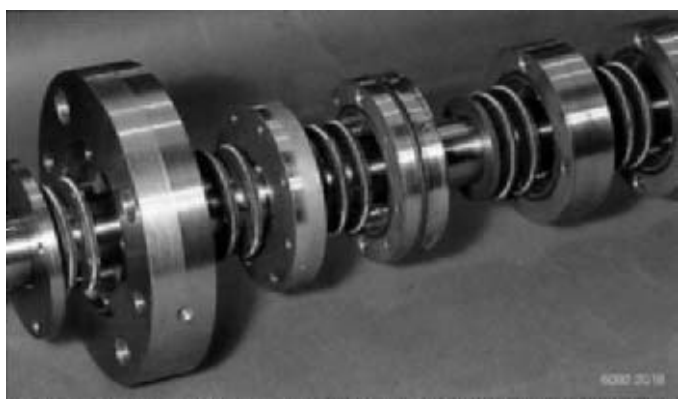


FIG. C-75 Typical piston rod packing. (Source: Sulzer-Burckhardt.)



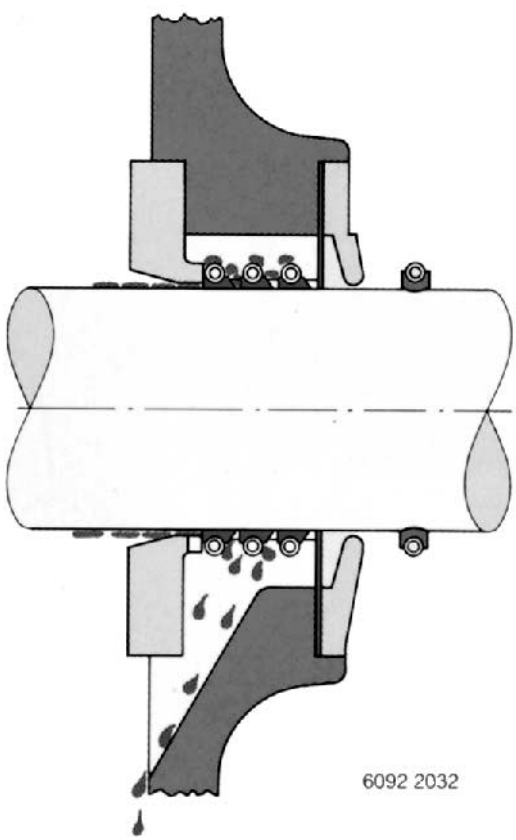


FIG. C-76 Typical oil wiper packing. (Source: Sulzer-Burckhardt.)

- Helium recovery systems
- NGV/CNG refueling stations
- Seismic research
- Gas production plants
- Petrochemical industry
- Chemical process plants
- Oil-well servicing

See Figs. C-77 through C-87.

Special features include:

- Proven heavy-duty trunk-piston compressor of high reliability
- Compact design, good accessibility, easy to maintain
- Fewer wear parts than in crosshead design
- Completely factory-assembled and tested, simple to install without assistance of manufacturer
- Modest space requirement, no special foundations needed; therefore especially suitable as replacement compressor for revamping of plants and/or "last-minute" delivery
- Modular component concept allows optimal adaptation to users' operating conditions
- Standard arrangement for 3, 4, and 5 stages allows short delivery times
- Pressure-tight crankcase for elevated suction pressures up to 16 bar abs (220 psig), with no gas losses to atmosphere
- Automatic condensate drain with integrated unloaded start system
- Combined concentric suction/discharge valves for high volumetric efficiency, low valve temperatures, and fewer valves
- Efficient cooling for continuous duty and low maintenance operation
- Low-pressure ratio per stage provides high-efficiency, low-discharge temperatures, and less wear
- Consider the multitrunk-piston compressor arrangement. The advantages over a single crosshead machine are:
- Minimal investment risk: You start modestly, then add capacity in reasonable steps as requirements increase.
- Reduced wear and longer life: According to demand, compressors either run at optimum load or remain idle. Simple control system.
- High availability: Service your compressors one at a time while the others keep delivering.

See Figs. C-88 through C-90.

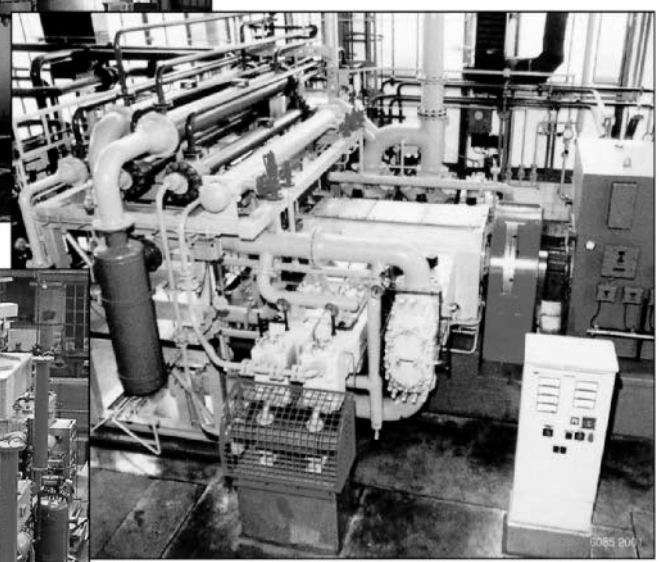
Design features

Crank mechanism and lubrication. The two-throw heavy-duty crankshaft with integral balance weights is supported on roller bearings located in detachable bearing covers. One of these covers also carries the built-in oil pump and filter. The connecting rods have replaceable trimetal bearing shells at the big end and roller bearings at the piston pin. See Fig. C-91.

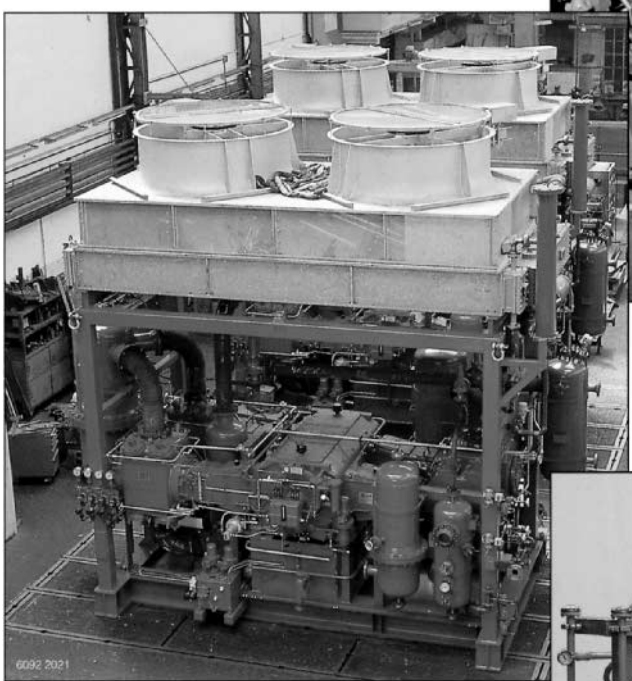
77



78



79



80

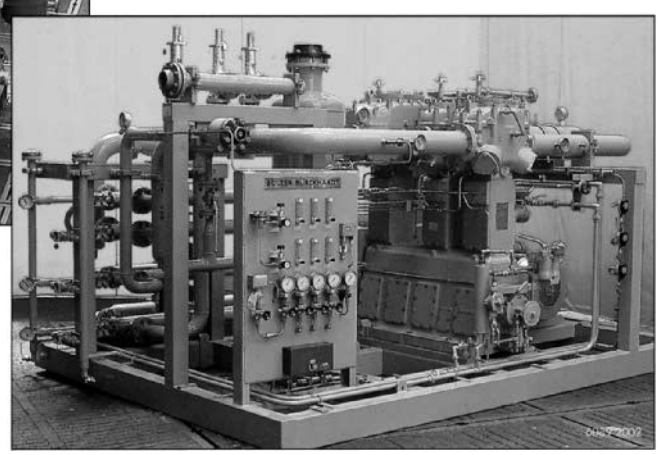
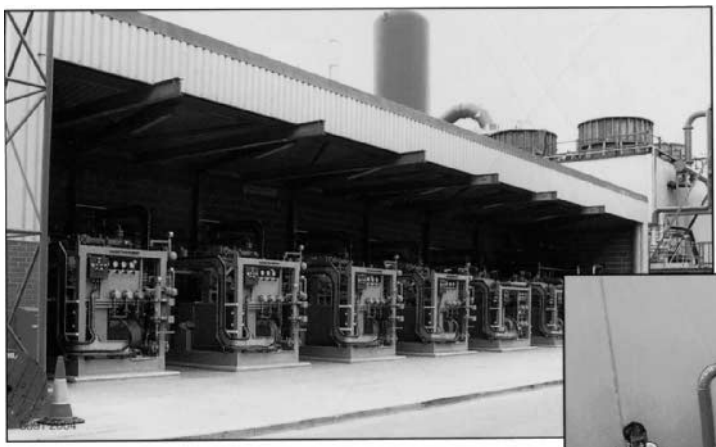


FIG. C-77 Combined booster and primary compressor for an ethylene plant in Spain. Suction pressure 1.7 bar abs, discharge pressure 286 bar abs. Compressor runs at 300 rpm with a power input of 2330 kW. (Source: Sulzer-Burckhardt.)

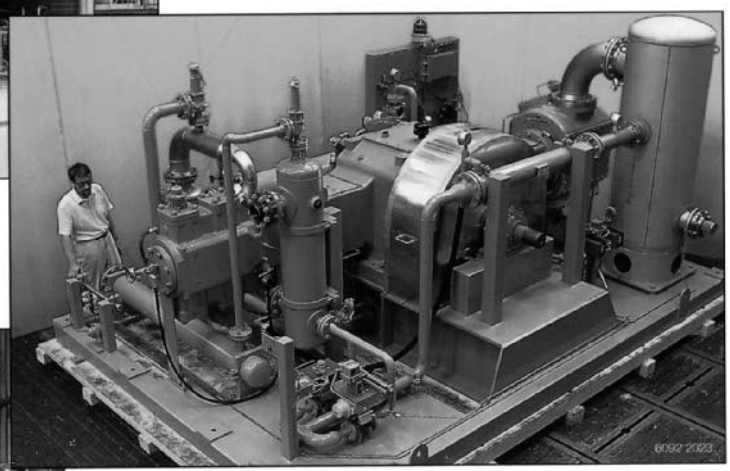
FIG. C-78 Compressor installation in Germany. 2600 m^3/h of hydrogen at 1 bar abs are compressed to 325 bar abs. The five-stage machine operates at 585 rpm with a power requirement of 700 kW. (Source: Sulzer-Burckhardt.)

FIG. C-79 Skid-mounted hydrocarbon gas compressors for offshore duty in Greece. Suction volume 2850 m^3/h, discharge pressure 19 bar abs, speed 420 rpm, power input 400 kW. (Source: Sulzer-Burckhardt.)

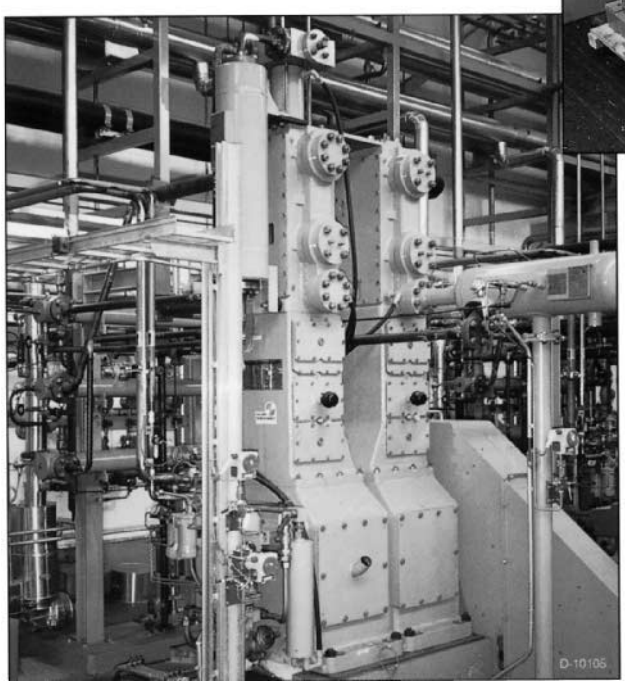
FIG. C-80 Chlorine compressor installed in a chlorine production plant in Colombia. 605 m^3/h of Cl_2 are compressed to 8.5 bar abs. The compressor operates at 480 rpm, the power input is 65 kW. (Source: Sulzer-Burckhardt.)



81



82



83



84

FIG. C-81 Hydrogen producing and bottling plant in Great Britain with two natural gas and four hydrogen compressors. The hydrogen compressors operate at 650 rpm, the discharge pressure is 235 bar abs and the power 80 kW. (Source: Sulzer-Burckhardt.)

FIG. C-82 Hydrogen sulfide compressor supplied for a chemical plant to compress 1040 m^3/h H_2S to 31.4 bar abs. The speed is 495 rpm and the power input 280 kW. (Source: Sulzer-Burckhardt.)

FIG. C-83 Nonlubricated high-pressure compressor in Finland, compressing 750 Nm^3/h dry hydrogen from 17 to 230 bar abs in three stages. Shaft power is 100 kW. (Source: Sulzer-Burckhardt.)

FIG. C-84 Two-stage compressor in the natural gas storage facility of Stadtwerke Bremen, Germany. Suction pressure 70 bar, discharge pressure 166 bar, speed 585 rpm, power 420 kW. (Source: Sulzer-Burckhardt.)



FIG. C-85 Tube trailer filling station; a typical application for hydrogen compressors. (Source: Sulzer-Burckhardt.)

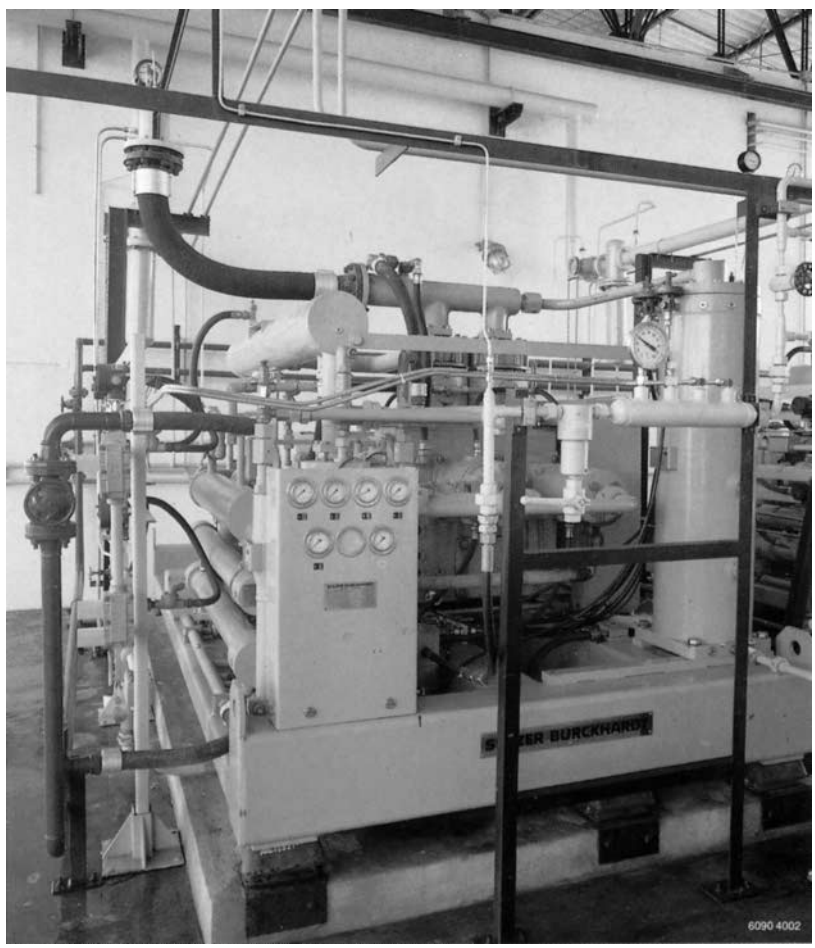


FIG. C-86 Compressor in a chemical works for hydrogen bottling in Brazil. (Source: Sulzer-Burckhardt.)

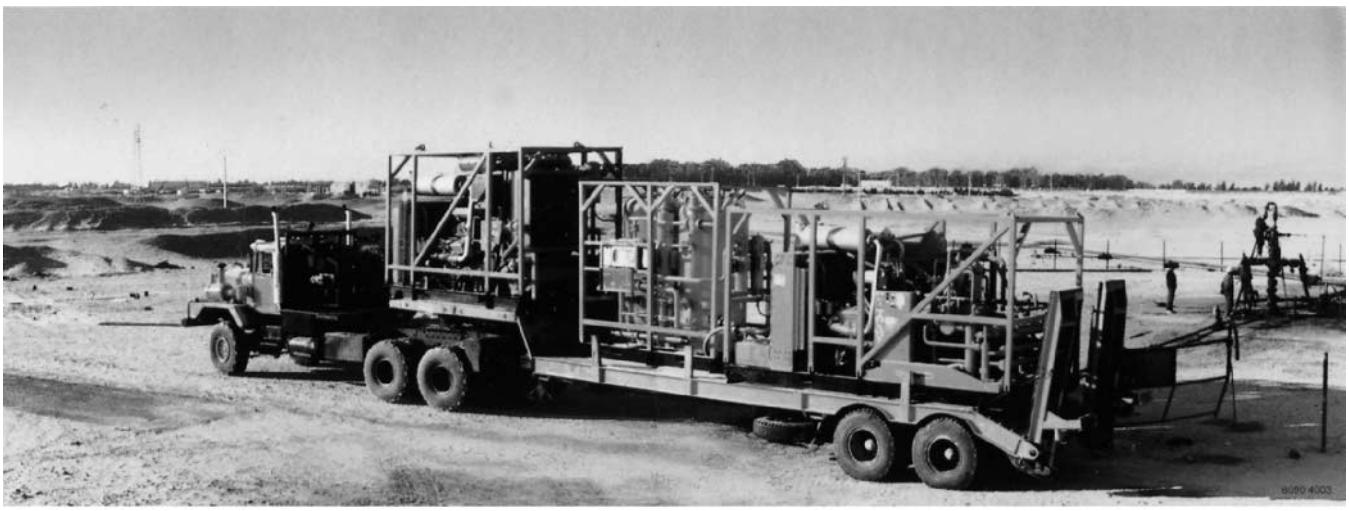


FIG. C-87 Mobile high-pressure nitrogen plant with low-pressure air feed module, PSA module, and diesel-driven nitrogen compressor module for oil-well servicing. (Source: Sulzer-Burckhardt.)

The crank pin bearings are forced-feed lubricated, while all other mechanical components as well as pistons and cylinders are amply lubricated by means of splash lubrication. The crankcase is sealed and vented to the compressor suction side.

Cylinders. The cylinders are made of treated high-quality cast iron and jacketed for ample cooling. The pistons are manufactured of high-quality cast iron and incorporate specially selected cast-iron piston ring combinations. The combined and concentric suction and discharge plate valves are removable as a unit. See Fig. C-92.

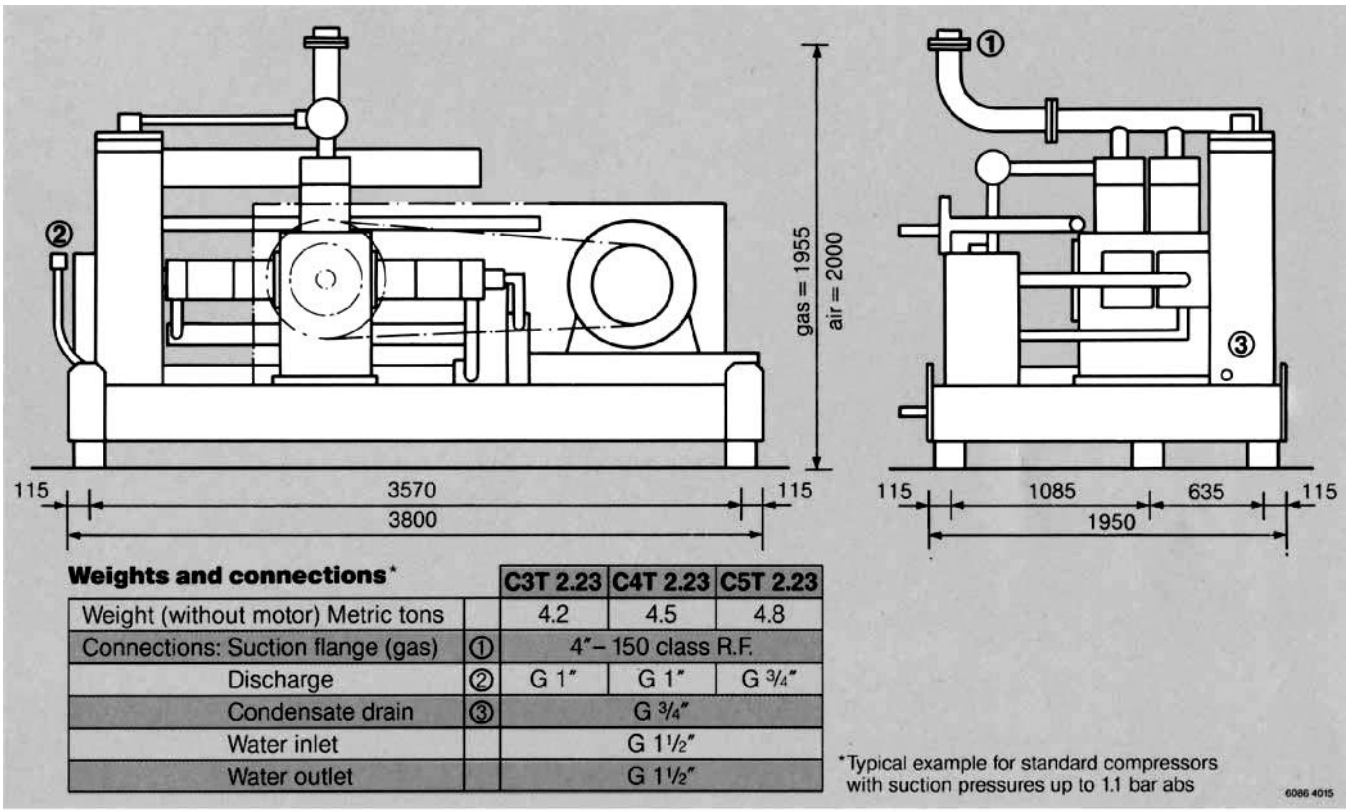
Cooling and piping. Water or another liquid coolant provides for all the compressor cooling requirements. From a manifolded inlet, the coolant is distributed to all critical points such as cylinder jackets by means of high-strength flexible hoses. See Fig. C-93.

The separators located at the outlet of each stage are regularly drained by means of diaphragm valves actuated by a timer-controlled solenoid. The condensate and escaping gas are led from the separators into a condensate receiving tank from where the gas is recycled into the suction line while the condensate can be manually drained off to atmosphere.

Instrumentation. Each compressor stage is equipped with a pressure gauge and safety valve. The gauges are arranged in a compact panel, which also contains the indicating oil-pressure switch. Temperature switches monitor the gas temperature on selected stages. The outlet of the safety valves is piped back to the suction of the compressor. Level gauges for crankcase oil and for condensate receiving tank are also included. See Fig. C-94.

Arrangement and drive. The basic gas compression system consists of a packaged unit with a sturdy steel skid (see Fig. C-95) on which the complete compressor,

Dimensions



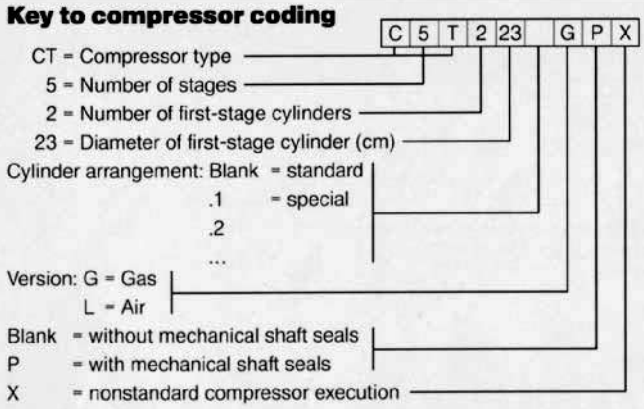
Weights and connections*

		C3T 2.23	C4T 2.23	C5T 2.23
Weight (without motor) Metric tons		4.2	4.5	4.8
Connections: Suction flange (gas)	①	4″ – 150 class R.F.		
Discharge	②	G 1″	G 1″	G ¾″
Condensate drain	③	G ¾″		
Water inlet		G 1½″		
Water outlet		G 1½″		

*Typical example for standard compressors with suction pressures up to 1.1 bar abs

6086 4015

Key to compressor coding



Materials

Crankcase	Nodular cast iron
Crankshaft	Nodular cast iron
Connecting rods	Nodular cast iron
Cylinders	Treated cast iron
Pistons	Cast iron
Piston rings	Special cast iron
Valves	Carbon steel (inner parts Cr-alloyed)
Separators / Condensate tank	Carbon steel
Gas coolers	Carbon steel
Gas piping	Carbon steel
Hoses (gas)	Steel-reinforced polyamide

FIG. C-88 Compressor selection, dimensions, coding, and materials. (Source: Sulzer-Burckhardt.)

flywheel, V-belt drive, coolers, separators, condensate receiving tank, and piping (see also Fig. C-94) are installed. An integral universal motor base is also provided. The skid, designed on the basis of field measurement data and of finite element calculations, is supplied with six vibration dampening elements.

The air compression system has a similar arrangement as above, but with open relief valves and with an automotive-type suction filter instead of the flexible inlet header. See also Figs. C-96 through C-98.

Machines with suction pressures up to 1.1 bar abs
discharge pressures up to 351 bar abs

Piston stroke: 125 mm

Type	Ps bar abs	Pd bar abs	Compressor speed (rpm) 570 Vs (m³/h)	570 P (kW)	760 Vs (m³/h)	760 P (kW)	1030 Vs (m³/h)	1030 P (kW)
C3T 223								
3 stages	1	31	271	41.5	359	57.5	478	83
		37	271	44	359	61	478	87.5
Qw (m³/h)			3		4		5.6	
C3T 223.2								
3 stages	1	41	270	45	357	62	476	88.5
		46	269	46.5	357	64	476	91
Qw (m³/h)			3.2		4.3		5.8	
C3T 323								
3 stages	1	26	*for less volume see C3T 223*				712	121.5
		31					711	127
Qw (m³/h)							8.2	
C4T 223								
4 stages	1	51	269	46.5	357	64.5	477	93.5
		76	269	51	357	71	476	101.5
		101	269	55	357	75.5	476	108.5
Qw (m³/h)			3.8		5.1		7.1	
C4T 320								
4 stages	1	51	*for less volume see C4T 223*		*for less volume see C4T 223*		537	105
		76					537	114
		101					537	121
Qw (m³/h)							7.9	
C4T 323								
4 stages	1	36	*for less volume see C3T 223 or C3T 223.2*				703	129.5
		41					703	133.5
		46					703	136.5
Qw (m³/h)							7.3	
C5T 223								
5 stages	1	151	267	58	356	80.5	475	115.5
		201	267	61.5	356	85	475	122
		251	267	64.5	356	89.5	475	127.5
		301	267	67.5	355	93	475	132
Qw (m³/h)			4.7		6.4		8.7	
C5T 223.1								
5 stages	1	201	262	60.5	349	83.5	466	119
		251	262	63.5	349	87.5	466	124
		301	262	66	349	90.5	466	128.5
		351	262	68	349	93.5	466	132.5
Qw (m³/h)			4.7		6.4		8.7	

Key to symbols

Ps = Suction pressure
Pd = Discharge pressure
Vs = Suction flow rate related to suction conditions
P = Power required at compressor shaft
Qw = Cooling-water consumption

Compressor speed (rpm)
Vs (m³/h)
P (kW)
Qw (m³/h)

Basis

- The above figures relate to air with a relative humidity of 50%, a suction pressure of 1 bar abs and a suction temperature of 20 °C.
- Tolerance on suction flow rate and specific power requirement is ± 5%.
- The values are approximately valid also for gases with similar density.
- Binding data for light gases, e.g. hydrogen, helium or heavy gases, e.g. argon, carbon dioxide, are available upon request.
- Cooling-water temperature inlet / outlet = 20 / 32 °C.

Motor power

- The motor power must be at least 12% above the power requirement at the compressor shaft.
- Ambient temperature higher than 40 °C and an altitude of installation above 1000 m must receive special consideration.

Standard compressor speeds (rpm)

50 Hz: 570, 640, 680, 720, 760, 810, 910, 1030
60 Hz: 610, 650, 680, 770, 820, 860, 920, 970

Intermediate compressor speeds are possible at an extra charge (optional equipment).

FIG. C-89 Main compressor operating data (air and similar gases). (Source: Sulzer-Burckhardt.)

Machines with suction pressures from 1.2 to 16 bar abs discharge pressures up to 351 bar abs

Piston stroke: 125 mm

Type	Ps bar abs	Pd bar abs	Compressor speed (rpm)					
			V̇n (m³/h)				P (kW)	
C5T 223.2			570		760		1030	
5 stages	1.4	251	344	72.5	458	99	616	140
Qw (m³/h)			6.5		8.7		11.8	
C5T 216			570		760		1030	
5 stages	3	251	388	67.5	518	93	699	133
Qw (m³/h)			6.3		8.4		11.5	
C5T 220.1			570		760		1030	
5 stages	2	251	382	74	510	101	687	143.5
Qw (m³/h)			6.8		9		12.3	
C5T 217.1			570		760		1030	
5 stages	2.5	251	384	70	512	96	691	137
Qw (m³/h)			6.5		8.7		11.8	
C4T 123			570		760		1030	
4 stages	2.2	251	280	55	371	74.5	494	105
Qw (m³/h)			4.9		6.6		8.9	
C4T 123.1			*for less volume see C4T 120*		720		960	
4 stages	3.3	251			560	101.5	736	140.5
Qw (m³/h)					9.1		12.2	
C4T 117			570		760		1030	
4 stages	4.5	251	348	55	462	75	622	106.5
Qw (m³/h)			5.1		6.9		9.3	
C4T 119.1			*for less volume see C4T 115*		720		910	
4 stages	6	251			715	108	898	141.5
Qw (m³/h)					10		12.7	
C4T 114			570		760		1030	
4 stages	8.4	251	450	58	599	79.5	806	113.5
Qw (m³/h)			5.6		7.4		10.1	
C4T 113			*for less volume see C3U 208*		760		1030	
4 stages	11	251			633	76	854	108
Qw (m³/h)					7.1		9.6	
C4T 121			570		760		1030	
4 stages	2.8	251	306	56.5	407	76.5	544	108.5
Qw (m³/h)			5.1		6.9		9.3	
C4T 119			570		760		1030	
4 stages	3.8	251	341	57	453	77.5	607	109.5
Qw (m³/h)			5.3		7		9.5	
C4T 120.1			*for less volume see C4T 116*		720		910	
4 stages	5	251			664	105.5	834	138
Qw (m³/h)					9.7		12.3	
C4T 115			570		760		1030	
4 stages	6.5	251	390	55	519	74.5	698	105.5
Qw (m³/h)			5.2		6.9		9.3	
C4T 116.1			*for less volume see C4T 113*		720		910	
4 stages	8.8	251			794	103.5	1000	135.5
Qw (m³/h)					9.7		12.3	
C4T 111			*for less volume see C3U 207*		720		910	
4 stages	14.8	251			641	68.5	807	89.5
Qw (m³/h)					6.5		8.2	
C3T 211			570		720		910	
3 stages	12	251	691	82	869	105.5	1089	137.5
Qw (m³/h)			7.7		9.6		12	
C3T 210			*for less volume see C4T 111*		680		860	
3 stages	14.8	251			940	104	1181	135
Qw (m³/h)					9.6		12	
C4T 120			570		760		1030	
4 stages	3.3	251	323	56.5	429	77	574	109
Qw (m³/h)			5.2		6.9		9.4	
C4T 121.1			*for less volume see C4T 119*		720		960	
4 stages	4	251			593	101	782	140
Qw (m³/h)					9.2		12.2	
C4T 116			570		760		1030	
4 stages	5.6	251	378	55.5	503	75.5	677	106.5
Qw (m³/h)			5.2		7		9.4	
C4T 117.1			*for less volume see C4T 114*		720		910	
4 stages	7	251			719	103.5	905	135.5
Qw (m³/h)					9.6		12.1	
C4T 115.2			*for less volume see C4T 113*		720		910	
4 stages	10.5	251			833	104	1048	137
Qw (m³/h)					9.8		12.4	
C4T 111.1			*for less volume see C3U 207*		720		910	
4 stages	16	251			642	66.5	809	87
Qw (m³/h)					6.3		7.9	
C3T 210.1			570		680		820	
3 stages	16	251	858	90	1021	109	1225	134.5
Qw (m³/h)			8.5		10.1		12	

Key to symbols

Ps = Suction pressure
Pd = Discharge pressure
V̇n = Suction flow rate related to 1.013 bar abs and 20 °C
P = Power required at compressor shaft
Qw = Cooling-water consumption

Compressor speed (rpm)
V̇n (m³/h)
P (kW)
Qw (m³/h)

Basis

- The above figures relate to a typical natural gas (component methane = 95% Vol.), a suction temperature of 20 °C and cooling-water temperature inlet / outlet = 25 / 35 °C.
- Tolerance on suction flow rate and specific power requirement is ± 5%.
- The values are approximately valid also for gases with similar density.
- Binding data for other gases and other suction and discharge pressures up to 351 bar abs are available upon request.

Motor power

- With increasing suction pressure, the motor power must be up to 20% above the power requirement at the compressor shaft.
- Ambient temperature higher than 40 °C and an altitude of installation above 1000 m must receive special consideration.

Standard compressor speeds (rpm)

50 Hz: 570, 640, 680, 720, 760, 810, 910, 1030
60 Hz: 610, 650, 680, 770, 820, 860, 920, 970

Intermediate compressor speeds are possible at an extra charge (optional equipment).

FIG. C-90 Main compressor operating date (gases). (Source: Sulzer-Burckhardt.)

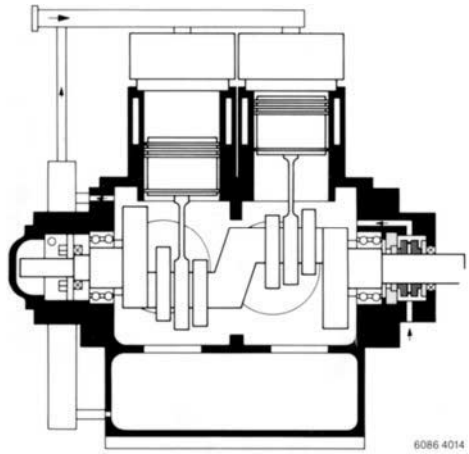


FIG. C-91 In pressure-tight execution (optional) the crankcase is equipped with oil-cooled double mechanical seals. (Source: Sulzer-Burckhardt.)

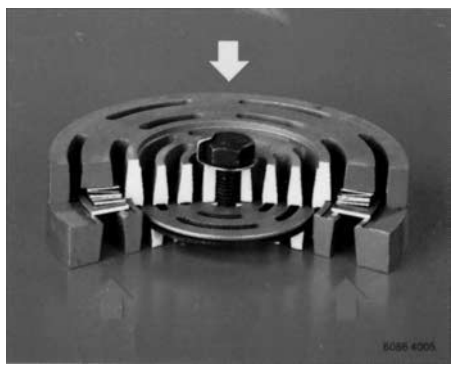


FIG. C-92 Typical valve (CT compressor type). (Source: Sulzer-Burckhardt.)

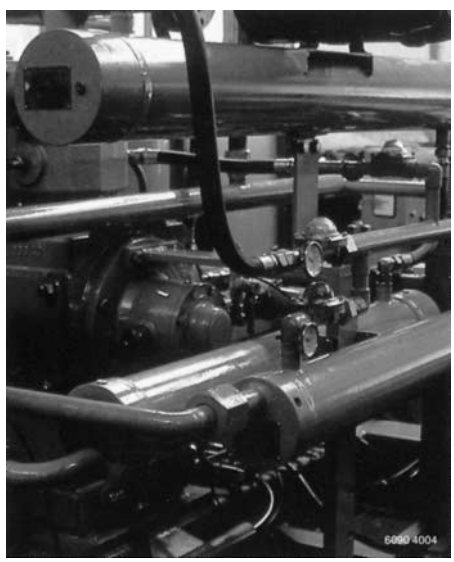


FIG. C-93 Typical pressure gauge and safety valve on each compressor stage. (Source: Sulzer-Burckhardt.)

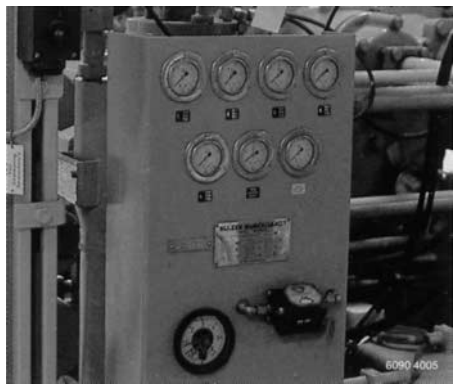


FIG. C-94 Standard panel, including optional contamination indicator for oil filter and gauge with pressure limit switch for sequential condensate drain system. (Source: Sulzer-Burckhardt.)

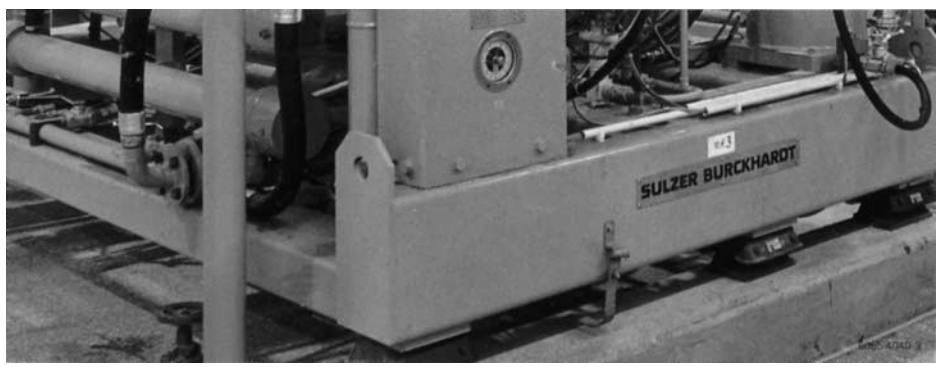


FIG. C-95 Typical compressor skid. (Source: Sulzer-Burckhardt.)

Standard supply scope. This may differ according to customers' specifications.

Compressor crank mechanism

- Crankcase with crankshaft seal for suction pressures up to 1.1 bar abs and vent line to suction or crankcase with mechanical seals for suction pressures from 1.2 to 16 bar abs (see Figs. C-96 through C-98).
- Two crankcase purging gas valves (gas compression only)
- Oil pump (crankshaft-driven), filter, pressure gauge, and level sight glass

Gas stream

- Flexible hose on suction or automatic-type air filter
- Interconnecting gas piping from first stage to outlet separator on last stage
- Shell and tube gas cooler and separator after each stage
- Automatic condensate drain consisting of:
  - diaphragm valve on each separator
  - condensate receiving tank with level sight glasses
  - manual drain valve on the condensate tank
  - return line condensate tank-suction line
- Pressure gauge and relief valve after each stage
- Flexible hose on discharge with nonreturn valve and compression fitting

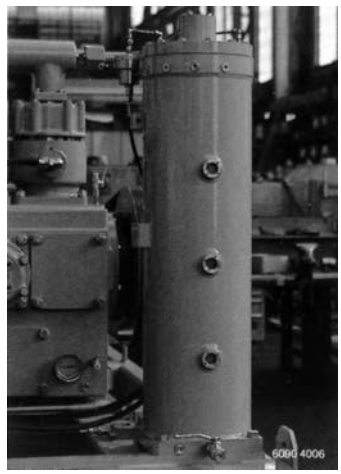


FIG. C-96 Standard condensate receiving tank. (Source: Sulzer-Burckhardt.)

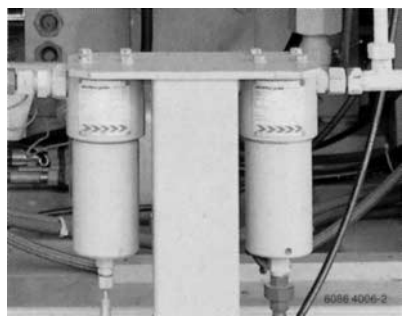


FIG. C-97 Filter system. (Source: Sulzer-Burckhardt.)



FIG. C-98 Remote control. (Source: Sulzer-Burckhardt.)

Water system

- Interconnecting water piping between inlet manifold, jackets, coolers, and outlet manifold, by means of high-strength flexible hoses
- Control, vent, and drain valves
- Flow sight glasses
- Temperature gauges

Electrical equipment

- Solenoid valve for condensate drain control with separate time relay
- Temperature switches on second and last stages
- Oil pressure switch

Package

- Skid (large enough to accommodate motor) mounted on six vibration dampening elements
- Gauge panel
- Flywheel, motor pulley, belts, drive guard
- Special tools

Test

- Mechanical run
- Standard performance test with air (DIN 1945/ISO 1217)

Options

- Pressure-tight execution for suction pressures from 17 to 100 bar abs
- Direct drive
- Oil refill system to permit oil refill during operation
- Noise-retaining weatherproof housing
- Suction filter 10 μ
- Coalescence and activated carbon filter to remove oil downstream of compressor, to get oil aerosol content down to 0.2 ppm (weight) approximately
- Pressure-maintaining valve
- Additional instrumentation
- Automatic condensate tank drain (level controlled)
- Sequential condensate drain system to ensure that each separator drains individually
- Crankcase heater (at or below +5°C ambient temperature)
- Additional manual condensate drain valves for each separator
- Closed execution for diaphragm valves for condensate drain (included for H_2 service)
- Terminal box on or beside compressor skid
- Remote control box on or beside compressor skid
- Control cabinet (including motor starter) (Fig. C-99)

Nonlubricated sealing system in "LABY®" compressors

Significant inventions often depend on simple principles that seem self-evident in hindsight. This is true of the labyrinth sealing technique. An extremely large number of throttling points provides the sealing effect around pistons and piston rods. No contact seals are used. See Figs. C-100 and C-101.

Whereas plastic sealing rings depend on permanent mechanical friction for efficient performance, the labyrinth principle embodies an extremely small

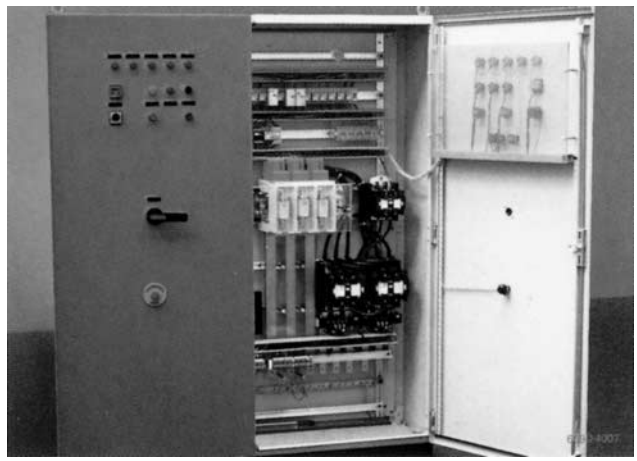


FIG. C-99 Control cabinet. (Source: Sulzer-Burckhardt.)

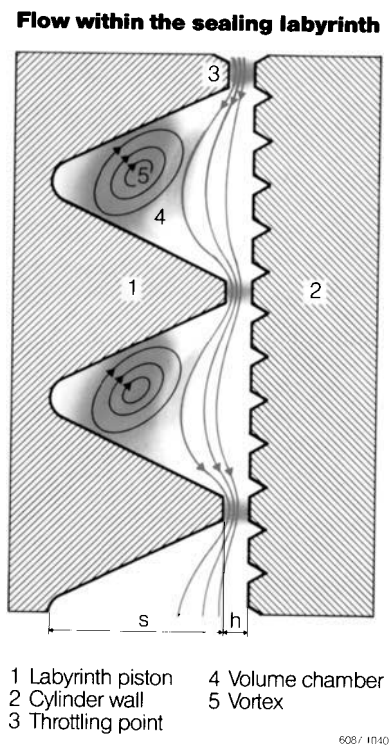


FIG. C-100 How a labyrinth seal works. (Source: Sulzer-Burckhardt.)

clearance between sealing element and counterpart. This is the key to the durability, reliability, and availability of this compressor type and, therefore, to its economic operation.

Above all, the unique labyrinth sealing technique is employed for applications where no lubricants are allowed in the cylinder and where no abrasion particles are accepted in the process gas. This is particularly true for oxygen compression, where safety is the most important aspect. At the other extreme it is also employed

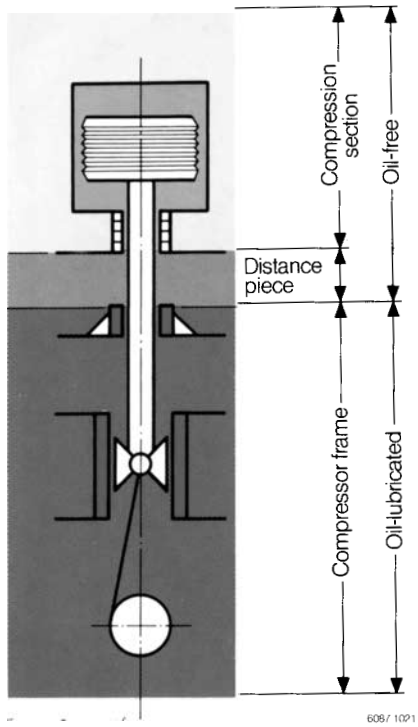


FIG. C-101 Typical section showing oil-lubricated and oil-free zones. (Source: Sulzer-Burckhardt.)

for applications where the process gas is heavily contaminated with impurities, such as polymerization products or other very small and hard particles. They have effectively no influence on the labyrinth seal performance, compressor reliability, wear rate, and maintenance intervals.

Piston and piston rod are guided by the crosshead and the guide bearing, which are located in the oil-lubricated crankcase. Both guiding elements are made of metal and are oil lubricated, thus ensuring a precisely linear operation of the labyrinth piston as well as an extremely long life of the piston/piston rod guiding system.

The distance piece separates the gas compressing section from the oil-lubricated crankcase.

LABY® design options and features

A large variety of standard labyrinth-piston compressors, with many additional cylinder blocks, is available with suction volumes up to 11,000 m^3/h and discharge pressures exceeding 300 bar. See Figs. C-102 through C-104 for various types. Design features of the totally closed "K"-type compressor with gas- and pressure-tight crankcase are illustrated in Fig. C-103C and D. See these figures and Fig. C-105.

Common features

The labyrinth piston (see Fig. C-106)

- May be double- or single-acting (depending on application)
- Seals by repeated gas throttling

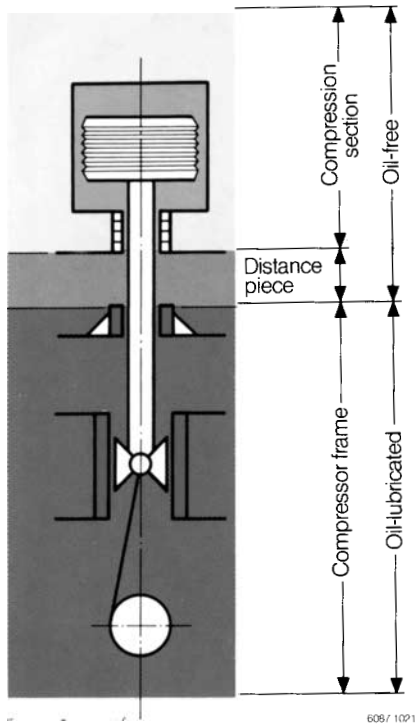


FIG. C-101 Typical section showing oil-lubricated and oil-free zones. (Source: Sulzer-Burckhardt.)

for applications where the process gas is heavily contaminated with impurities, such as polymerization products or other very small and hard particles. They have effectively no influence on the labyrinth seal performance, compressor reliability, wear rate, and maintenance intervals.

Piston and piston rod are guided by the crosshead and the guide bearing, which are located in the oil-lubricated crankcase. Both guiding elements are made of metal and are oil lubricated, thus ensuring a precisely linear operation of the labyrinth piston as well as an extremely long life of the piston/piston rod guiding system.

The distance piece separates the gas compressing section from the oil-lubricated crankcase.

LABY® design options and features

A large variety of standard labyrinth-piston compressors, with many additional cylinder blocks, is available with suction volumes up to 11,000 m^3/h and discharge pressures exceeding 300 bar. See Figs. C-102 through C-104 for various types. Design features of the totally closed "K"-type compressor with gas- and pressure-tight crankcase are illustrated in Fig. C-103C and D. See these figures and Fig. C-105.

Common features

The labyrinth piston (see Fig. C-106)

- May be double- or single-acting (depending on application)
- Seals by repeated gas throttling

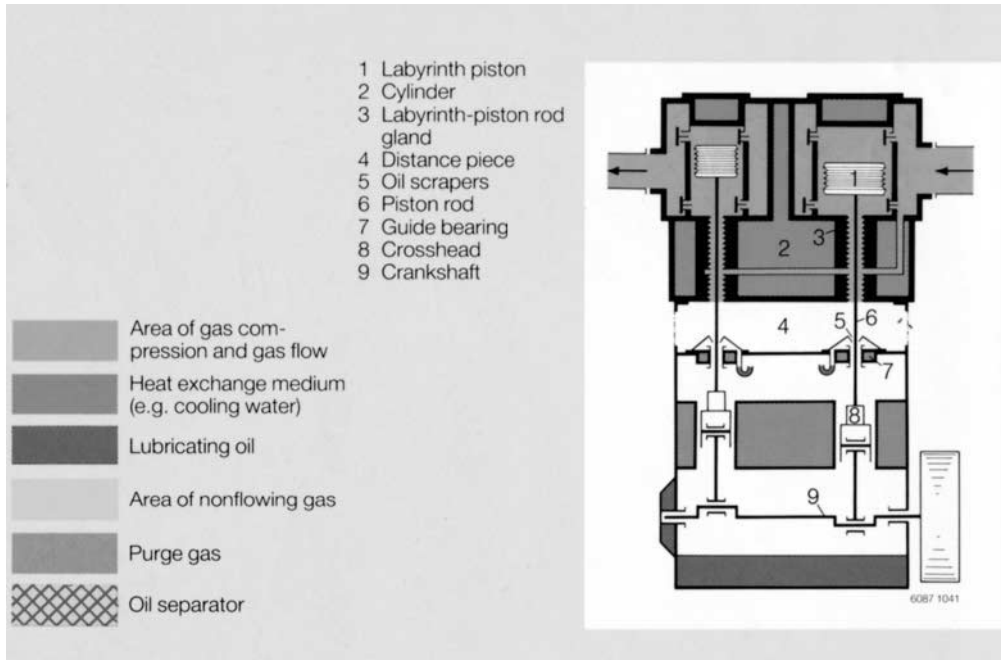


FIG. C-102A With open distance piece: This standard compressor is equipped with an open distance piece and a nonpressurized crankcase. It is used for compression of gases, where a strict separation between cylinder and crankcase is essential and where process gas is permitted in the open distance piece (e.g., for O_2, N_2, CO_2, process air, etc., generally in the industrial gas industry). (Source: Sulzer-Burckhardt.)

- Consists of a very small number of parts
- Is made of solid metal without any plastic material
- Avoids permanent mechanical friction
- Avoids contamination or fouling of the process gas
- Guarantees extremely long sealing element life and assures low maintenance cost
- Accepts a wide range of operating temperatures (–160 to +270°C and higher)
- Is insensitive to impurities in the gas
- Ensures unexceeded reliability in oxygen service

The compressor valve (see Fig. C-107)

- Helps to achieve ideal combinations of cylinder design, valve size, and compressor plant components
- Ensures high reliability and availability of the compressor
- Embodies frictionless guided plates with very low lift and extremely good fatigue properties
- Consists of identical parts for suction and discharge side, but special design features prevent inadvertent wrong assembly of valves into the cylinder
- Comprises stationary parts, such as valve seat and stroke limiter not being cast, but machined out of special stainless steel

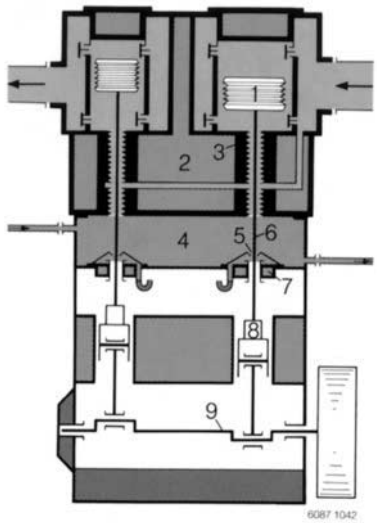


FIG. C-102B With closed and purged distance piece: The distance piece of the standard open-type compressor is closed and purged with nitrogen, air, or another suitable gas. It is used for compression of gases, where a strict separation between cylinder and crankcase is essential and where no process gas may leak to the surroundings or no ambient air may enter the distance piece (e.g., for weather protection). (Source: Sulzer-Burckhardt.)

- Incorporates dynamically moving parts, such as valve plates and damper plates, manufactured according to most modern techniques and made of special stainless steel
- Includes aerodynamically optimal shapes with low pressure drop

The piston rod gland (see Fig. C-108)

- Features radially floating and self-centering labyrinth sealing rings made of graphite
- Comprises stainless-steel gland chambers for the sealing rings
- Incorporates a leak-gas collecting chamber at the lower end for feeding the leak gas, where possible, back to the suction-side first stage
- Allows, if necessary, for specially designed applications with several connections

The piston rod guide bearing (see Fig. C-109)

- Is available in cooled or uncooled application
- Is available with a replaceable bush and is splash-lubricated
- Is combined with the oil scrapers and designed to exclude oil from the distance piece, the piston rod gland area, or the cylinder section. It eliminates the necessity for additional oil-vapor removal equipment.

The crankshaft seal for open-type compressors (see Fig. C-110)

- Is equipped with an oil slinger ring and, except on the smaller compressors, with an additional packing ring
- Is designed to exclude dust and dirt from the crankcase and to provide an oil-tight crankshaft passage through the crankcase wall

Click here for next page

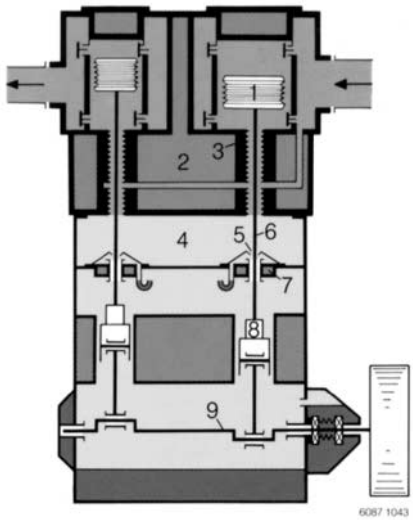


FIG. C-102C With gas-tight crankcase and mechanical crankshaft seal: The distance piece of the standard open-type compressor is closed, and the crankshaft bears a mechanical gas-tight seal where it passes through the crankcase wall. This design is used for compression of gases which are compatible with the lubricating oil (e.g., for hydrocarbon gases, CO, He, H_2, Ar, etc.) and where no process gas may leak to the surroundings. The suction pressure is limited by the design pressure of the crankcase. (Source: Sulzer-Burckhardt.)

The crankshaft seal for compressors with gas-tight crankcase (see Fig. C-111)

- Is equipped with a mechanical sealing system completely immersed in lubricating oil
- Prevents oil from draining into the crankcase during standstill periods
- Incorporates an additional sealing ring to exclude dust and dirt from the shaft seal area
- Provides and ensures a gas- and oil-tight crankshaft passage through the crankcase wall

LABY® research and development projects

The compressor valves. Valves installed in a reciprocating compressor have a tough life. They have to open and close automatically once every crankshaft revolution, quickly and reliably under severe temperature and pressure conditions. Troubles and excessive wear or losses are avoidable if valve quality as well as the match of valve, compressor, and operating conditions are optimal. See Fig. C-112.

Painstaking design and years of feedback from operations have raised compressor valves to a very high standard in terms of material, manufacturing technology, and aerodynamic shape. Nevertheless, to retain an OEM's lead in valve technology, continue to invest in valve research. There are further possibilities to reduce stress peaks in dynamically loaded parts, to optimize the aerodynamic characteristics, to influence the movement of the dynamic parts, to introduce improved materials and manufacturing techniques.

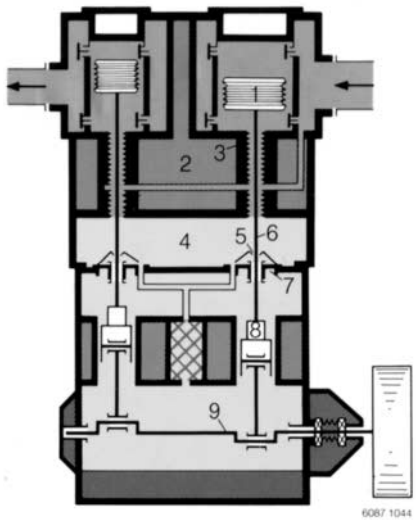


FIG. C-102D With gas- and pressure-tight crankcase and mechanical crankshaft seal: This standard compressor is equipped with a closed single-piece crankcase designed for a gas pressure of 15 bar or higher. All openings are closed and sealed with o-rings. The crankshaft bears a mechanical gas-tight seal where it passes through the crankcase wall. Since the crankcase is filled with process gas, this machine is used to compress gases that are compatible with the lubricating oil and where no process gas may leak to the surroundings. Suction pressure may range between subatmospheric and crankcase design pressure. This machine finds its applications in closed cycles, for hydrocarbon gases, refrigerants, VCM, CO, N_2, CO_2, He, H_2, Ar, etc. (Source: Sulzer-Burckhardt.)

The sealing labyrinth. This component is the subject of ongoing research. Considerable time and effort is invested in exploring the flow behavior of gases in oscillating sealing labyrinths and comparing the results with simulated computer calculations. Better understanding of the influence of piston speed, labyrinth shape, labyrinth clearance, and other factors on compressor performance emerges. See Figs. C-111 through C-113.

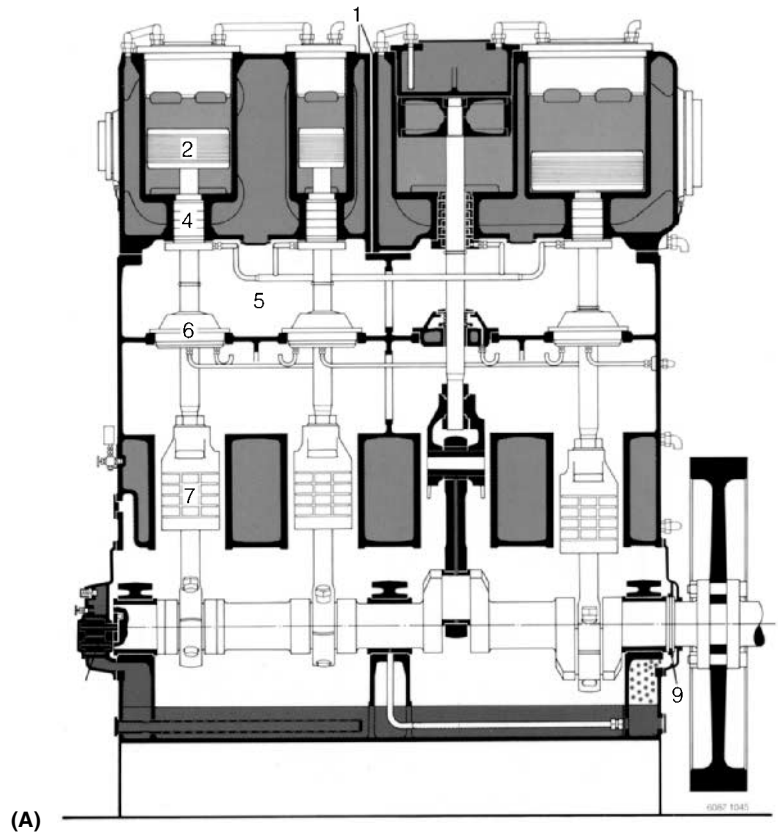
Dynamic crank throw behavior. This remains a subject of investigation. Compressor parts are not entirely rigid, but rather flexible, and may oscillate or vibrate during operation. It is important to have fundamental and detailed knowledge of means to eliminate or suppress undesired movements. See Figs. C-114 and C-115.

Acoustic calculations. Nowadays these are important. With computer technology one can choose between both digital and analog studies according to API 618.

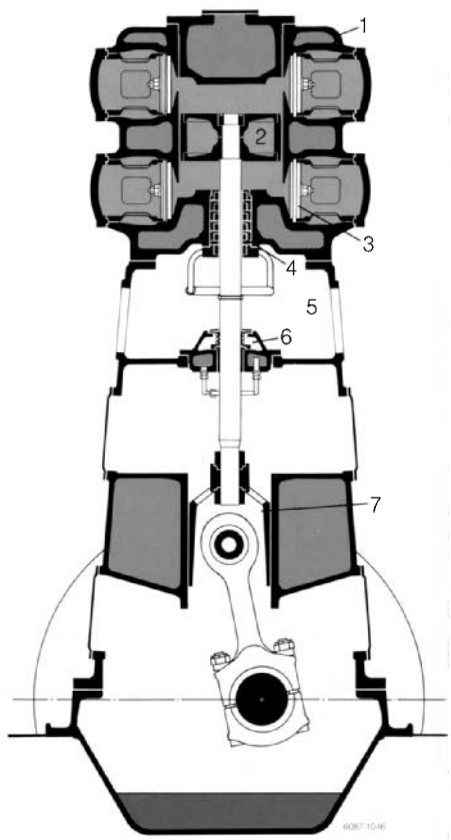
New design materials. For cylinders, pistons, piston rods, and other parts, new design materials are under consideration to meet new requirements from customers or to employ labyrinth-piston compressor in new applications.

Quality control (QC) for reciprocating compressors. QC is regularly adapted to possible new requirements of the market as well as to new measuring and monitoring methods to obtain optimal quality.

Quality inspection is performed during the manufacturing process after each important step. All pressure-stressed parts, such as cylinders, cylinder covers,

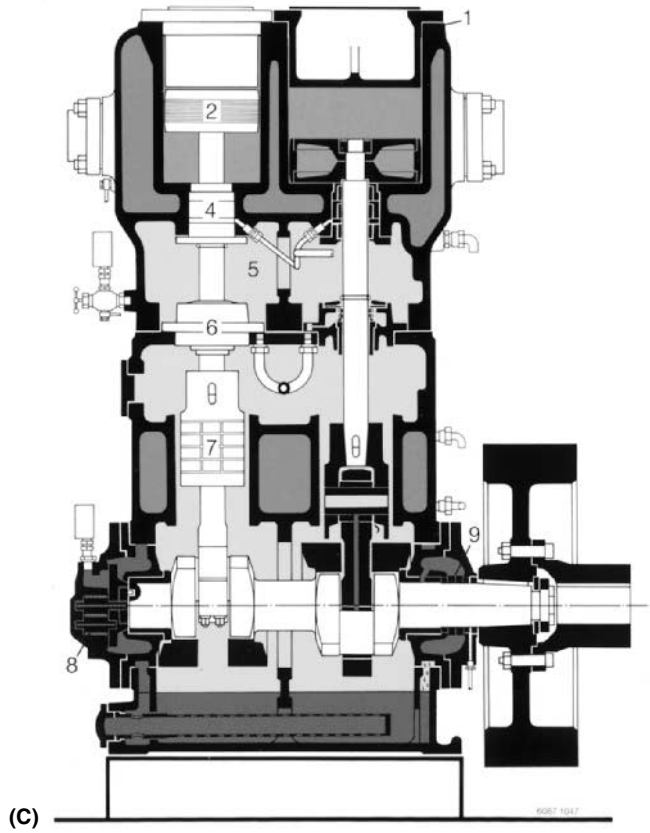


(A)

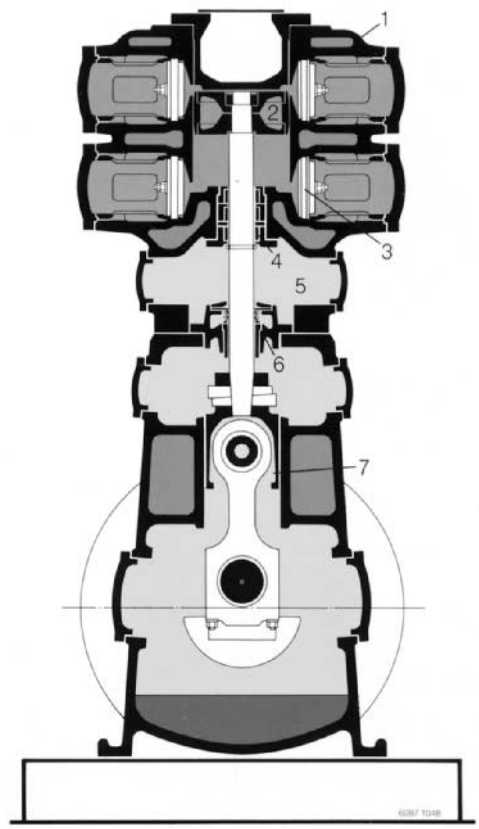


(B)

FIG. C-103A, B Design features of D- and E-type compressors with open distance piece. (Source: Sulzer-Burckhardt.)



(C)

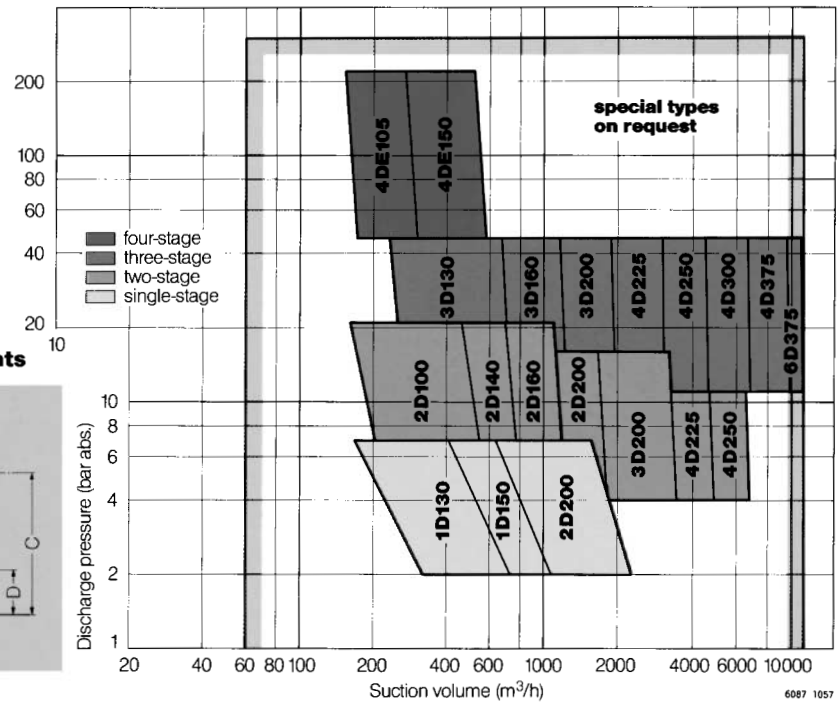


(D)

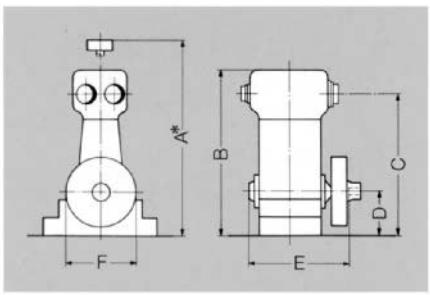
FIG. C-103C, D Design features of the totally closed K-type compressor with gas- and pressure-tight crankcase. (Source: Sulzer-Burckhardt.)

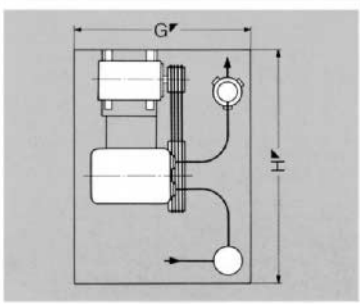
Performance range based on diatomic gases with molecular mass between 15 and 50 kg/kmol

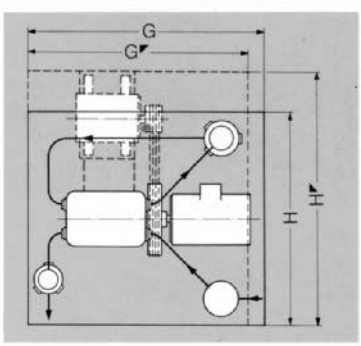
Volume flow based on suction condition 1 bar abs, 20 °C

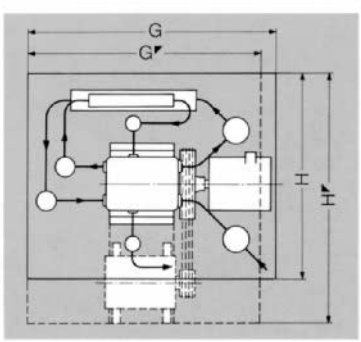


Principal dimensions and weights









6087 1055

Single-stage compressors

Type	Dimensions A* mm	B mm	C mm	D mm	E mm	F mm	Compr. weight kg	Foundation** G* mm	H* mm
1D130-1A	3465	2385	2010	500	765	Ø 910	1900	2500	3300
1D150-1A	3710	2570	2185	550	870	Ø1000	2500	2800	3600
2D200-1A	4450	3090	2597	700	1510	Ø1300	5200	3200	4200

Two-stage compressors

Type	Dimensions A* mm	B mm	C mm	D mm	E mm	F mm	Compr. weight kg	Foundation** G mm	H mm	Foundation** G* mm	H* mm
2D100-2A	2740	1975	1585	430	970	Ø 770	1800	3500	2500	2800	3200
2D140-2A	3160	2305	1845	500	1110	Ø 900	2500	4000	2800	3200	3800
2D160-2A	3510	2550	2060	550	1220	Ø1000	3200	4500	3200	3500	4400
2D200-2A	4400	3090	2600	700	1510	Ø1200	5600	4500	3500	–	–
3D200-2A	4230	3025	2530	700	2200	Ø1200	7200	6000	4000	–	–
4D225-2A	4500	3210	2695	700	2605	Ø1200	11700	7500	5000	–	–
4D250-2B	5000	3670	3100	800	3005	Ø1500	15800	8500	5500	–	–

Three-stage compressors

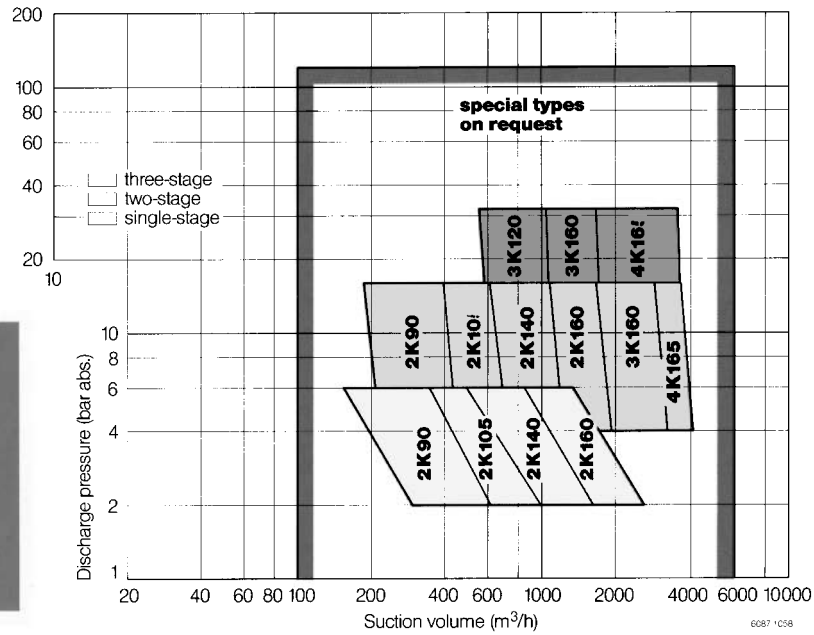
Type	Dimensions A* mm	B mm	C mm	D mm	E mm	F mm	Compr. weight kg	Foundation** G mm	H mm	Foundation** G* mm	H* mm
3D130-3A	3300	2325	1910	500	1345	Ø 900	3500	4500	3200	4000	3800
3D160-3A	3560	2825	2110	600	1470	Ø1000	4500	5000	4000	4300	4800
3D200-3A	4230	3010	2530	700	2200	Ø1200	7200	7000	4500	–	–
4D225-3A	4500	3210	2695	700	2605	Ø1300	11600	8000	5000	–	–
4D250-3A	5000	3670	3105	800	3005	Ø1500	16400	9000	6000	–	–
4D300-3A	6000	4315	3595	1000	3515	Ø1800	24600	10000	7000	–	–
4D375-3A	6750	4950	4075	1050	4210	Ø2000	39800	11000	8000	–	–

* Required height to remove piston and piston rod ** Foundation dimensions are for estimates only

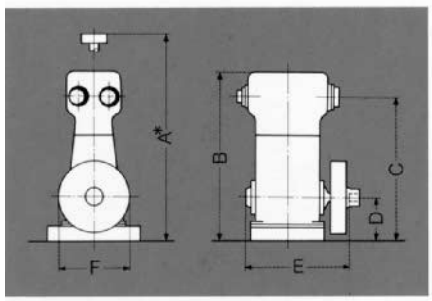
FIG. C-104 Dimensions and performance parameters for D- and E-type compressors. (Source: Sulzer-Burckhardt.)

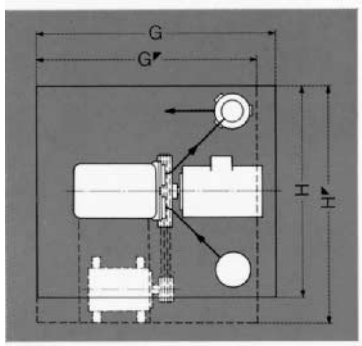
Performance range based on diatomic gases with molecular mass between 15 and 50 kg/kmol

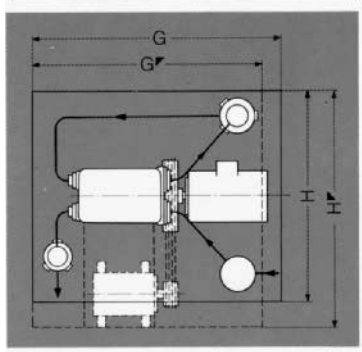
Volume flow based on suction condition 1 bar abs, 20°C

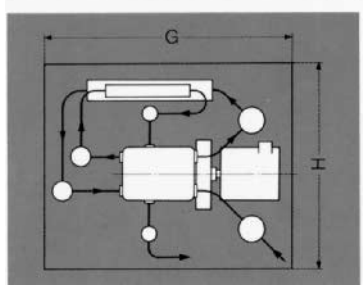


Principal dimensions and weights









Single-stage compressors

Type	Dimensions A* mm	B mm	C mm	D mm	E mm	F mm	Compr. weight kg	Foundation** G mm	H mm	Foundation** G* mm	H* mm
2K 90-1A	2260	1655	1420	520	1010	Ø 600	1600	3200	2500	2800	3200
2K105-1A	2640	1895	1615	570	1165	Ø 660	2300	3800	2800	3200	3800
2K140-1A	3180	2325	1985	650	1365	Ø 900	4000	4200	3200	3500	4400
2K160-1A	3470	2635	2155	710	1520	Ø1000	5500	5000	3500	–	–

Two-stage compressors

Type	Dimensions A* mm	B mm	C mm	D mm	E mm	F mm	Compr. weight kg	Foundation** G mm	H mm	Foundation** G* mm	H* mm
2K 90-2A	2270	1710	1420	520	1000	Ø 600	1700	3200	2800	3000	3500
2K105-2A	2640	1915	1615	570	1165	Ø 660	2500	4200	3000	3800	4000
2K120-2A	3150	2360	1870	630	1250	Ø 900	2600	4500	3500	3800	4500
2K140-2A	3200	2360	1985	650	1365	Ø 900	4300	4500	3500	4000	4500
2K160-2A	3470	2635	2155	710	1520	Ø1000	5900	5000	4000	–	–
3K160-2C	3580	2685	2210	710	2175	Ø1000	8200	6000	4200	–	–
4K165-2C	3760	2800	2320	760	2975	Ø1000	14100	7500	4500	–	–

Three-stage compressors

Type	Dimensions A* mm	B mm	C mm	D mm	E mm	F mm	Compr. weight kg	Foundation** G mm	H mm
3K120-3E	3100	2290	1900	630	1750	Ø 900	5700	5000	4000
3K160-3E	3150	2630	2170	670	2175	Ø1000	8200	7000	4800
4K165-3E	3760	2800	2320	760	2975	Ø1000	14100	8000	5000

6087 1056

* Required height to remove piston and piston rod ** Foundation dimensions are for estimates only

FIG. C-105 Dimensions and performance parameters for K-type compressor. (Source: Sulzer-Burckhardt.)

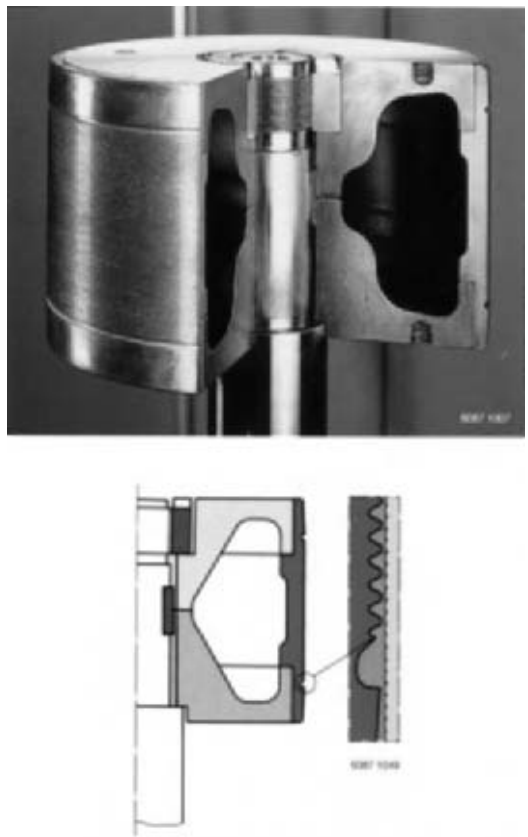


FIG. C-106 The labyrinth piston. (Source: Sulzer-Burckhardt.)

crankcases, oil-pump casings, and others, are hydraulically tested, and leakage tests are made on the assembled gas- and pressure-tight compressors. During assembly, all bearing and piston clearances are measured and recorded on request, and the alignment is checked.

Smaller compressor units are subject to a mechanical running test as well as to running-in of the pistons. A barring-over test is made on larger, completely assembled machines.

Test and material certificates are provided on request.

"LABY'S" for liquefied natural gas service (low gas temperature application)

See Fig. C-116A, which shows the complete diagramatic assembly of compressor and accessories in liquefied natural gas (LNG) service. Figure C-117A is a photograph of the plant. Figure C-116B depicts the operating temperatures involved in a specific application. An LNG boil-off compressor has to cope with a variety of basic physical problems for which a product designed to normal standards would be inadequate. Two application aspects are of special interest in this context.

Exposure to cryogenic temperatures. LNG at barometric pressure boils off at –160°C. This temperature is well below the limit where some of the common engineering



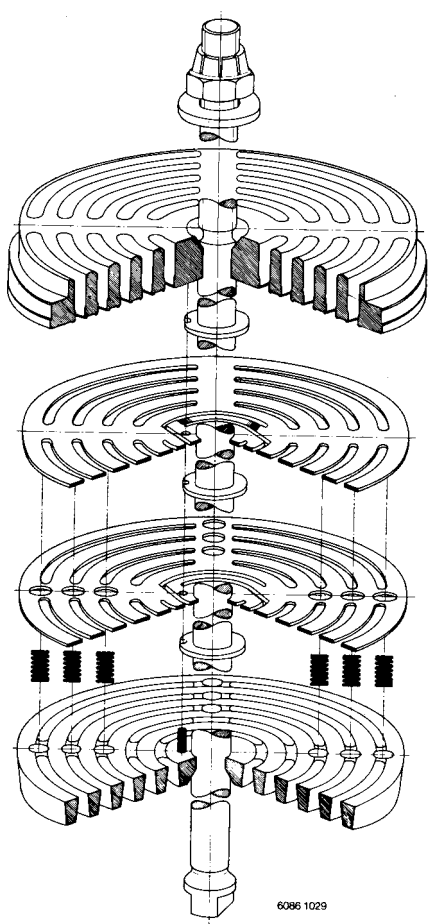


FIG. C-107 The compressor valve. (Source: Sulzer-Burckhardt.)

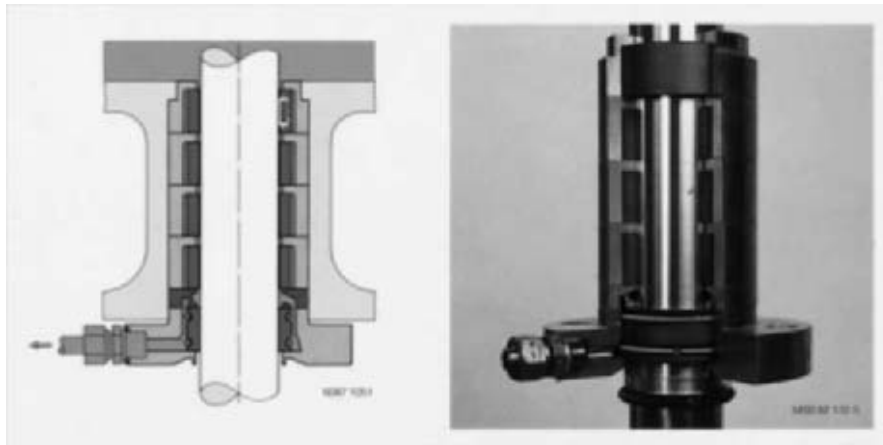


FIG. C-108 The piston rod gland. (Source: Sulzer-Burckhardt.)

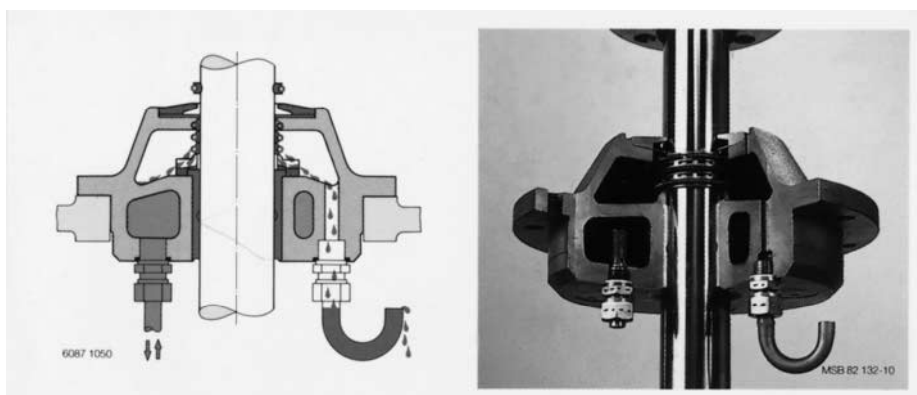


FIG. C-109 The piston rod guide bearing. (Source: Sulzer-Burckhardt.)

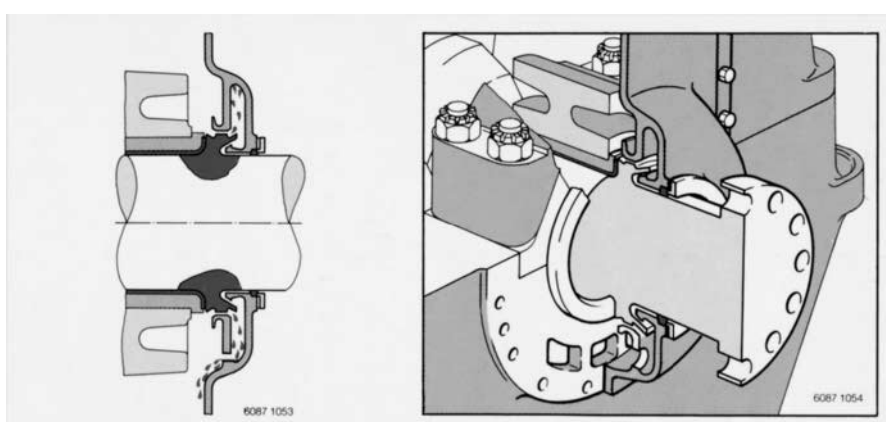


FIG. C-110 The crankshaft seal for open-type compressors. (Source: Sulzer-Burckhardt.)

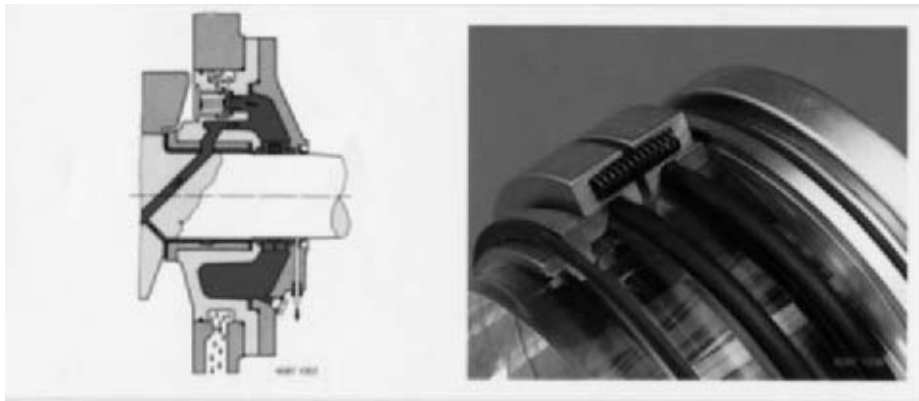
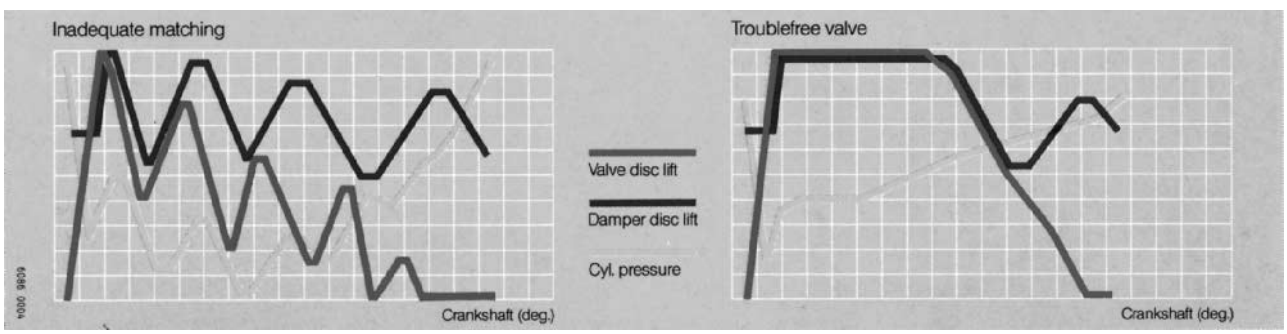


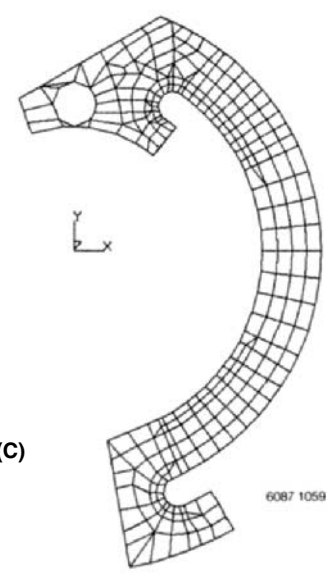
FIG. C-111 The crankshaft seal for compressors with gas-tight crankcase. (Source: Sulzer-Burckhardt.)



(A)



(B)



(C)



(D)

FIG. C-112 Valve design troubleshooting: stress graphs and computer modeling. (Source: Sulzer-Burckhardt.)

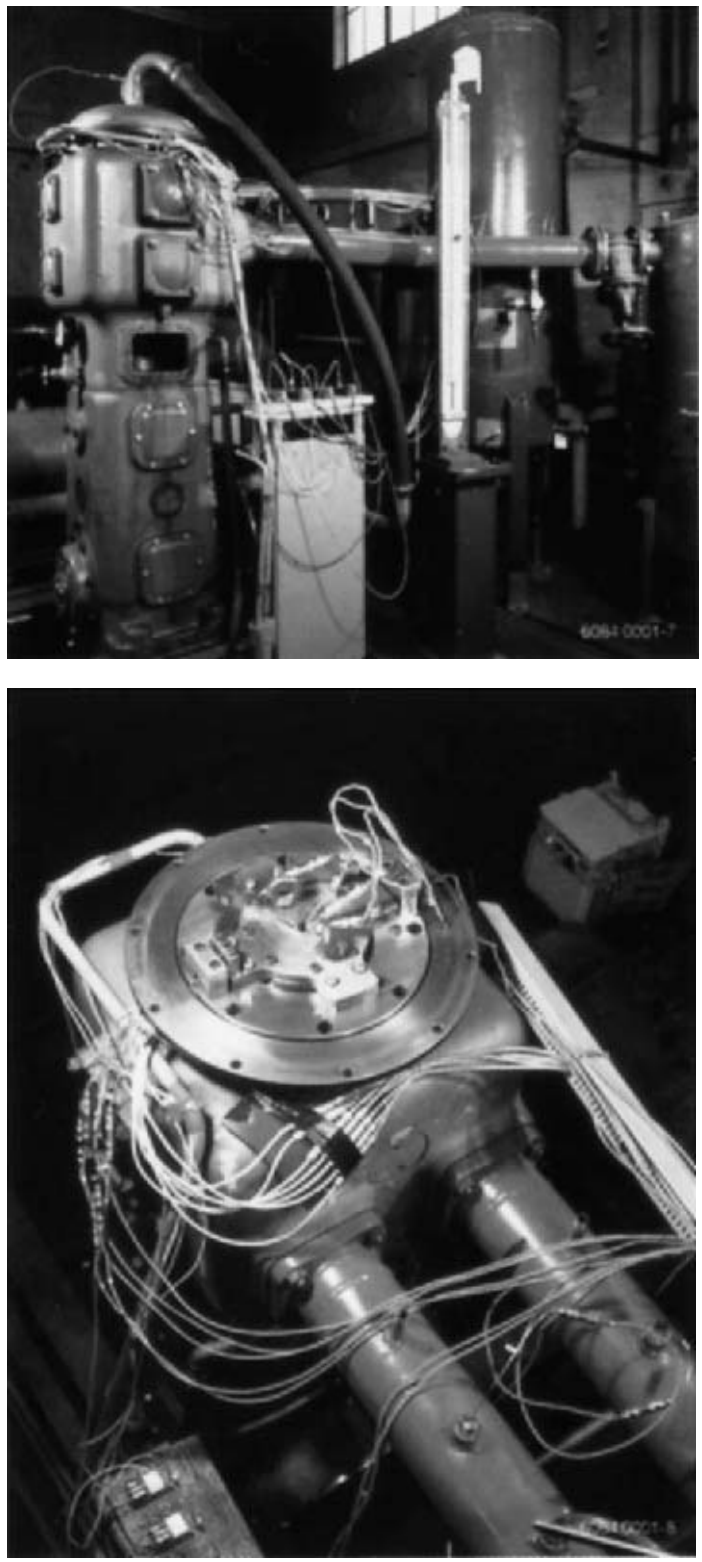


FIG. C-113 Sealing labyrinth R&D rig. (Source: Sulzer-Burckhardt.)

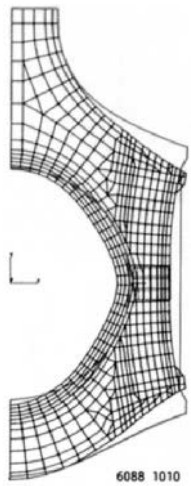


FIG. C-114 Stress map taken during R&D on the dynamic crank. (Source: Sulzer-Burckhardt.)

materials alter their properties. As an example, note the loss of ductility of most unalloyed carbon steels within a temperature span from 0°C to about –50°C. Figure C-117B shows the modes of impact transitions of C-steels and of Ni-alloyed nodular iron.

Bone dry gas. Natural gas in the form of boil-off is virtually free from water vapor as the dew point is as low as –160°C. Experience dictates to what extent moisture in a tribological system is an important parameter. Together with a number of other factors, it has a distinct bearing on wear rates under nonlubricated conditions. Those who decide to employ dry-running self-lubricating materials for piston rings must accept their mechanical and thermal constraints under bone dry running conditions. The operator must consequently set the stroke and speed of his machine in accordance with the gas conditions, so that the wear rate of the sealing and guiding elements can be held within acceptable limits. Already the initial choice of the dry-running material is itself subject to error because the designer is faced with a multitude of available material selections.

The operator is more free to optimize the design of individual parts of the compressor when the labyrinth principle is employed with the following main features:

- Avoidance of permanent mechanical friction
- Ability to use materials with known, easily certifiable qualities
- Simple design of the elements exposed to the process gas

Design and material selection of pistons and cylinders (application case study). One end-user installed a labyrinth piston compressor for handling LNG boil-off gas in a terminal and ran to 75,000 h quite inexpensively. Some end-users experience close or better time between overhauls (TBOs). See Table C-11.

The process data for the compressor in this application are as follows:

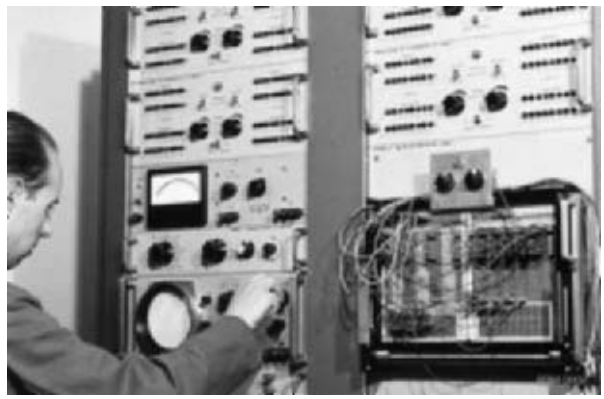




FIG. C-115 Experimental rigs for R&D work. (Source: Sulzer-Burckhardt.)

Gas		CH_4 (98%) + N_2 (2%)				
Suction	1st	1.036	bar	–90	to	–160°C
Discharge	1st	5.2	bar	+25	to	–53°C
Suction	2nd	5.2	bar	+25	to	–53°C
Discharge	2nd	13.6	bar	+38	to	+102°C
Suction	3rd	13.6	bar	+38	to	+48°C
Discharge	3rd	23.4	bar	+88	to	+160°C

Material selection for cylinders and pistons. The above data were the guideline for the materials selected for cylinders, labyrinth pistons, and other components of the machine. Figure C-118 illustrates a double-acting laby piston. The absence of tribological restrictions gained by using labyrinth sealing techniques allowed for the choice of the best-suited metals for the key components in each individual stage. For the 1st stage cylinders exposured to the lowest temperatures, the choice of GGG Ni35 was made. This is a nodular cast iron containing 35 percent nickel, also known under the trade name of Ni Resist D5. This alloy simultaneously exhibits remarkable ductility at low temperatures and one of the lowest thermal expansion coefficients known in metallurgy (Fig. C-119). The corresponding pistons were made of nickel-alloyed cast iron with laminar graphite.

Table C-12 indicates the outstanding thermal shock behavior of GGG Ni 35 in relation to other candidate materials. This is valuable, especially under transient

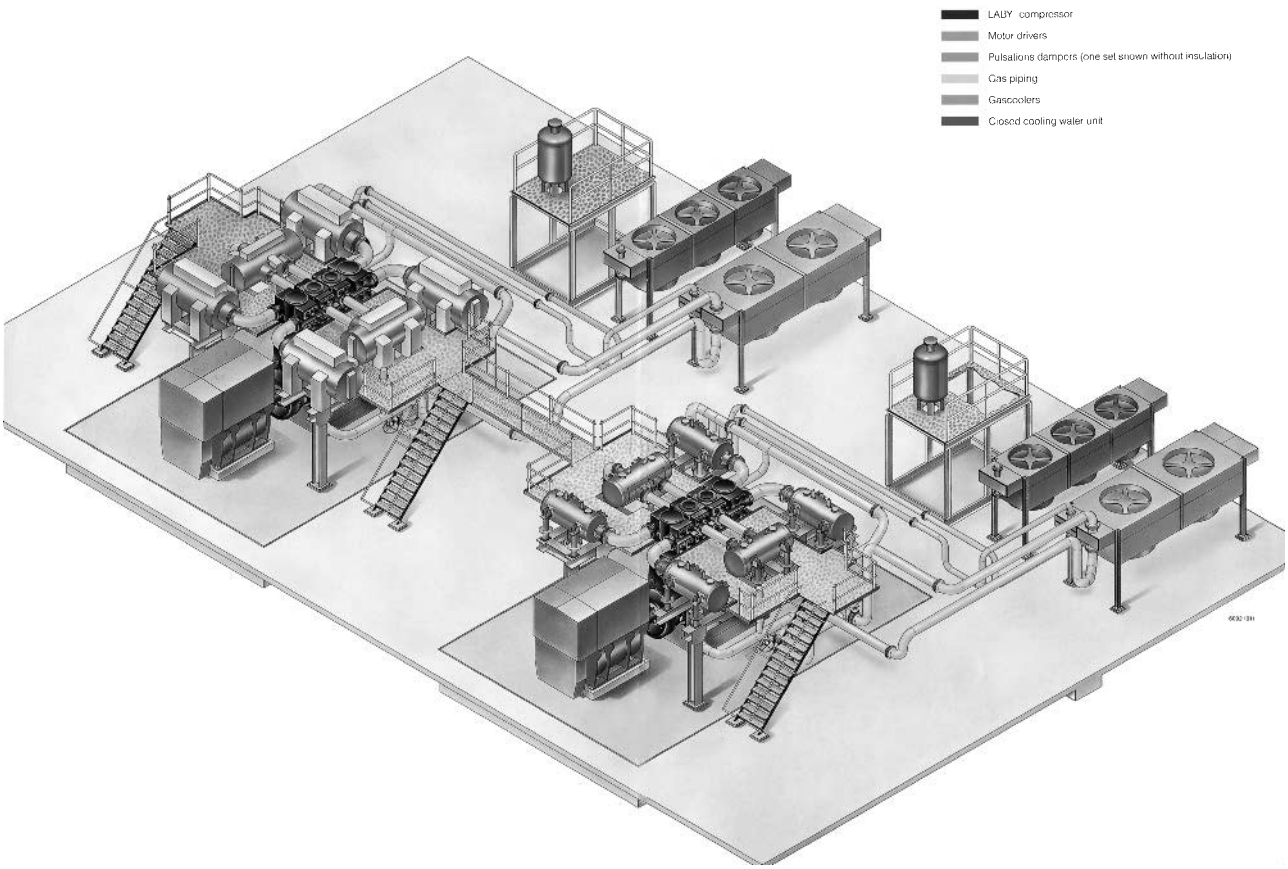


FIG. C-116A Skid of compressor and accessories in LNG service. (Source: Sulzer-Burckhardt.)

TABLE C-11 Maintenance Report on a Period of 75,000 Running Hours

Pistons (total of 4)	No replacement whatsoever	
Piston rod seals	1st replacement after	14,350 h
	2nd replacement after	36,993 h
	3rd replacement after	61,790 h
Piston rods	No replacement whatsoever	
Crankshaft seal	1st replacement after	14,350 h
	2nd replacement after	36,993 h
Bearings		
Piston rods guide bearings	1st replacement after	36,993 h
	2nd replacement after	61,790 h
Crossheads	No replacement	
Crosshead pin bearings	No replacement	
Connecting rod bearings	No replacement	
Crankshaft bearings	One bearing lost after	14,350 h
	No further replacement	

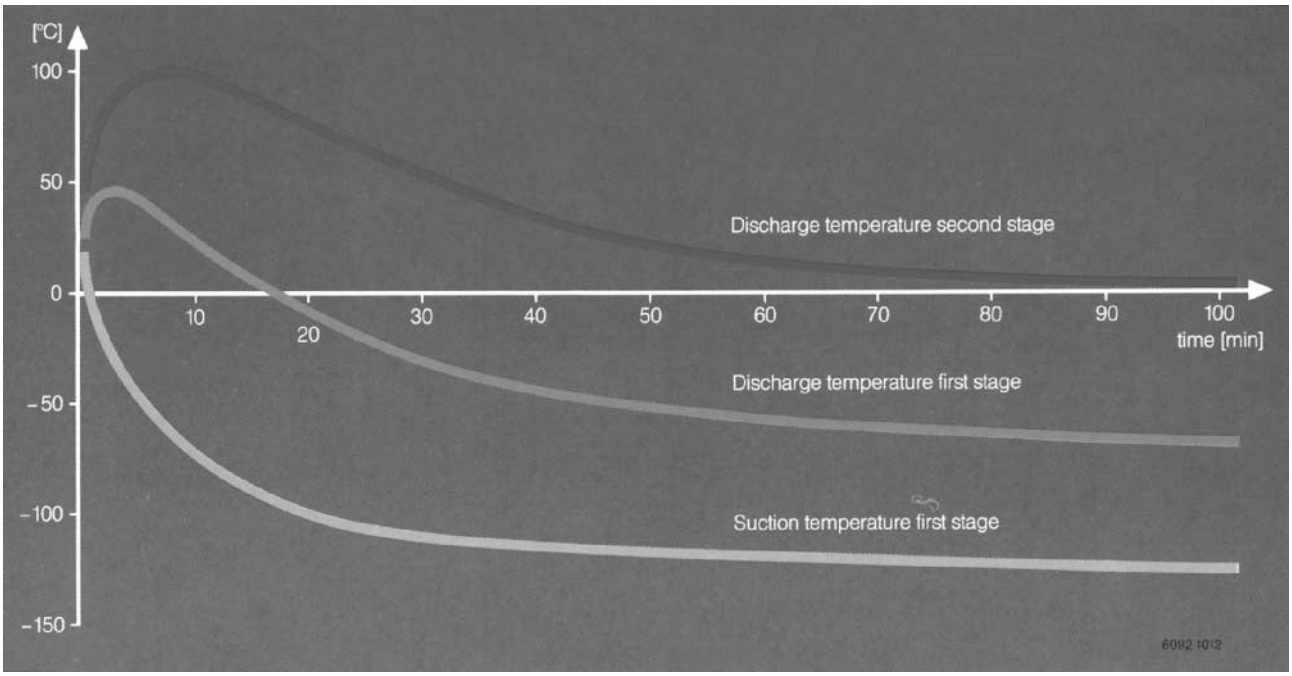


FIG. C-116B The labyrinth-piston compressor is especially well adaptable to the great temperature drop from ambient to –150°C during startup. It is for this reason that no slow cooling-down procedure prior to startup is required. (Source: Sulzer-Burckhardt.)



FIG. C-117A Four labyrinth-piston compressors, type 3K 140-3C, in a storage facility in Sweden handling cold ethylene at –95 °C suction temperature from 1.01 bar abs to 31.5 bar abs in three stages. (Source: Sulzer-Burckhardt.)

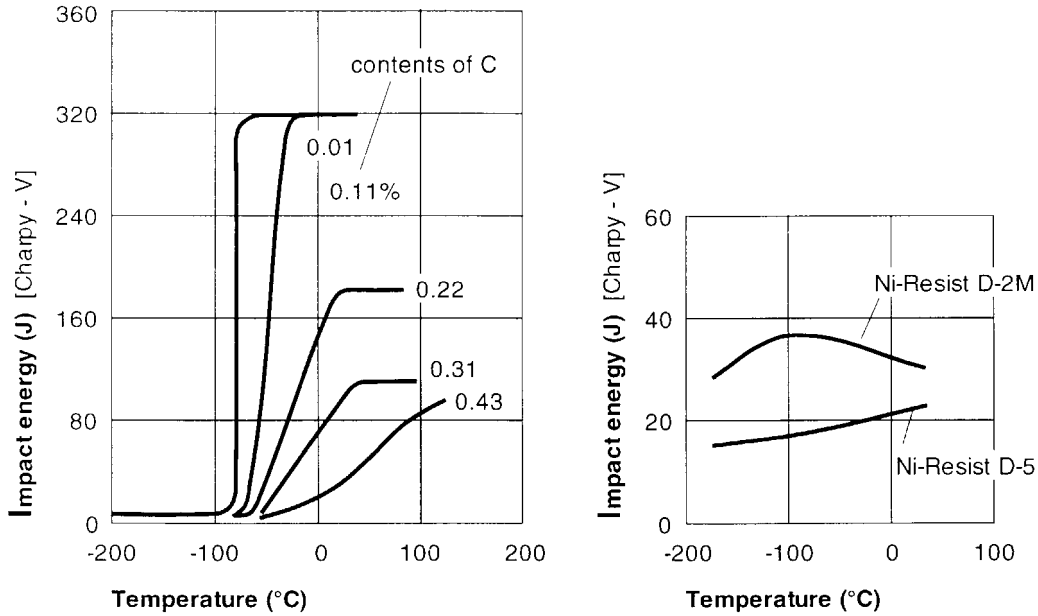


FIG. C-117B Transitions of impact energies of nonalloyed C-steels and Ni-alloyed nodular iron ("Ni Resist"). (Source: Sulzer-Burckhardt.)

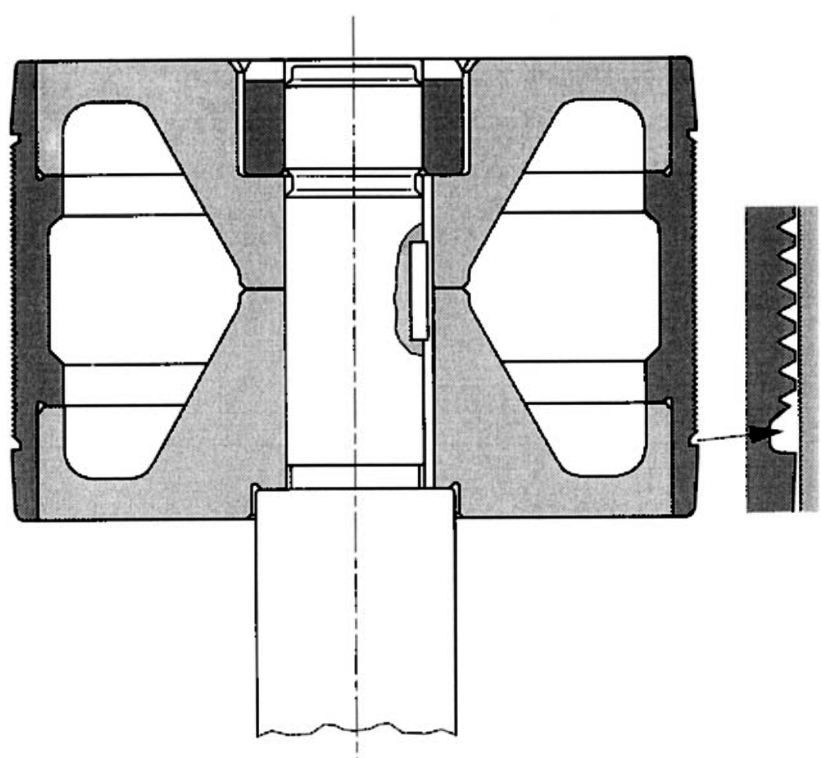


FIG. C-118 Cutaway view of a double-acting labyrinth piston. (Source: Sulzer-Burckhardt.)

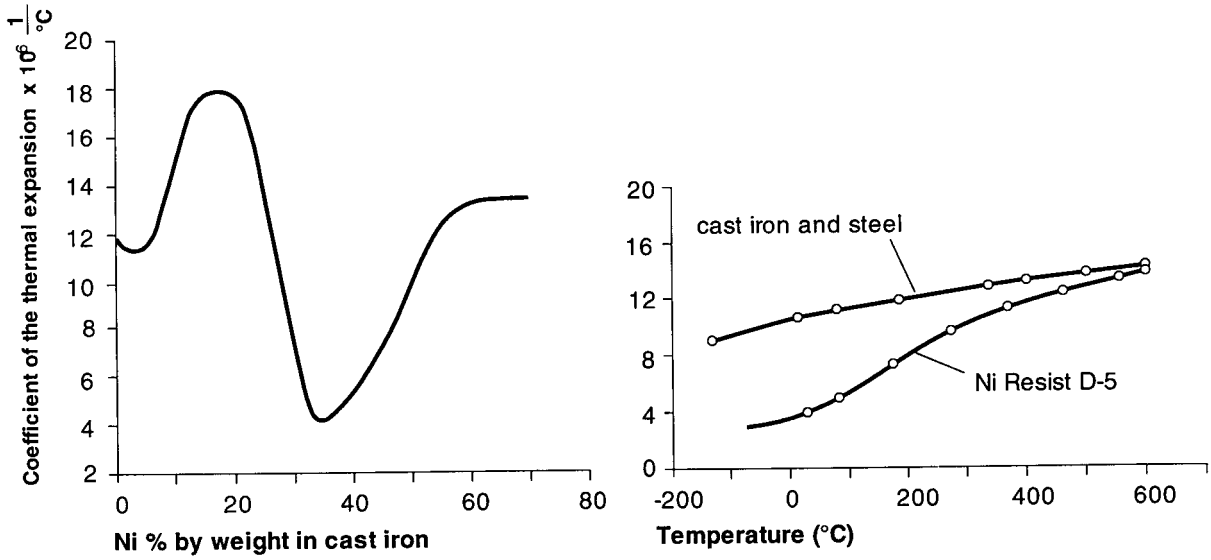


FIG. C-119 Coefficient of thermal expansion for cast iron and steel in function of % Ni and temperature. (Source: Sulzer-Burckhardt.)

TABLE C-12 Candidate Materials for Low Temperature Components—Comparative Combined Properties

	③ Tensile strength N/mm²	① Endurance limit N/mm²	Youngs modulus N/mm²	Thermal expansion coefficient 10^{-6}/°C	② Thermal shock stress N/mm² (t = 100 °C)	②/① Ratio	②/③ Ratio
Cast iron GG 18	180	80	85,000	11.70	100	1.25	0.55
Austenitic Steel CrNi	460 : 600	230 : 300	204,000	20	410	1.80 : 1.40	0.89
GGG NiCr 20 2 Type D2	430	190	125,000	17.6	220	1.10	0.51
GGG Ni 35 Type D5	410	185	127,000	4.50	58	0.32	0.14

temperature conditions. It allows the compressor to be started directly of ambient temperature without any precooling. The less severe temperatures in the 2nd stage allowed the use of ferritic nodular cast iron with good fracture toughness down to –100°C and bronze for the piston. The 3rd stage cylinder consists of normal cast iron grade GG20.

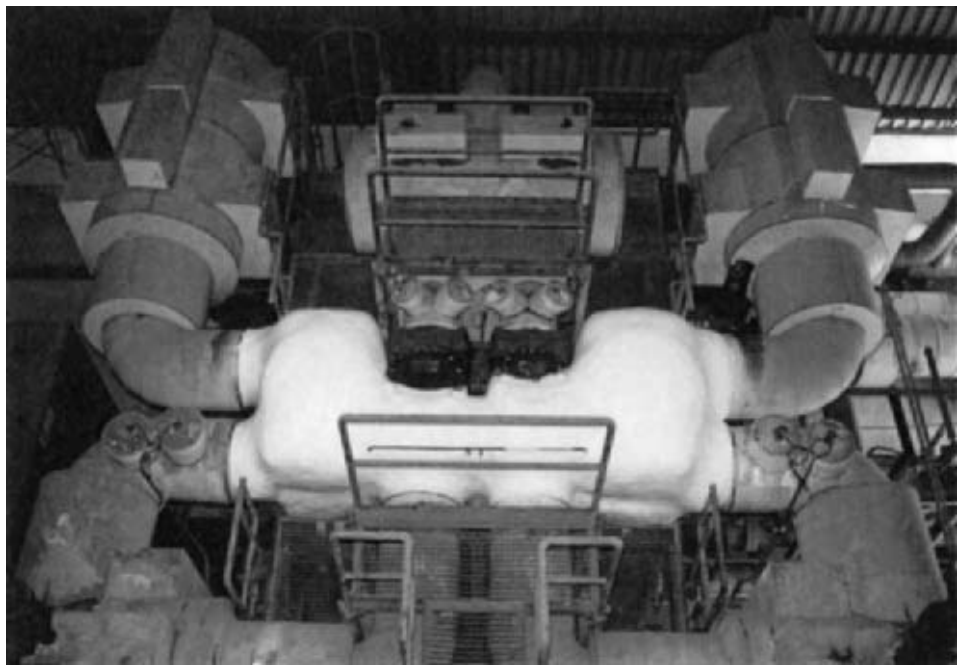


FIG. C-120 Icing of 1st stage cylinders after continuous operation of approximately 2–3 months. (Source: Sulzer-Burckhardt.)

Control of temperature and deformation of the crankcase. Gas temperatures at the 1st stage inlet valves are so low that energy imparted to the cylinders during gas compression raises their mean temperature to a value still well below that of ambient air. Therefore they do not have cooling jackets. They cool down well below the freezing point of the moisture in the natural atmosphere and consequently become covered with a thick layer of ice when the machine is running (Fig. C-120). To ensure a good alignment of the path of the labyrinth pistons, cold deformation of the crankcase underneath the 1st stage cylinders had to be prevented. This was achieved by means of a special water jacket that extends along the upper face of the crankcase and acts as a thermal barrier. See Fig. C-121 for component details.

Internal and external leakage. Consistent with the design of the pistons, labyrinth seals were also used between the double-acting cylinders and the distance piece at the upper end of the crankcase (Fig. C-122). Figure C-122 also shows details of the labyrinth seals around the piston rods. Each gland has a collector chamber before the lower end of the labyrinths from where the leak gas is internally returned to the suction upstream of the 1st stage cylinders.

To attain a perfect external tightness of the machine the passage of the crankshaft through the wall of the crankcase was sealed off by a rotating double-sided ring seal immersed in oil. Thus, the entire inside of the frame could be integrated into the gas-containing system and could be pressurized with either natural gas or an inert gas. In the case presented here it was left at suction pressure level and filled with natural gas. Figure C-123 shows details of the shaft seal. The entire machine represents, therefore, one hermetically closed shell with no gas leakage to the environment.

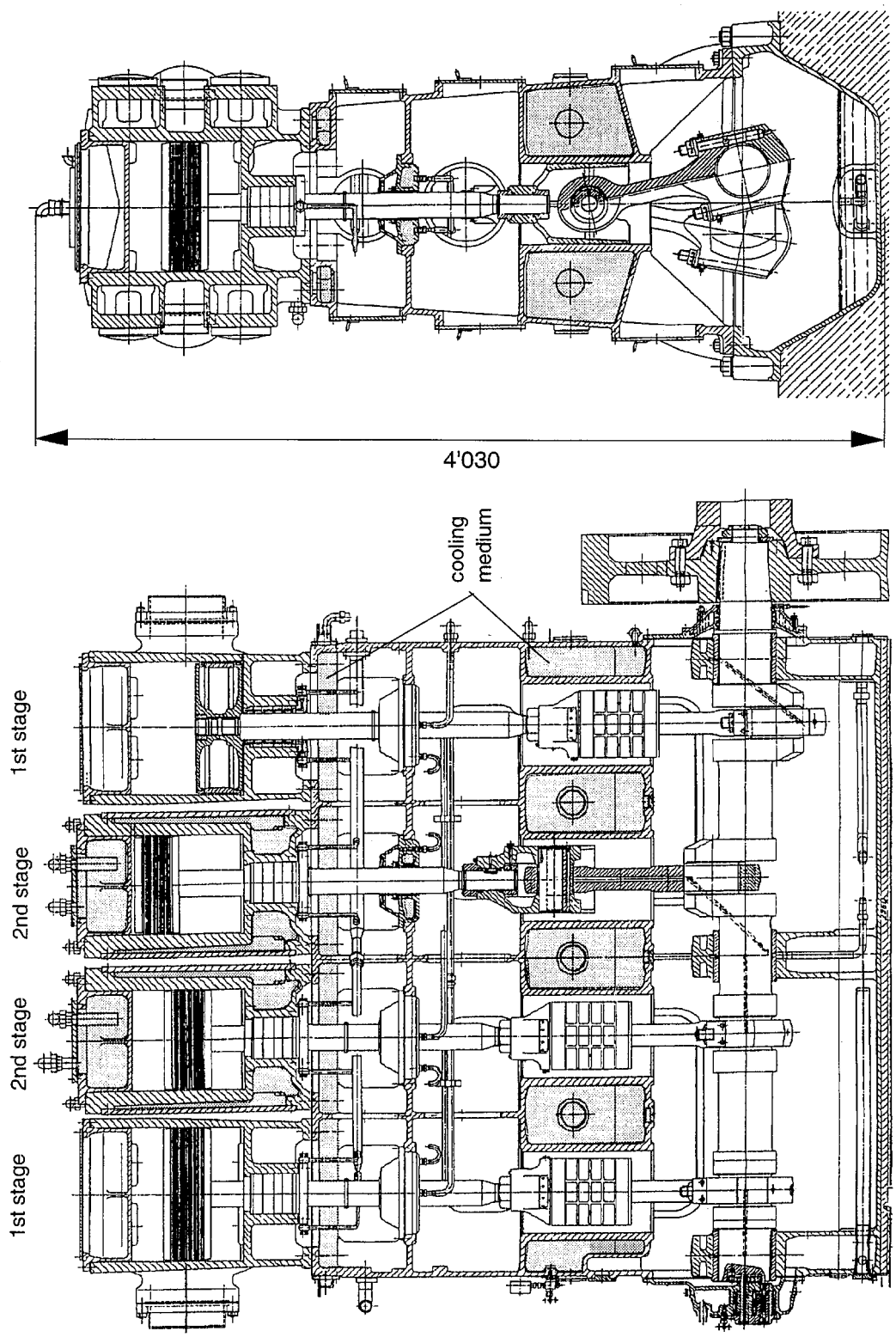


FIG. C-121 Labyrinth-piston compressor for LNG boil-off: 4 double-acting cylinders; 3 compression stages; closed crankcasing; suction temperature −160°C. (Source: Sulzer-Burckhardt.)

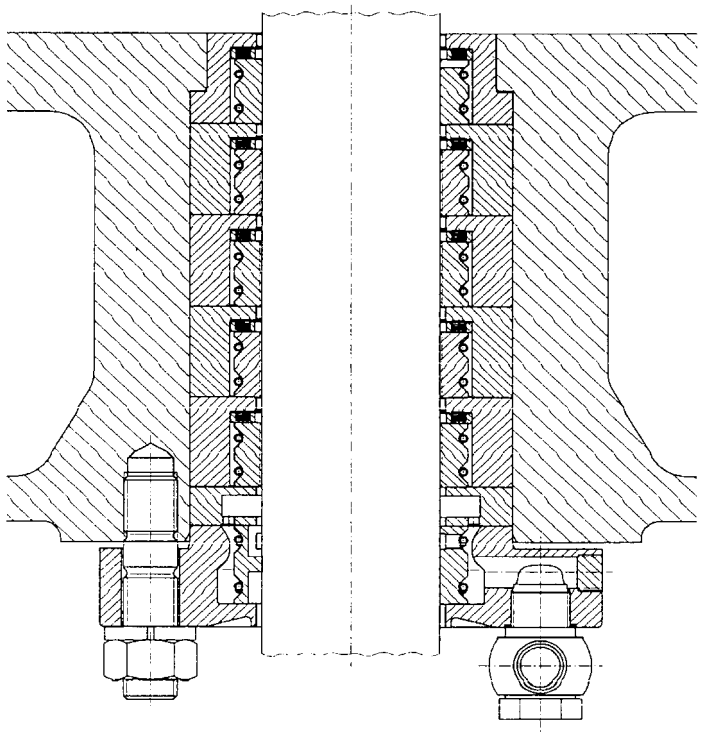


FIG. C-122 Internal labyrinth sealing between double-acting cylinder and distance piece of crankcasing (piston rod sealing). (Source: Sulzer-Burckhardt.)

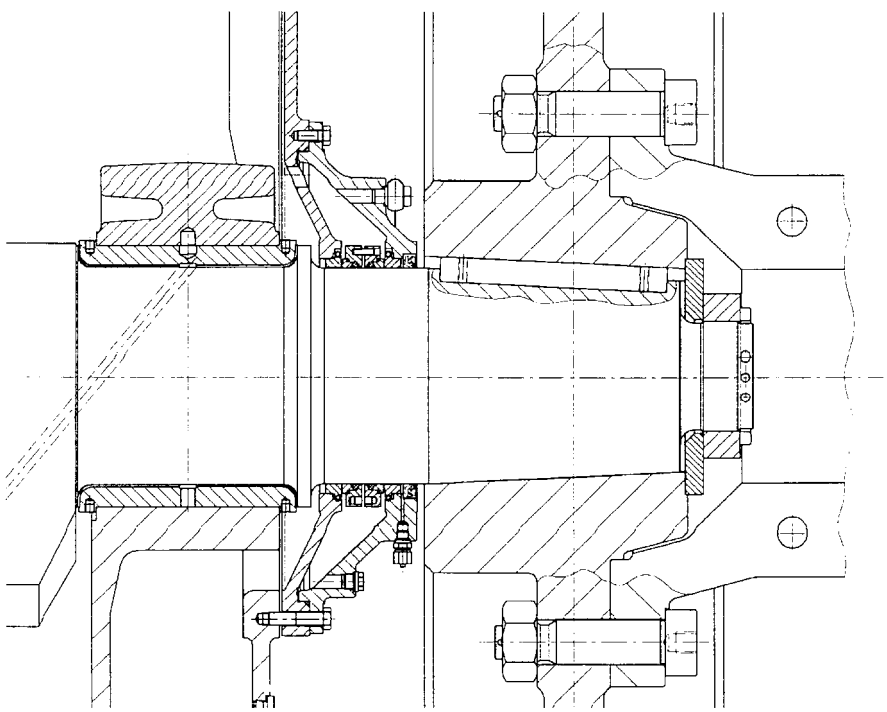


FIG. C-123 Gas-tight sealing of crankshaft between crankcasing and environments. (Source: Sulzer-Burckhardt.)

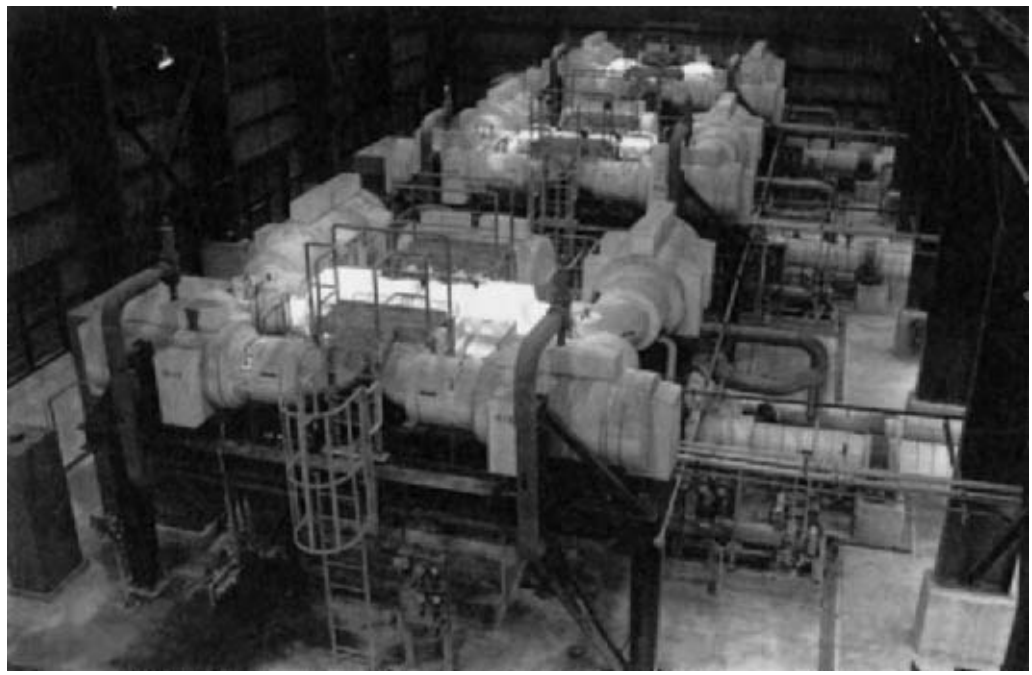


FIG. C-124 Group of three labyrinth-piston compressors for LNG boil-off in a Taiwanese LNG receiving terminal. (Source: Sulzer-Burckhardt.)

Valves. With valve discs, orders for replacment parts indicate an average life of at least 10,000 h with no changes based on running temperatures.

Other LNG terminals. The applications experience of the labyrinth piston compressor in this market segment has encouraged other terminals to install machines of the same kind. Figure C-124 shows a group of machines in a Taiwanese terminal where LNG is received, stored, and evaporated for distribution by a pipeline system throughout the island of Taiwan. When the terminal was extended in 1992, in view of the successful operation of these units an order for an additional compressor was placed. In this case a two-stage machine was required. As the discharge temperature after the 2nd stage reaches about −50 to 0°C only, both stages have uncooled cylinders.

The process data for this compressor are:

Gas		CH_4 (98%) + N_2 (2%)				
Suction	1st	1.036	bar	−106	to	−160°C
Discharge	1st	2.9	bar	−36	to	−97°C
Suction	2nd	2.9	bar	−33	to	−93°C
Discharge	2nd	6.8	bar	−45	to	+35°C

Maintenance report on these compressor applications. By the end of January 1996 the first three compressors had all passed 15,000 running hours. One compressor was opened up after 10,000 running hours. Piston and bearing clearances, piston rod seals, and valves were checked. Some gland rings and some minor valve parts were replaced. The maintenance report read: "like new!"

Startup procedure. Special attention was given to developing a simple startup procedure. The low thermal expansion coefficient of the chosen cylinder material

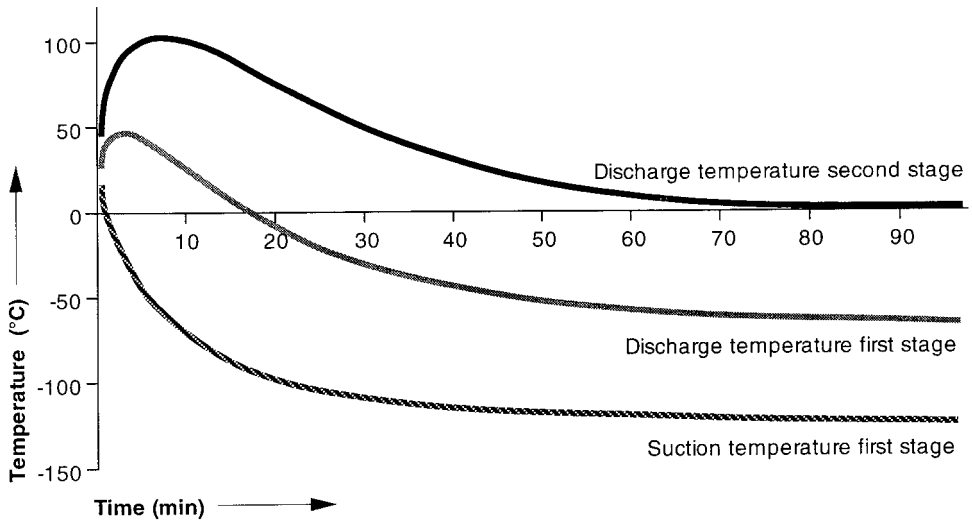


FIG. C-125 Gas temperature readings taken during the startup of a LNG boil-off gas labyrinth-piston compressor. (Source: Sulzer-Burckhardt.)

leads to low thermal stress. Together with carefully designed pulsation dampeners and a well-engineered gas piping arrangement, the system allows full automatic startup of the compressors. The transition from ambient temperature down to boil-off temperature is achieved without any precooling (Fig. C-125). Figure C-126 shows two other LNG boil-off gas compressors in the design stage commissioned in a Korean LNG receiving terminal.

Summary of operational experience thus far. Low gas temperature challenges gas compressors in two ways:

- Physical contact with cold gas and consequences for material properties
- Absence of humidity (low dew point) with a strong bearing on tribology in nonlubricated areas

The application of labyrinth seals in reciprocating compressors is a logical answer to these problems. Labyrinth piston compressors have demonstrated this in industrial operation successfully down to boil-off temperature of natural gas at –160°C. Such machines can be built with zero leakage to the environment. They need little maintenance. Plant operators and maintenance crews claim low attention requirements.

Reference and Additional Reading

1. Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Turbocompressors (Includes Centrifugal, Axial, and Isothermal Compressors, as Well as Fans)

General Information*

A diagrammatic summary of operating ranges of various categories under this subdivision is presented here, followed by further details and information on some of the more important compressor categories. The model designations are specific to this OEM information source. When selecting a compressor, one way of not

*Source: Sulzer-Burckhardt, Switzerland. Adapted with permission.

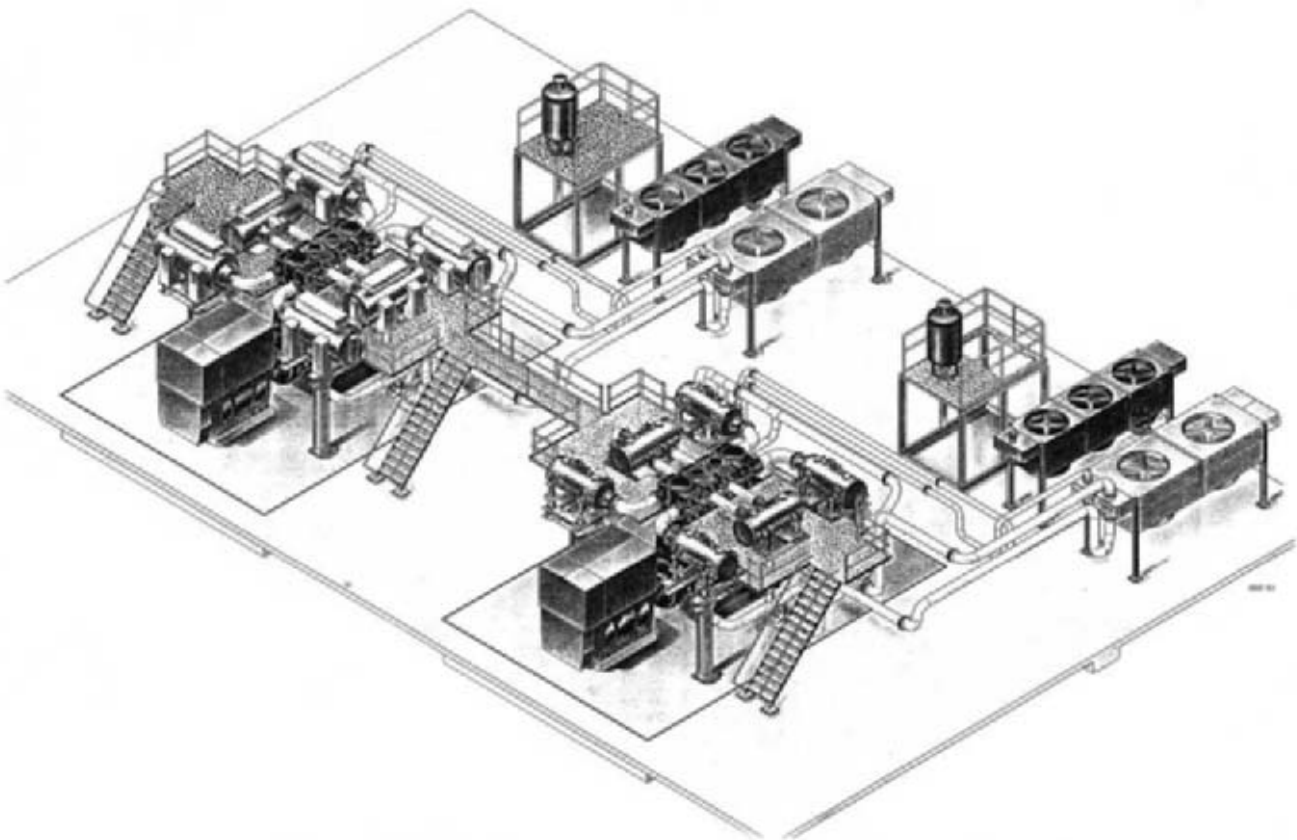


FIG. C-126 Two identical labyrinth-piston compressor plants, put into operation in a Korean LNG receiving terminal in December 1993. (Source: Sulzer-Burckhardt.)

omitting any vital features is to consider the specifications of one well-known OEM and compare other OEMs for equivalent features.

The pressure-volume diagram (Fig. C-127) shows the duty range of the standard model series for this OEM source. Each compressor's size is arrived at with different numbers of stages and nominal pressures, depending on the application. Pictorial operational summaries of these main categories are shown in Figs. C-129 through C-141.

Turbocompressor controls (*see* Control Systems)

Axial compressors*

Axial compressors are being used increasingly for applications that not long ago were considered the domain of centrifugal machines. Thanks to their high specific flow capacity, the corresponding low weight, reduced space requirement, and particularly their high efficiency, the axial compressors play a major role in the reliable and economic operation of modern, large-scale industrial plants. They now form a vital and indispensable integral part of installations like blast furnaces, air separation plants, fluid catalytic cracking units, nitric acid plants, jet-engine test facilities, thermocompression, LNG, and synfuel processes. See Fig. C-128.

*Source: Sulzer-Burckhardt, Switzerland. Adapted with permission.

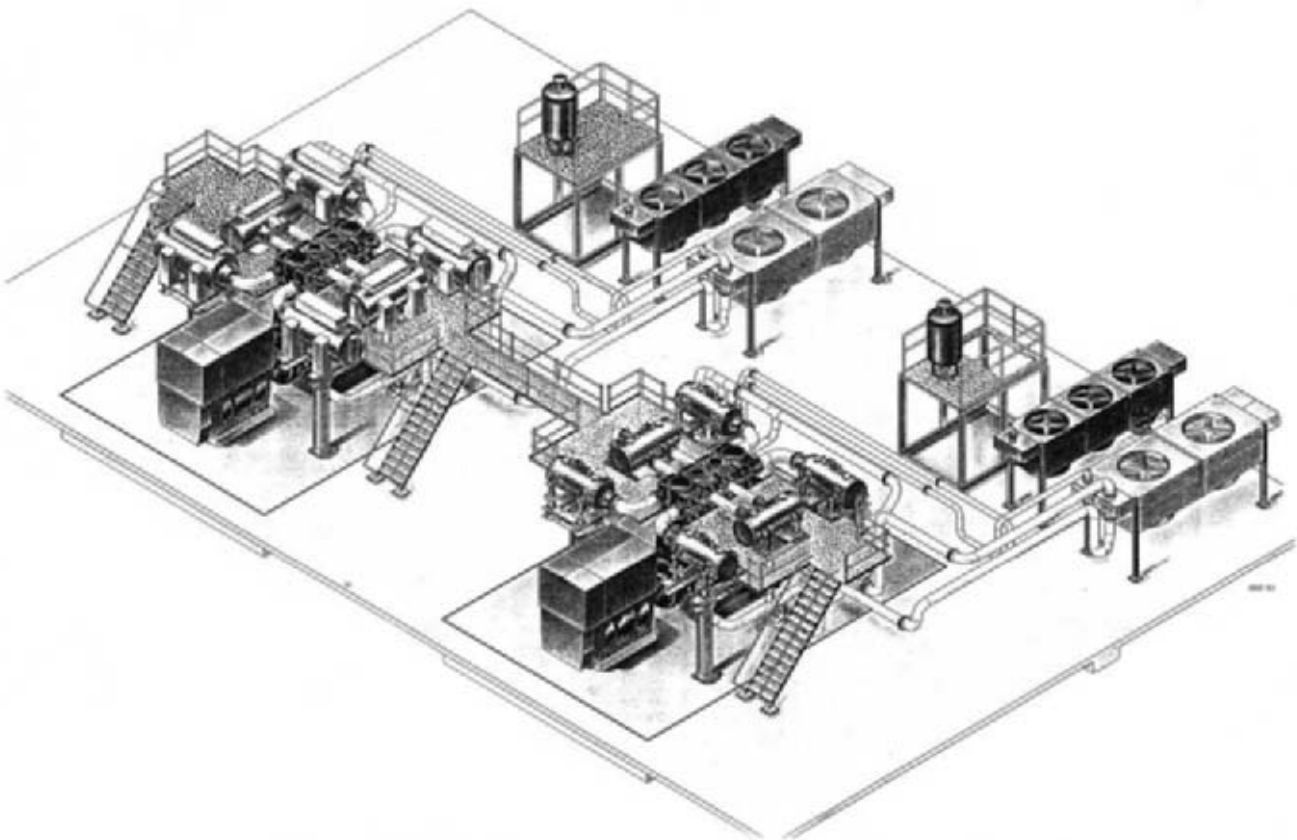


FIG. C-126 Two identical labyrinth-piston compressor plants, put into operation in a Korean LNG receiving terminal in December 1993. (Source: Sulzer-Burckhardt.)

omitting any vital features is to consider the specifications of one well-known OEM and compare other OEMs for equivalent features.

The pressure-volume diagram (Fig. C-127) shows the duty range of the standard model series for this OEM source. Each compressor's size is arrived at with different numbers of stages and nominal pressures, depending on the application. Pictorial operational summaries of these main categories are shown in Figs. C-129 through C-141.

Turbocompressor controls (*see* Control Systems)

Axial compressors*

Axial compressors are being used increasingly for applications that not long ago were considered the domain of centrifugal machines. Thanks to their high specific flow capacity, the corresponding low weight, reduced space requirement, and particularly their high efficiency, the axial compressors play a major role in the reliable and economic operation of modern, large-scale industrial plants. They now form a vital and indispensable integral part of installations like blast furnaces, air separation plants, fluid catalytic cracking units, nitric acid plants, jet-engine test facilities, thermocompression, LNG, and synfuel processes. See Fig. C-128.

*Source: Sulzer-Burckhardt, Switzerland. Adapted with permission.

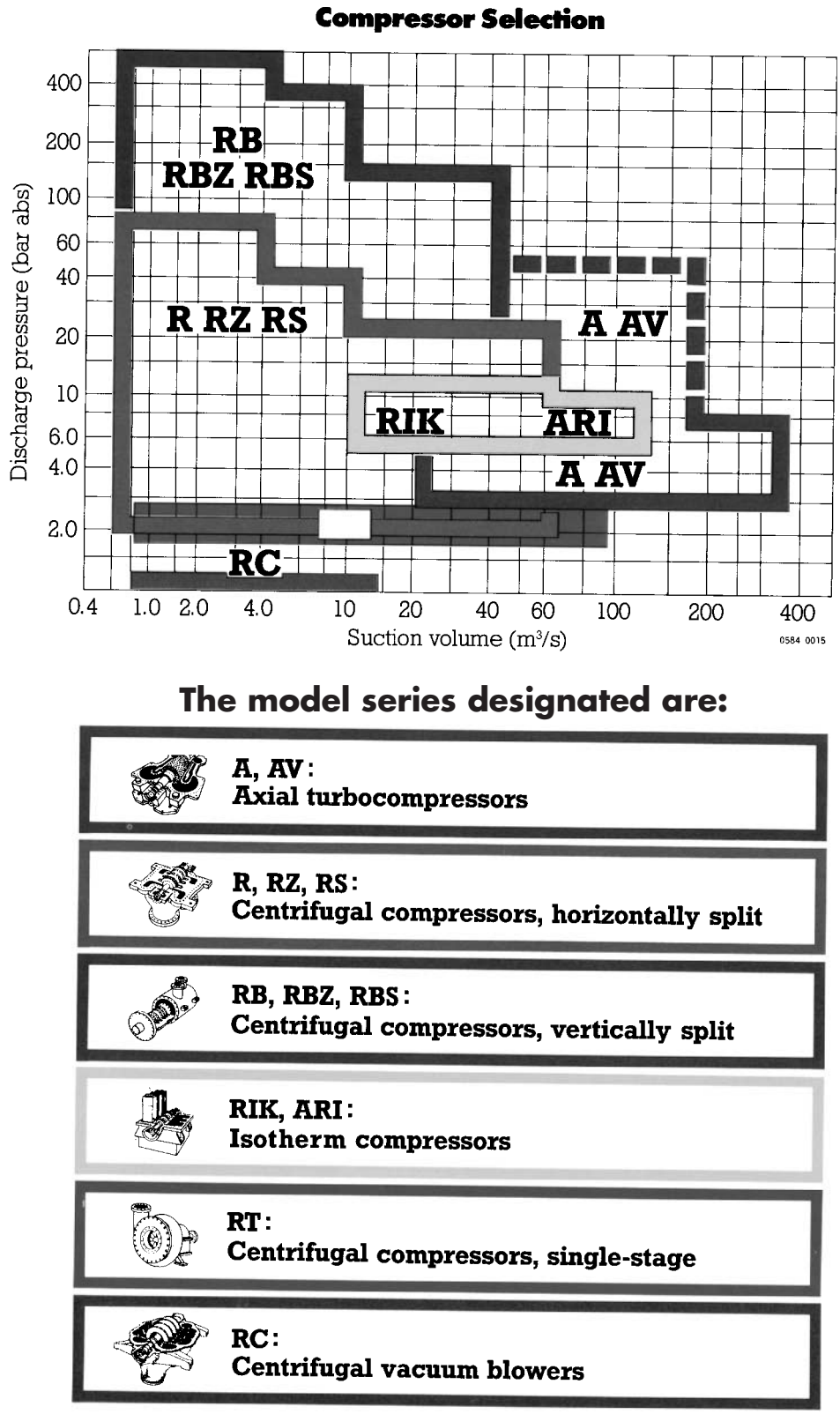


FIG. C-127 Turbcompressor application envelopes. (Source: Sulzer-Burckhardt.)

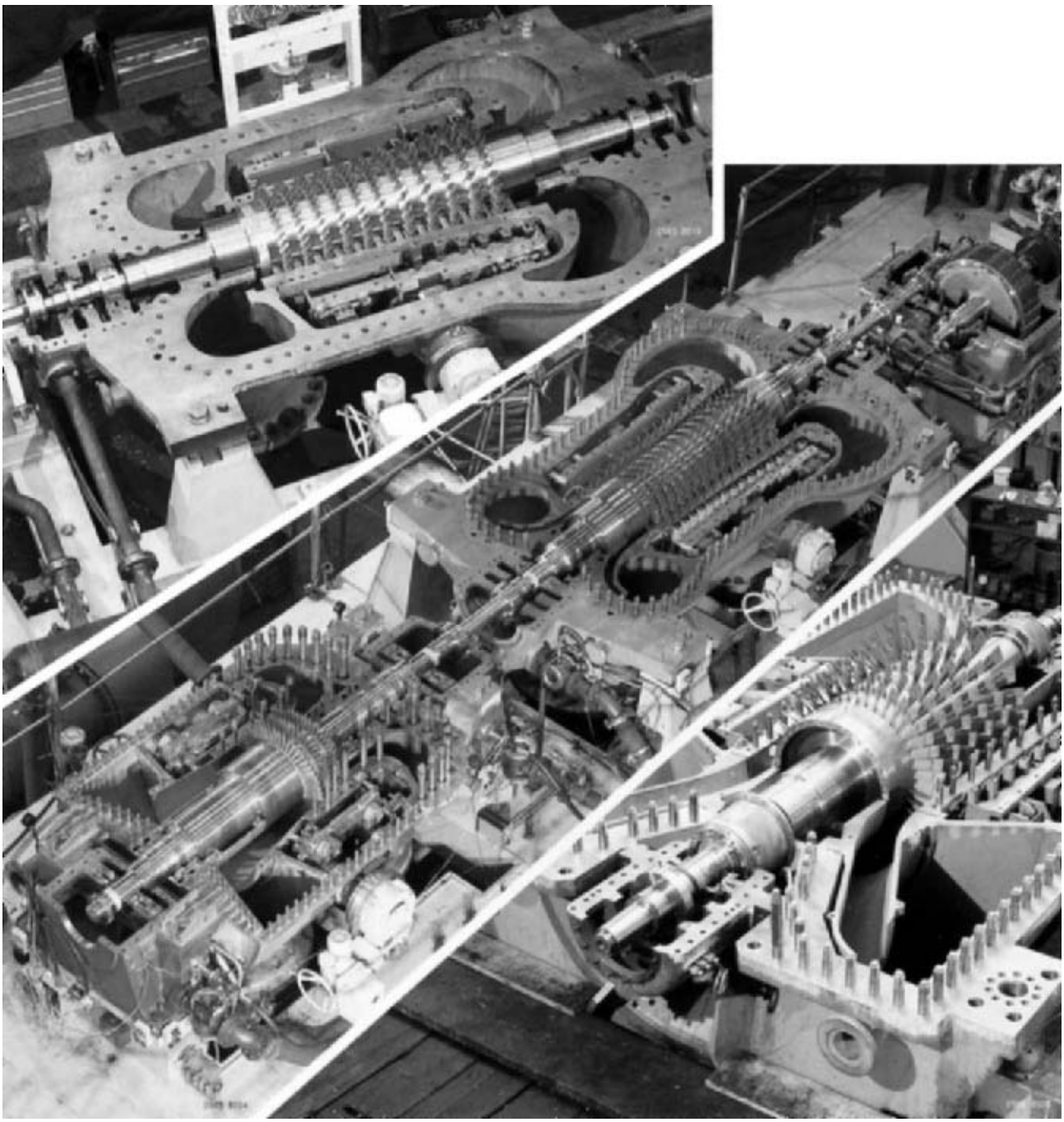
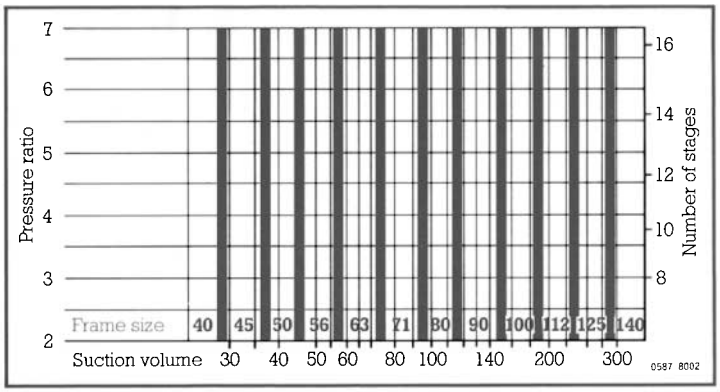
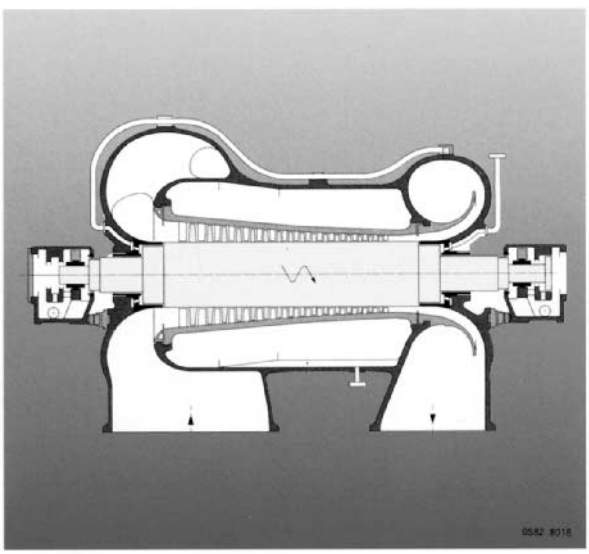
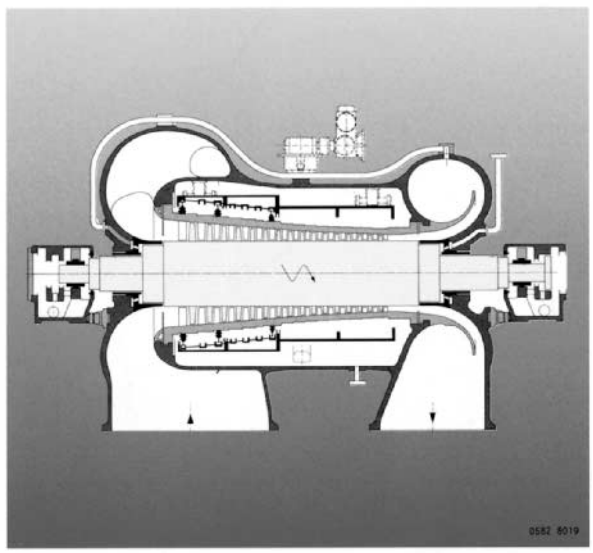


FIG. C-128 Sections through typical axial compressors. Applications include blast furnaces, refineries, LNG plants, nitric acid plants, chemical and petrochemical plants, compressed air storage, and aeroengine research. (Source: Sulzer-Burckhardt.)

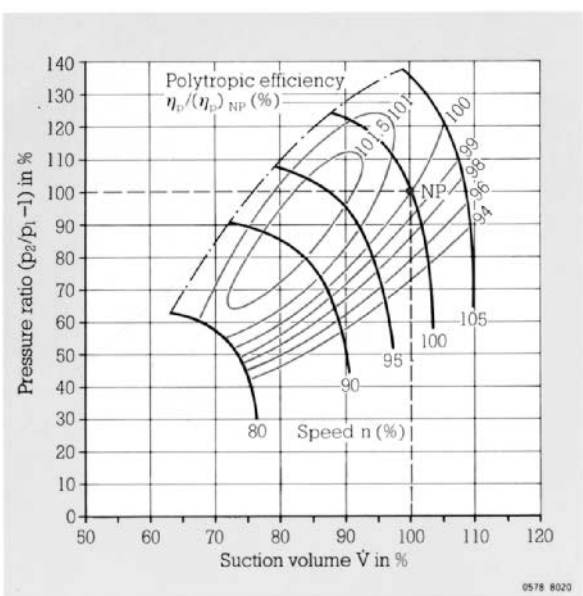




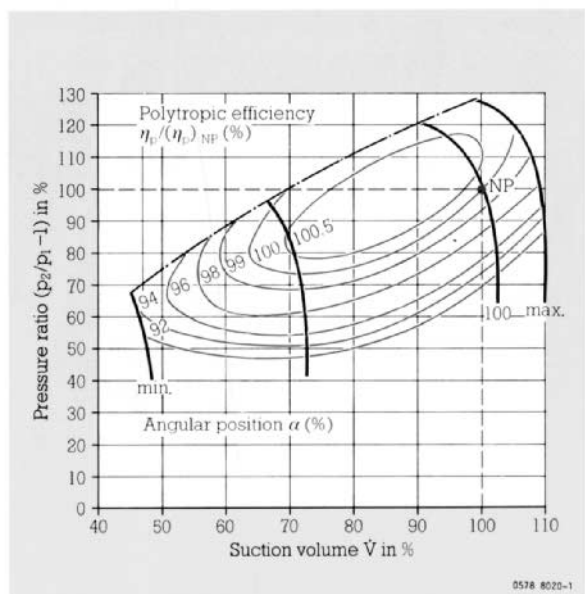
Series A
with fixed stator blades (FIXAX).



Series AV
with adjustable stator blades (VARAX).



Performance map for axial compressors
with speed variation.



Performance map for axial compressors
with adjustable stator blades.

FIG. C-129 Two generic model designations (FIXAX and VARAX) provide 12 frame sizes, with rotor diameters ranging from 40 to 140 cm, covering suction volume ranging from 70,000 to 1,250,000 m^3/h, materials to suit specific applications. (Source: Sulzer-Burckhardt.)

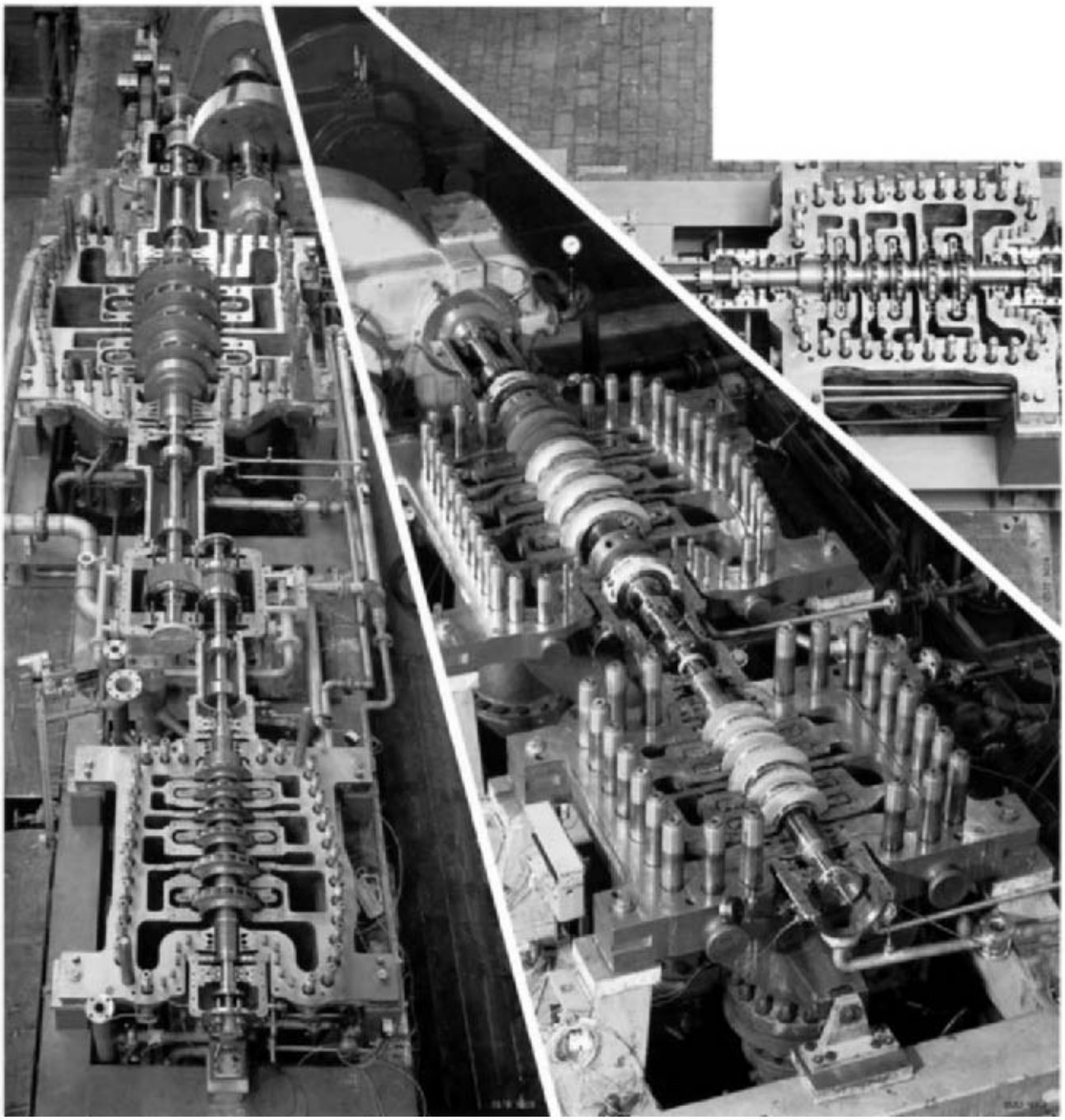
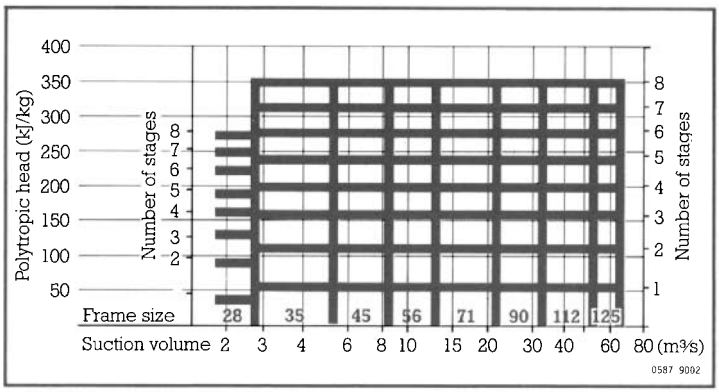
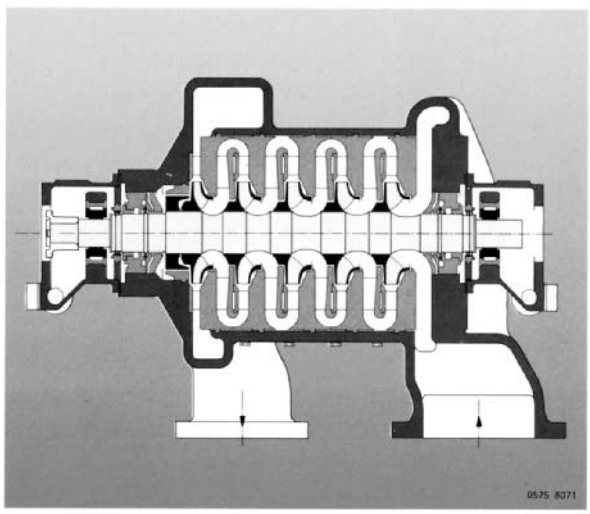
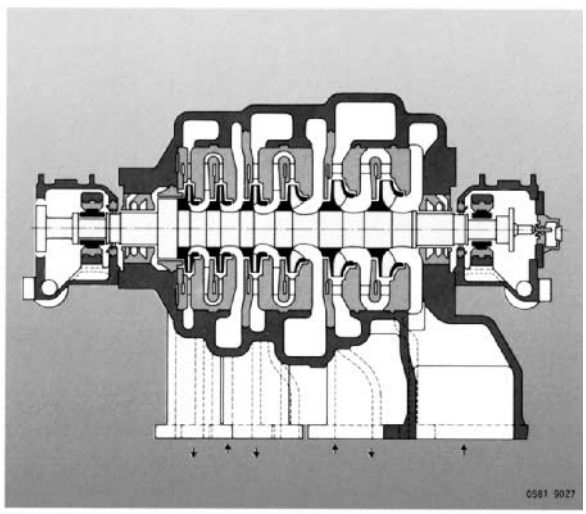


FIG. C-130 Sections through centrifugal turbocompressors, horizontally split. Applications include oxygen compression, the chemical industry, the petrochemical industry, high-pressure air compressors, and refrigeration. (Source: Sulzer-Burckhardt.)

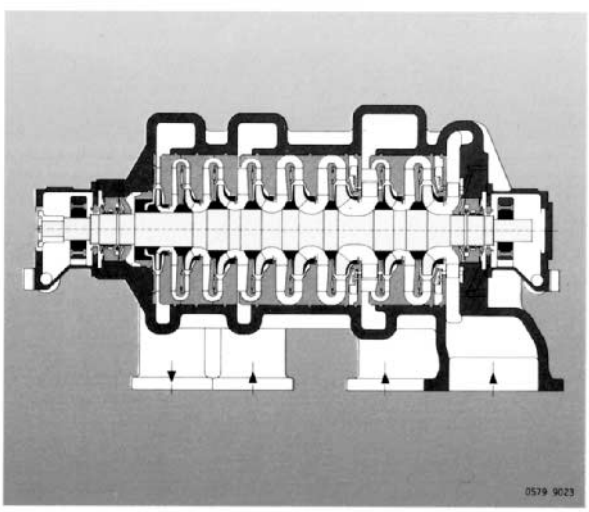




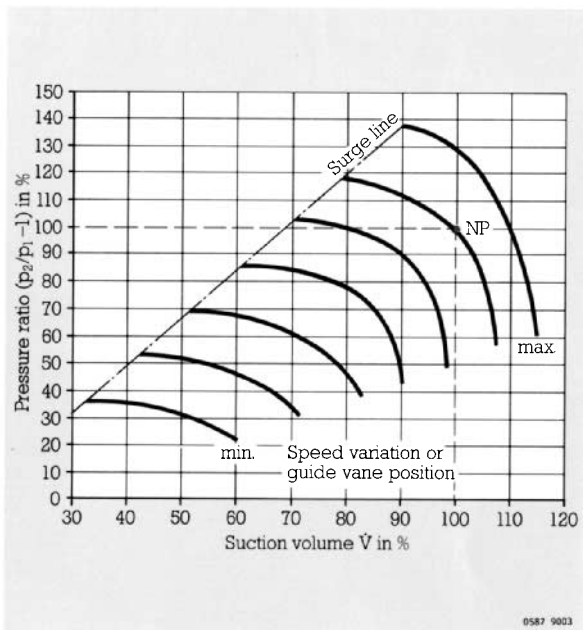
Series R
Machines of one to eight stages with horizontally split casing.



Series RZ
Machines of four to eight stages with horizontally split casing, one to two pairs of intermediate nozzles generally for connecting external intercoolers.



Series RS
Machines of four to eight stages with horizontally split casing and one or two nozzles for side stream intake or extraction.



Typical performance map for horizontally split centrifugal compressors.

FIG. C-131 Centrifugal compressor basic designations and performance envelopes that require eight frame sizes, for discharge pressures up to 70 bar, suction volumes from 4000 to 220,000 m³/h. (Source: Sulzer-Burckhardt.)

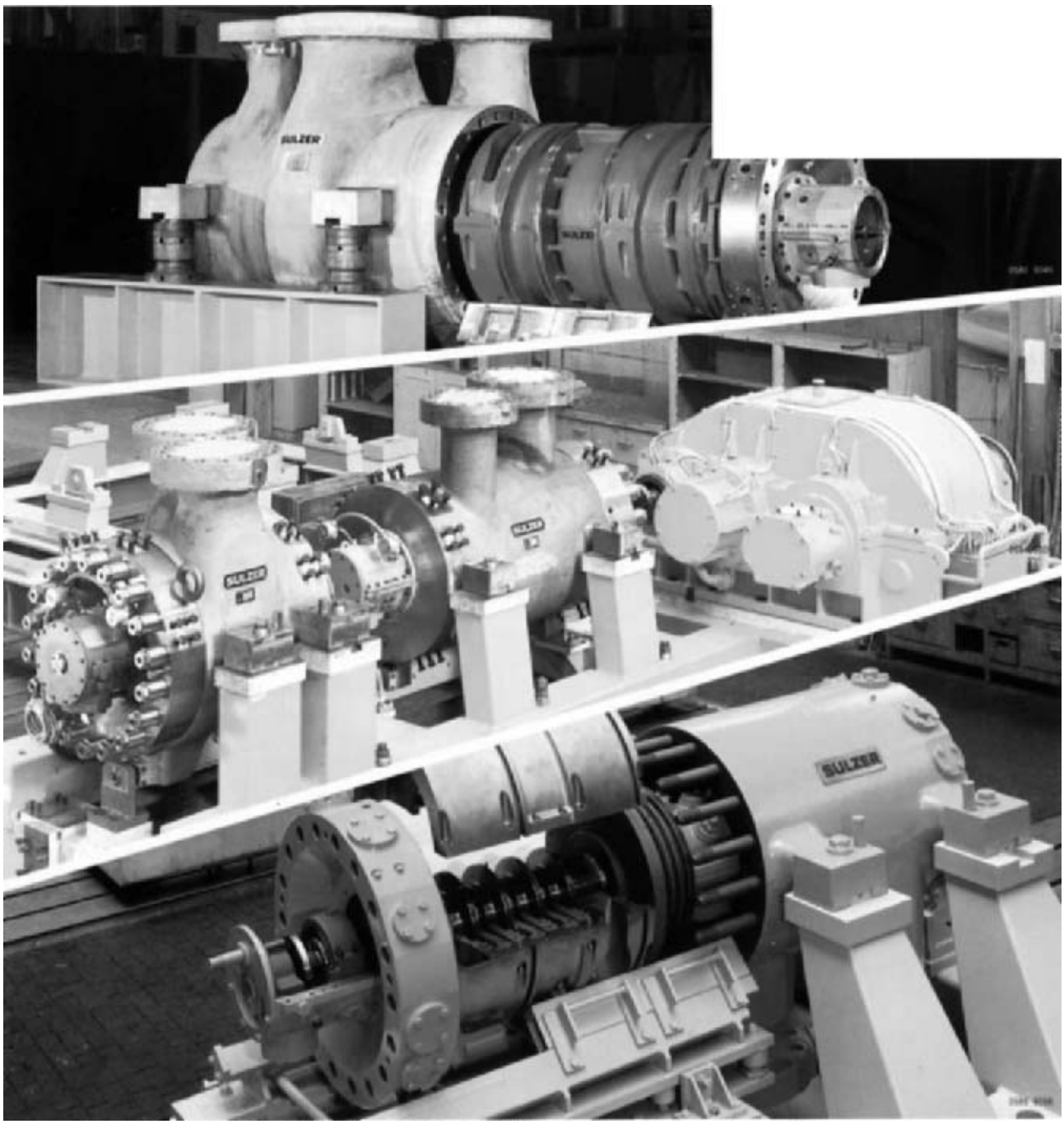
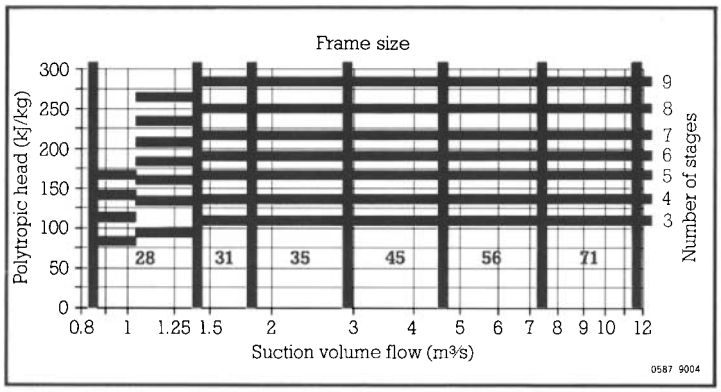
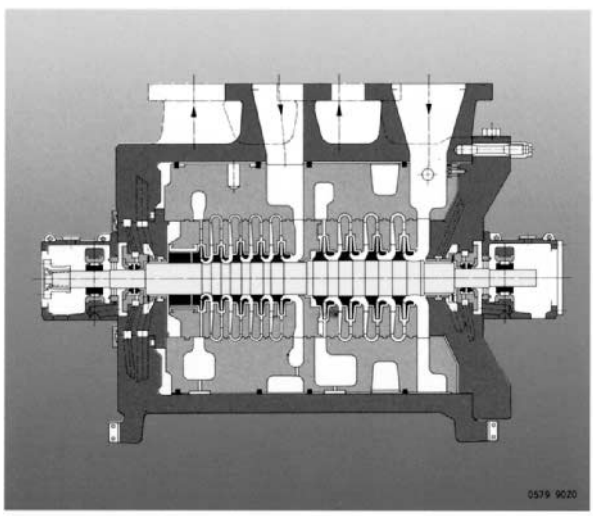
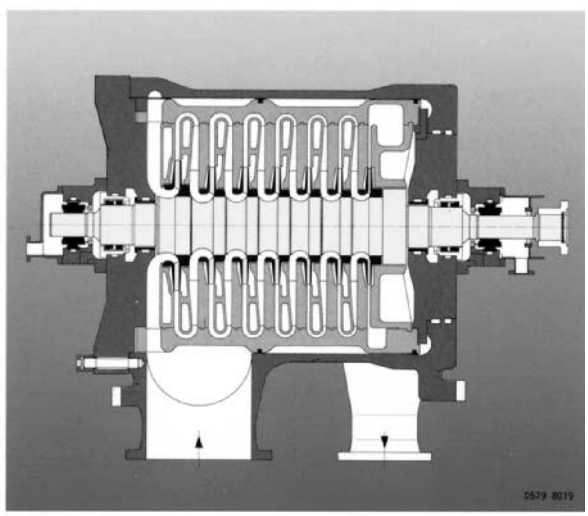


FIG. C-132 Vertically split centrifugal compressors. Applications include high-pressure oxygen; syngas compression; hydrocarbon compression; refinery gas compression; gas transmission; and reinjection, gas lift. (Source: Sulzer-Burckhardt.)

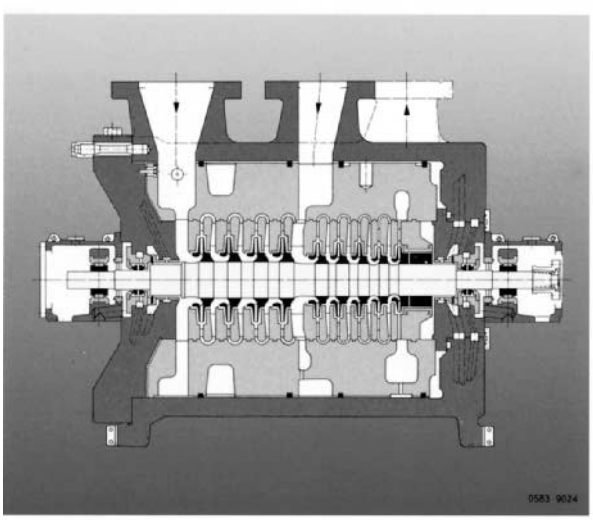




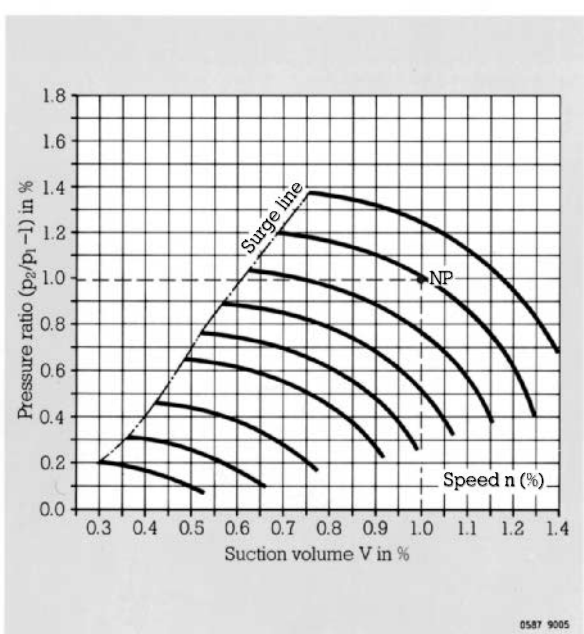
Series RBZ
Machines of four to nine stages with vertically split casing and one pair of intermediate nozzles.



Series RB
Machines of two to eight stages with vertically split casing (barrel type).



Series RBS
Machines of four to nine stages with vertically split casing and additional inlet or outlet nozzles.



Typical performance map for vertically split centrifugal compressors.

FIG. C-133 Vertically split axial turbocompressors for this OEM incorporate six frame sizes, for discharge pressures up to 700 bar, suction volumes from 3000 to 45,000 m³/h. (Source: Sulzer-Burckhardt.)

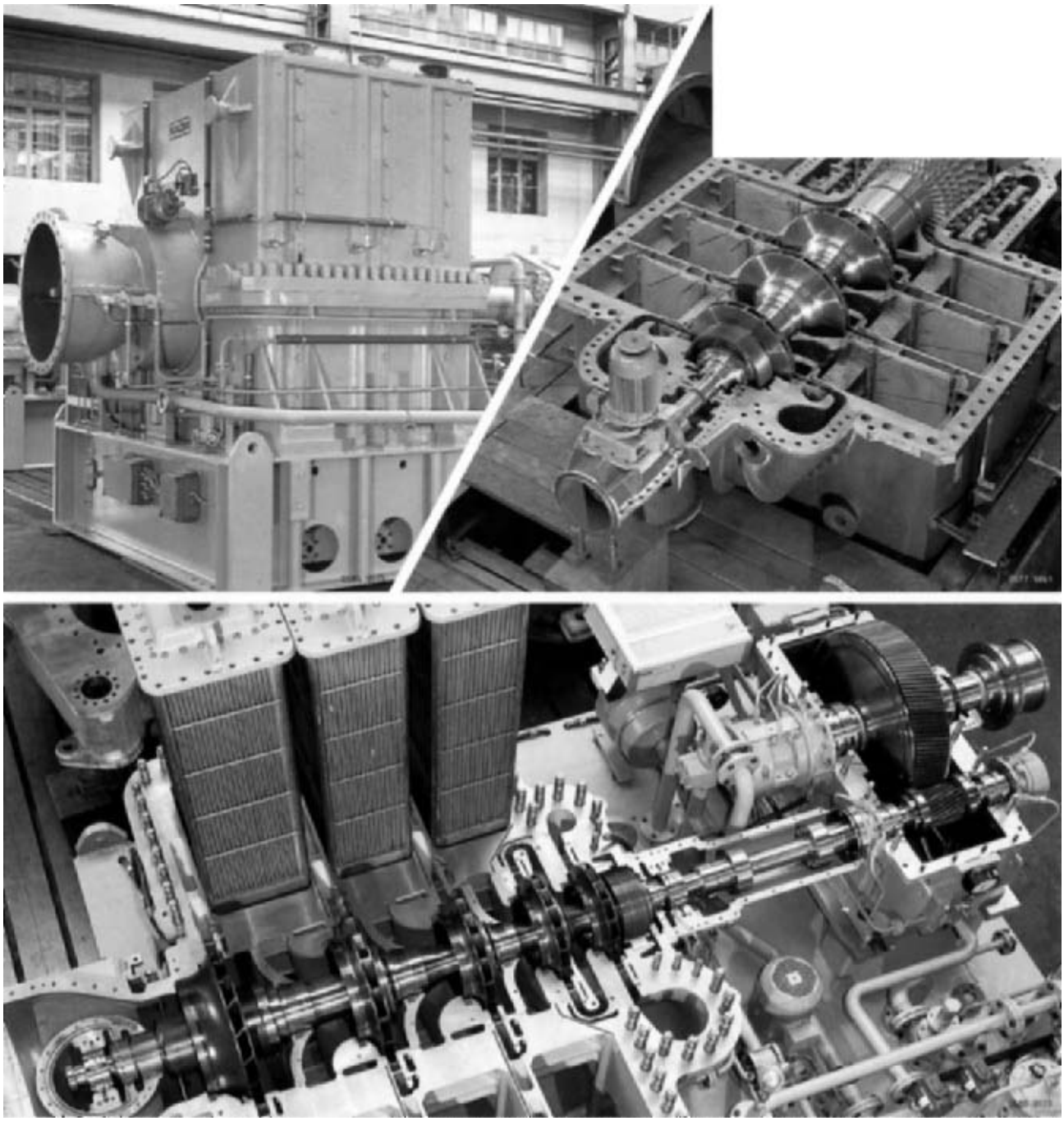
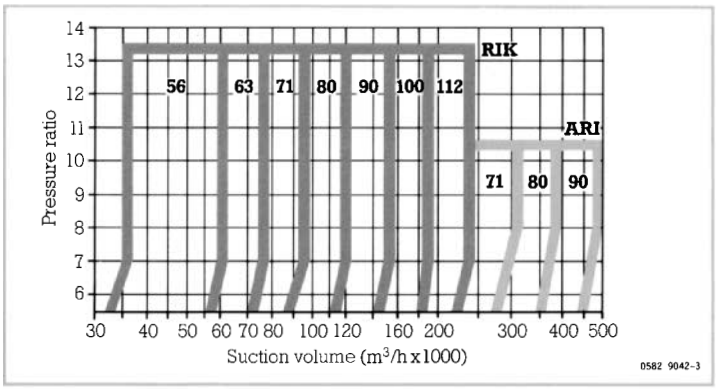
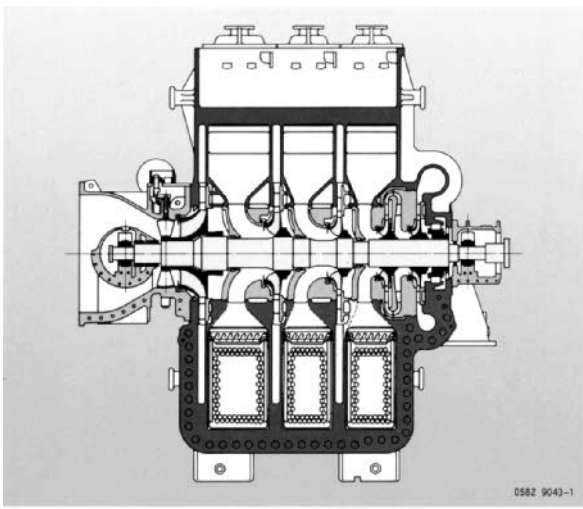


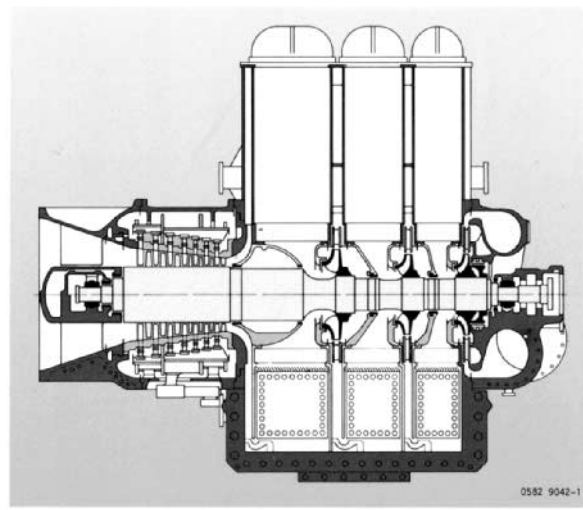
FIG. C-134 Isothermal compressors offer low power consumption and the lowest space requirement of any compressor. They are generally available with axial inlet section for very large flows and with a washing device for severe intake conditions. Applications include air separation, mining, process air in the chemical industry, and instrument air. (Source: Sulzer-Burckhardt.)

Click here for next page

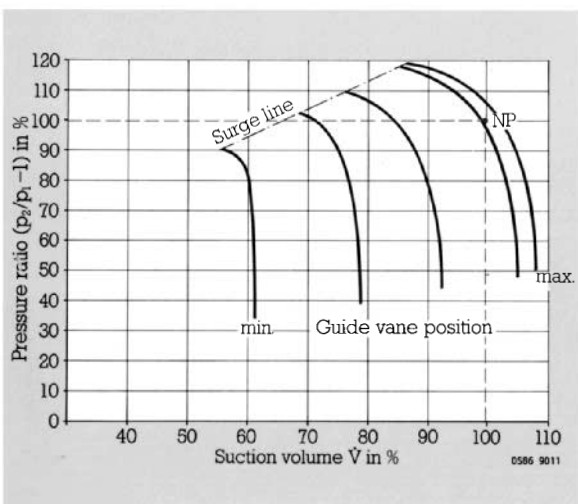




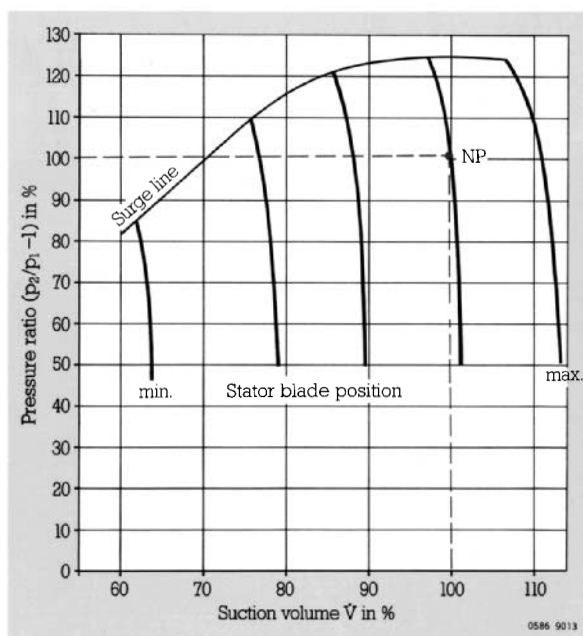
Series RIK
Compact five-stage centrifugal design
Series RIO (adapted for compressing oxygen)



Series ARI
Compact design with six axial and three centrifugal stages.



RIK compressor with inlet guide vanes and constant-speed driver.



ARI compressor with adjustable stator blades and constant-speed driver.

FIG. C-135 Isothermal compressors with this OEM incorporate 10 frame sizes, providing an applications envelope over flow ranges from 35,000 to 500,000 m³/h. (Source: Sulzer-Burckhardt.)

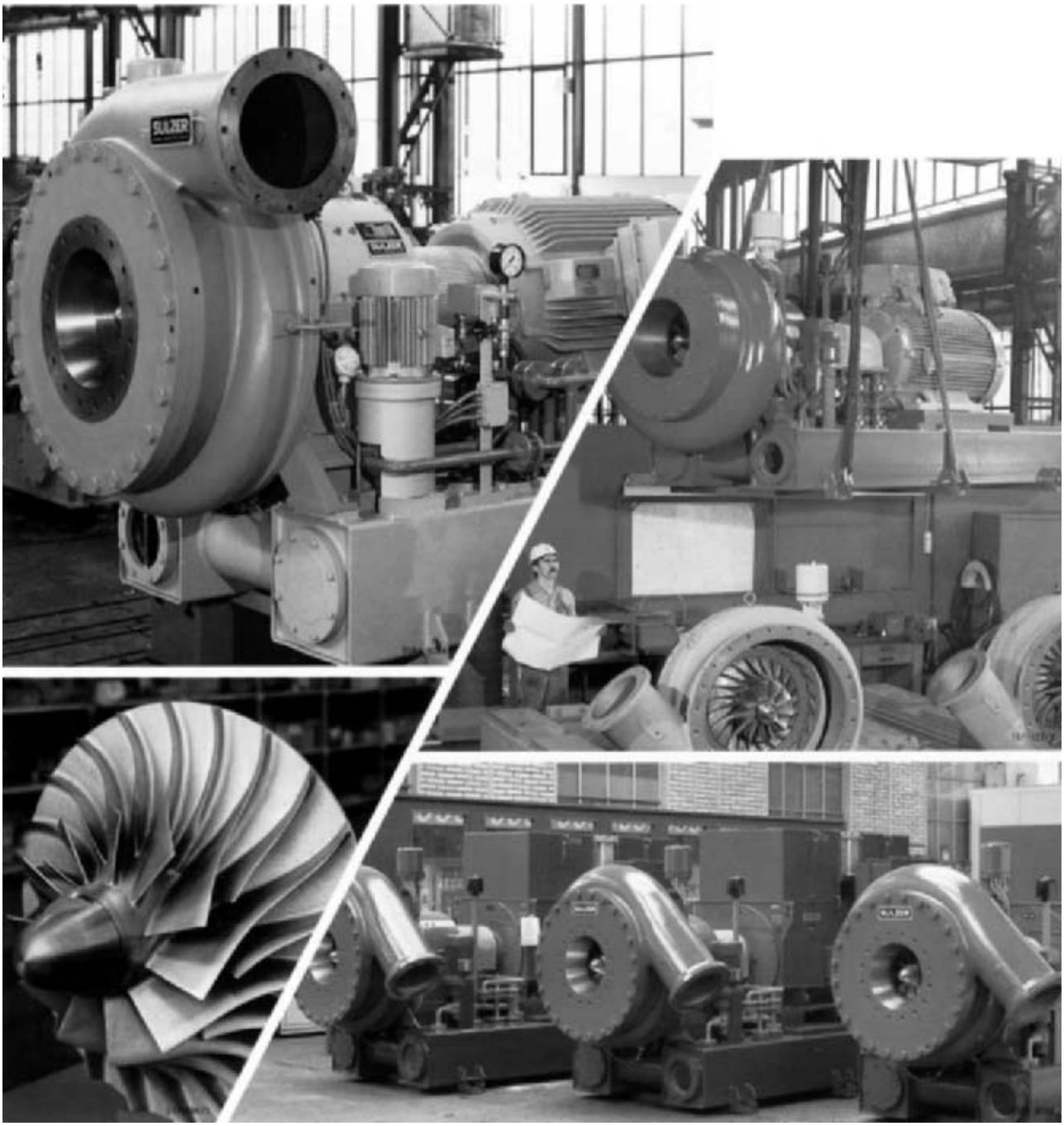
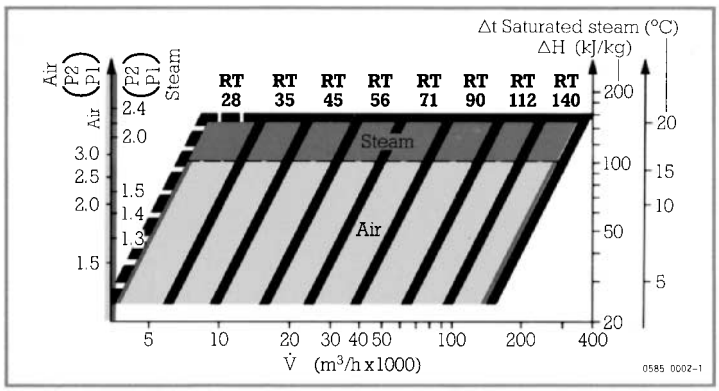
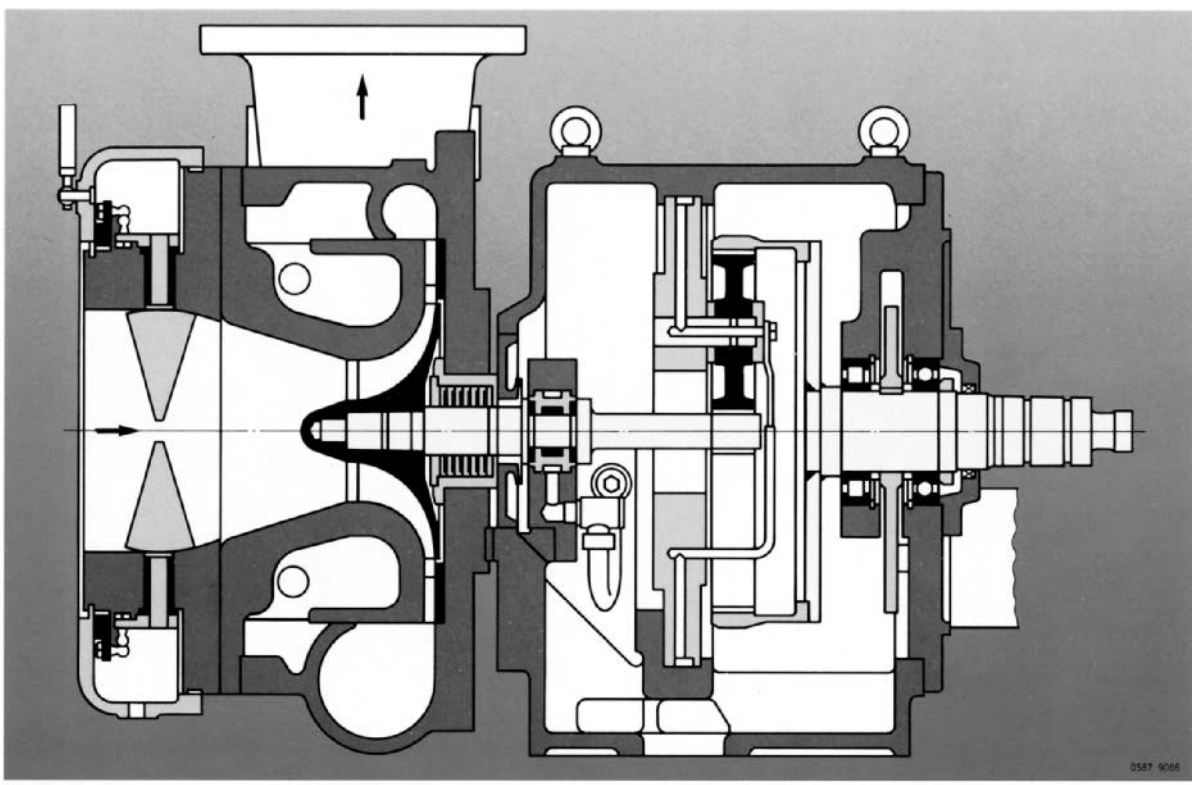
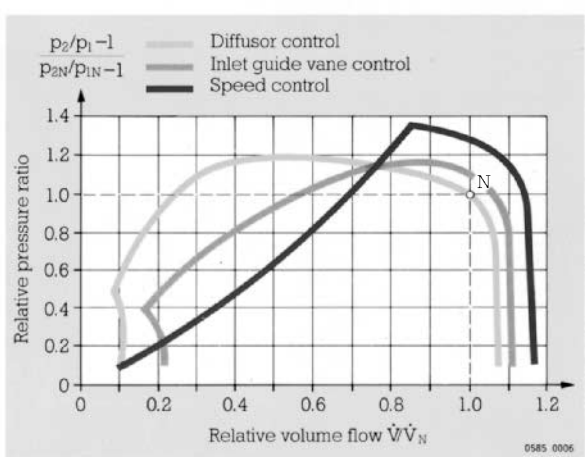


FIG. C-136 Single-stage turbocompressors offer compact design, good efficiency, adaptability to process requirements, and low noise emission. Additional stages are possible. Applications include mechanical vapor compression, effluent evaporation, sewage treatment, waste-heat utilization, chemical processes, vacuum generation, seawater desalination, compressed air systems, and flue gas desulfurization. (Source: Sulzer-Burckhardt.)





Series RT
Axial inlet. Direction of discharge nozzle to suit installation. Adjustable inlet guide vane or diffusor guide vane control. Multistage arrangement with single helical gear for high-pressure ratio also available.



Comparison of characteristics with different control devices.

FIG. C-137 Single-stage turbocompressors with this OEM incorporate eight frame sizes, for suction volumes from 5000 to 250,000 m³/h. (Source: Sulzer-Burckhardt.)

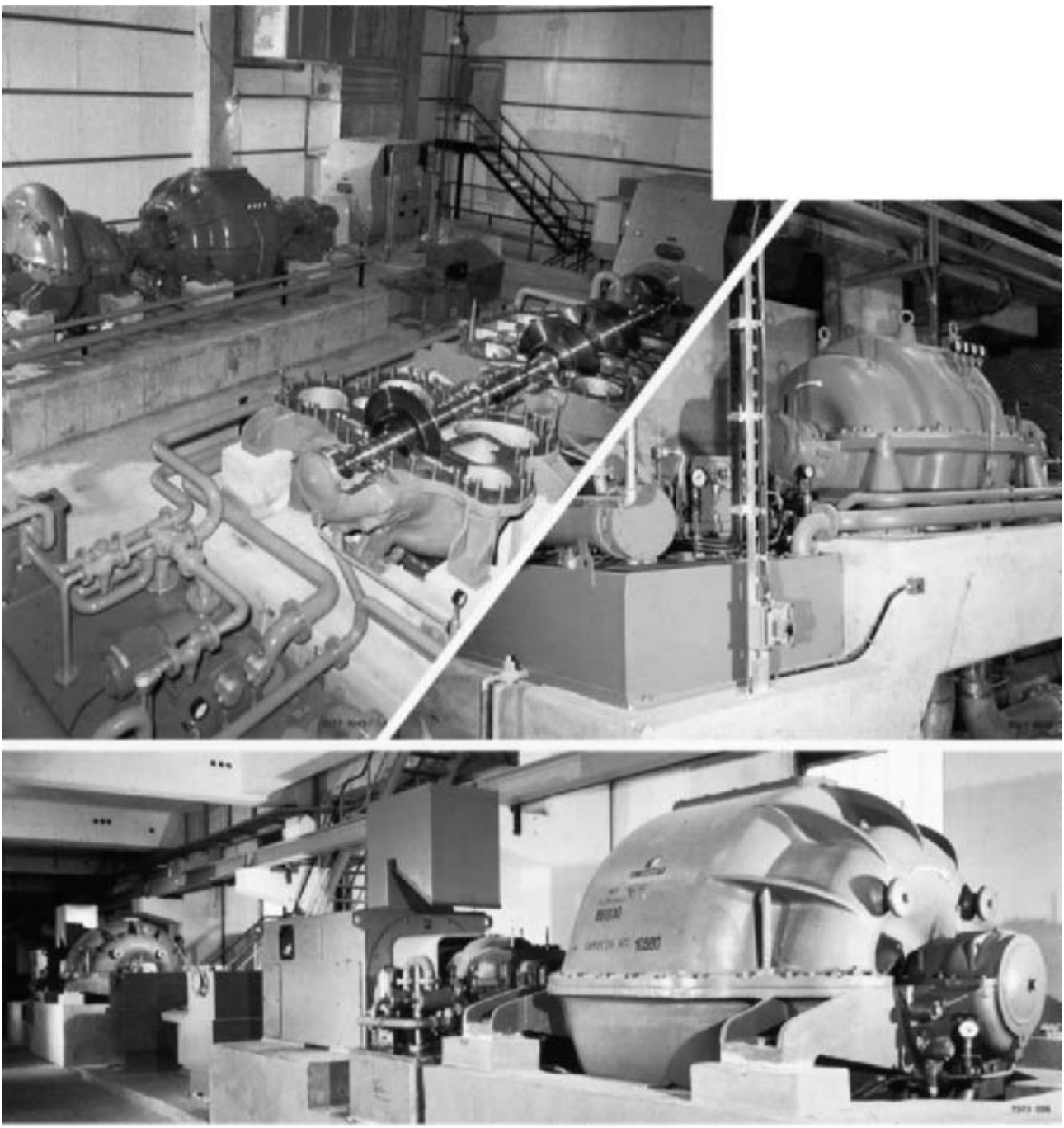
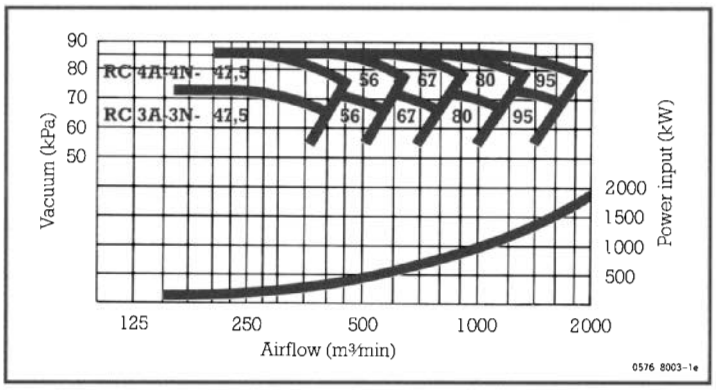
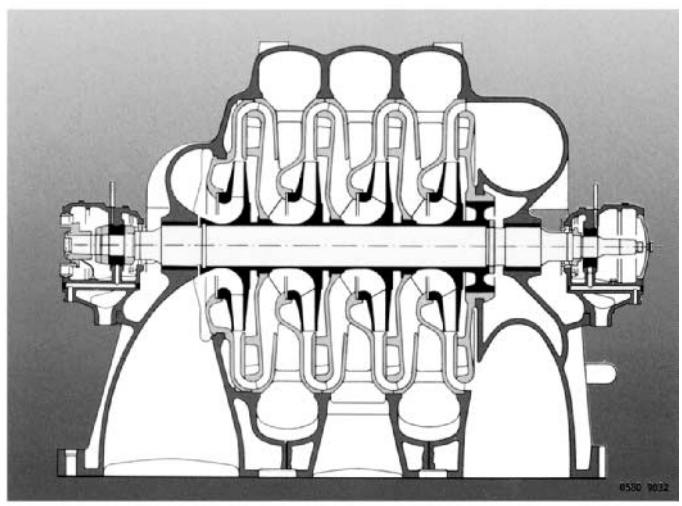


FIG. C-138 TURBAIR™ vacuum blowers offer compact design, several different vacuum levels, low water consumption, the best possible matching to performance requirements, a flat characteristic over a wide working range, an economical vacuum system, good reliability, and low maintenance costs. One application was specially developed for extracting water from paper, board, and pulp machines of all sizes. (Source: Sulzer-Burckhardt.)



Series RC
Equipped alternatively with three or four compression stages, thus permitting best possible adaptability to each condition.



TURBAIR vacuum system.

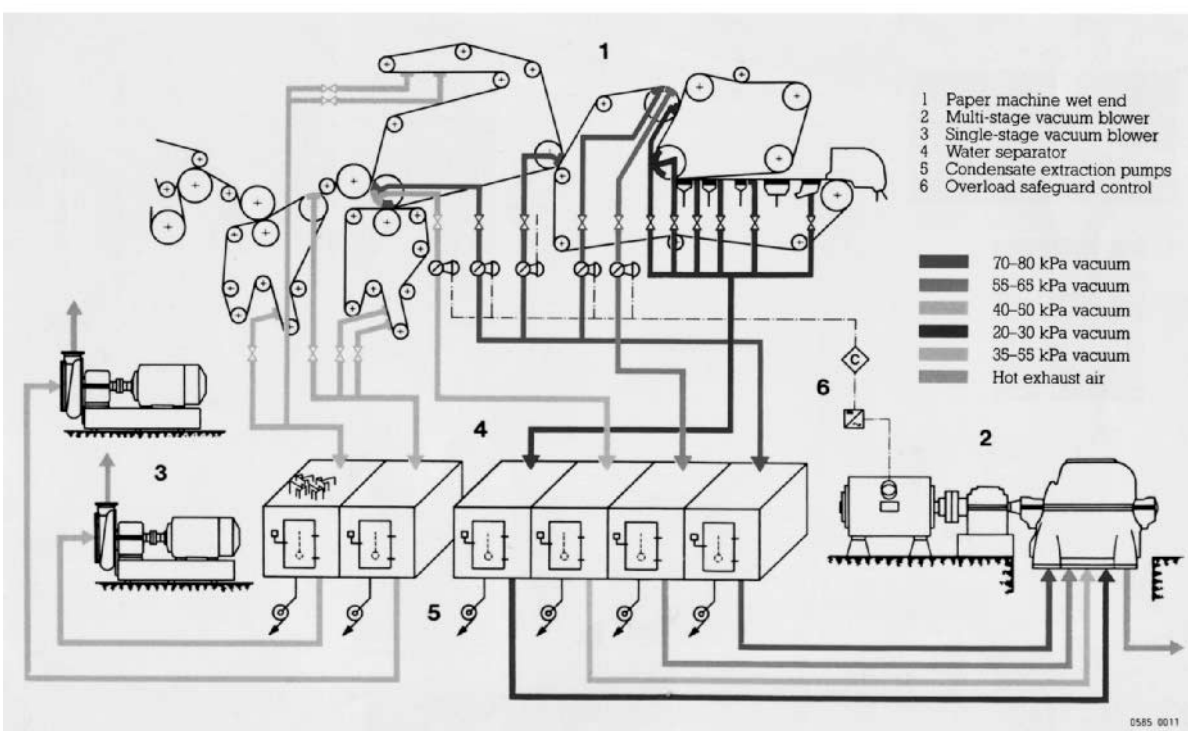


FIG. C-139 TURBAIR vacuum blowers with this OEM incorporate five frame sizes to cover practical vacuum requirements. (Source: Sulzer-Burckhardt.)

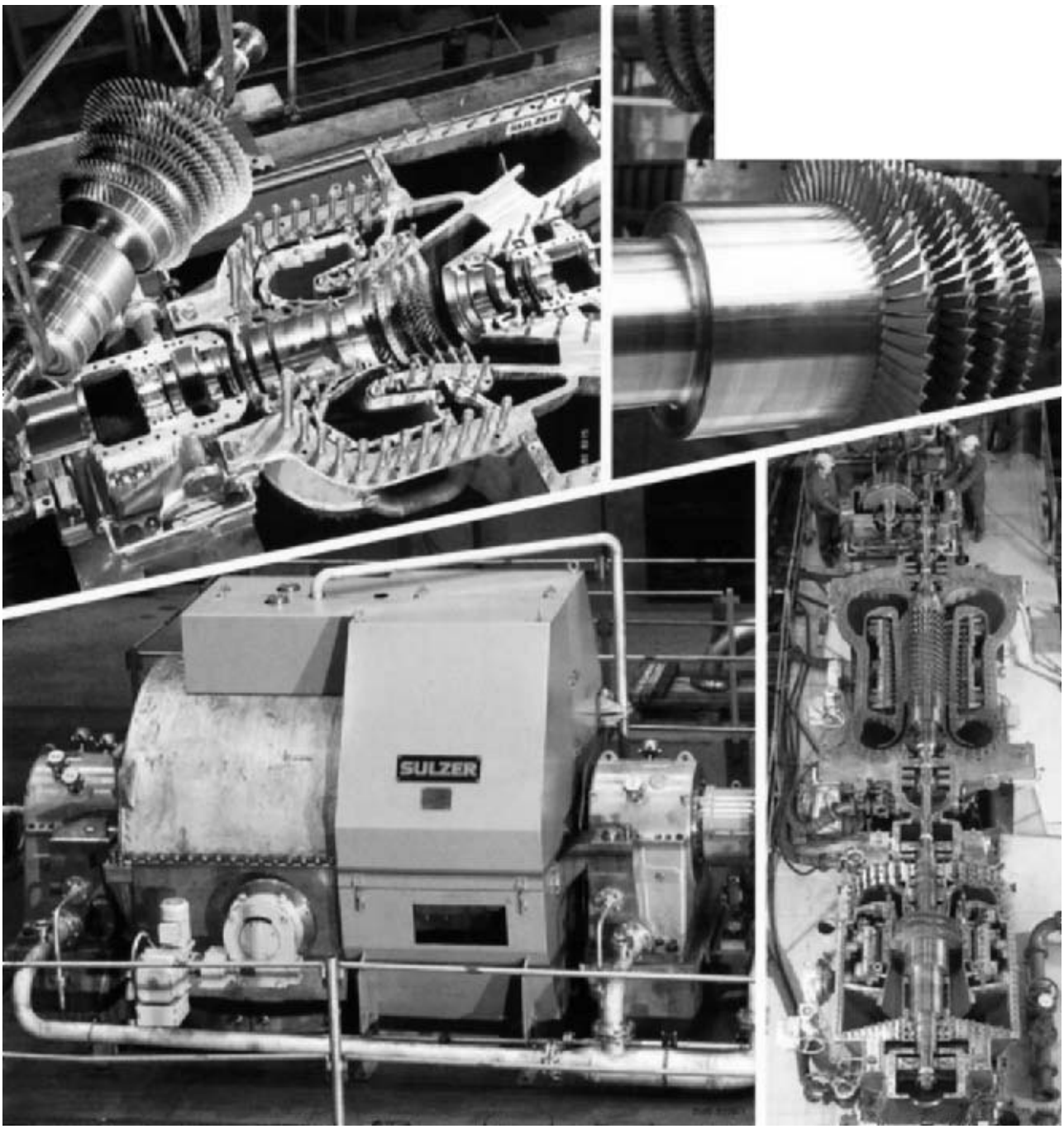
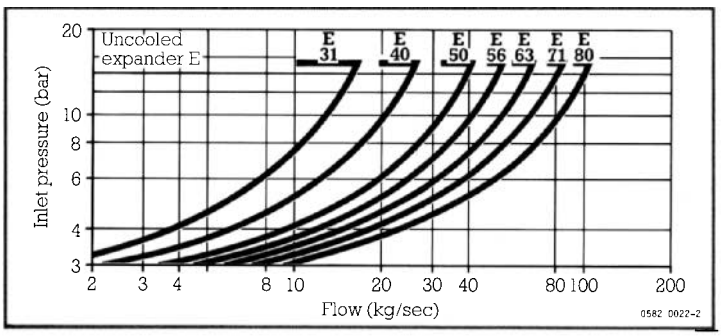
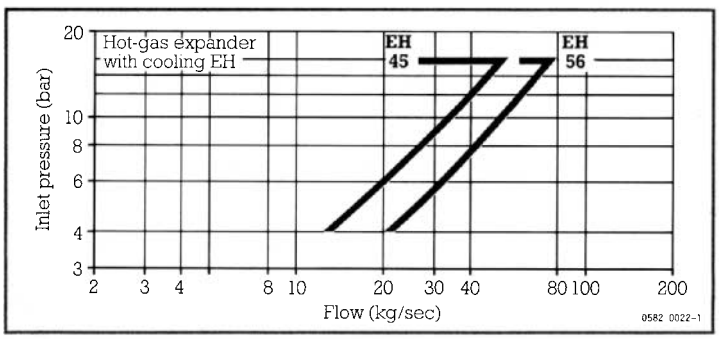
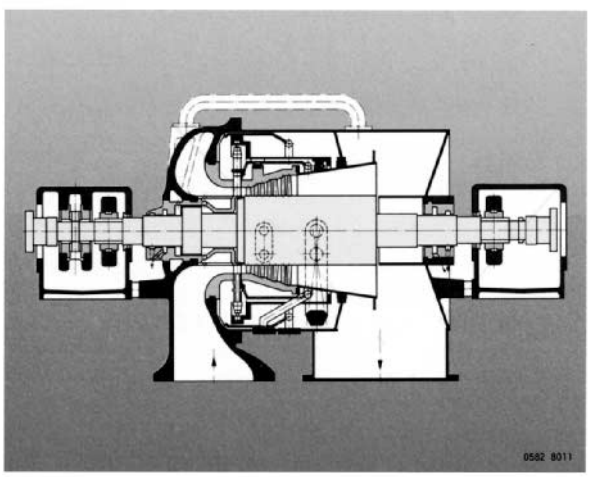


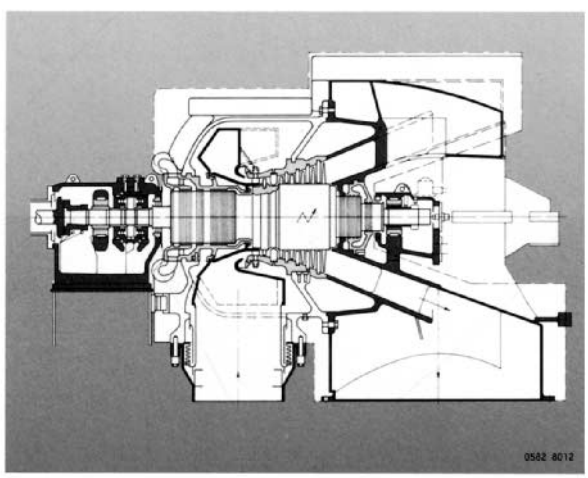
FIG. C-140 Expanders (turboexpanders) offer full arc admission, high efficiency, and robust design with a separate blade carrier, and are available with adjustable stator blades for first stage. Applications include nitric acid plants and power recovery. (Source: Sulzer-Burckhardt.)



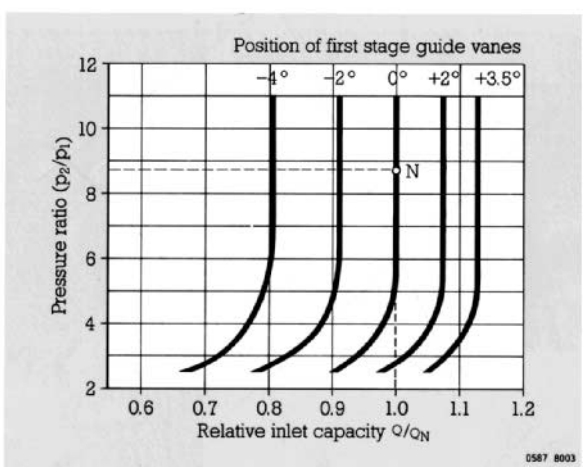


Series EH, cooled
For inlet temperatures up to 730 °C at 9 bar.
Design pressure 17 bar.





Series E, uncooled
For inlet temperatures up to 510 °C at 13 bar.
Design pressure 17 bar, power ratings up to 20 MW.



Tail gas expander performance curve.

FIG. C-141 Turboexpanders with this OEM incorporate eight frame sizes, with rotor diameters ranging from 31 to 80 cm. Materials are selected to suit the specific application. (Source: Sulzer-Burckhardt.)

The power input ratings of industrial applications for the compressors vary between 2000 and 90,000 kW. (See Fig. C-142.)

With a view to economical manufacturing and stocking of the major components, the range of compressors has been standardized. Most OEMs stock major components. That facilitates prompt delivery of machine parts such as rotor blades and stator blades, bearings, joints, etc., for service requirements. The systematic design of components over the whole range of sizes enables the compressor to be adapted optimally to the required operating conditions. Measurements conducted on the blading of various designs and sizes ensure exact conformity of the design data with the operating conditions.

Operating range. OEMs compete to offer the widest range of operating parameters. How they do so is specific to their individual design philosophies. This information source's axial compressor program comprises two type ranges:

Type A. Compressors with fixed stator blades (FIXAX) (see Fig. C-131).

Type AV. Compressors with adjustable stator blades on all or only some stages at the inlet (VARAX) (see Fig. C-131).

Each type range consists of 12 geometrically graduated sizes with rotor diameters extending from 40 to 140 cm. This completely covers a suction volume range of 70,000 to 1,250,000 m^3/h. (See Fig. C-143.)

The required compressor size and number of stages, together with the corresponding standardized overall length, are selected according to the suction volume and the thermodynamic head.

Type A (FIXAX). The A-type FIXAX compressor is generally used whenever the driver is a steam turbine, a split-shaft gas turbine, or a variable-frequency high-speed synchronous motor.

The required operating points can be attained by speed variation, and there is basically no need for adjustable stator blades. Fixed-blade machines are also selected for installations where only minor flow variations are required, or if the mass flow is adapted by variation of the suction pressure, as in aerodynamic test facilities, for example.

Type AV (VARAX). The AV series with adjustable stator blades permits a large stable operating range at constant speed. It is therefore used for constant-speed electric motor drive. Nevertheless, this type is being increasingly preferred for steam turbines and single-shaft gas turbines as well. In this particular case, the stator blade control either facilitates operation with limited speed control range (increased reliability of operation for certain turbine types) or, in combination with the speed control, provides an additional extension of the operating range and an improvement of the overall efficiency at part load. Furthermore, it offers the advantage of quick adaptation of the compressor to changed operating conditions without acceleration of the set—a characteristic that is of great interest for the periodic charging of air heaters in blast-furnace blowing plants. Many axial compressors are fitted with adjustable stator blades.

Stator blade setting with electric servomotor. For a great number of processes, the reference value of pressure or mass flow is selected at the process control panel and transmitted to the compressor servomotor. In this case this servomotor is of the electric type with the additional possibility of manual adjustment of the stator blades.

Stator blade control with hydraulic servomotor. If the process calls for automatic pressure or mass flow control, the stator blade adjusting mechanism will be operated by a synchronized pair of hydraulic servomotors.

With the exception of the stator blades and their adjusting mechanism, the same standardized construction elements are used for both FIXAX and VARAX types.

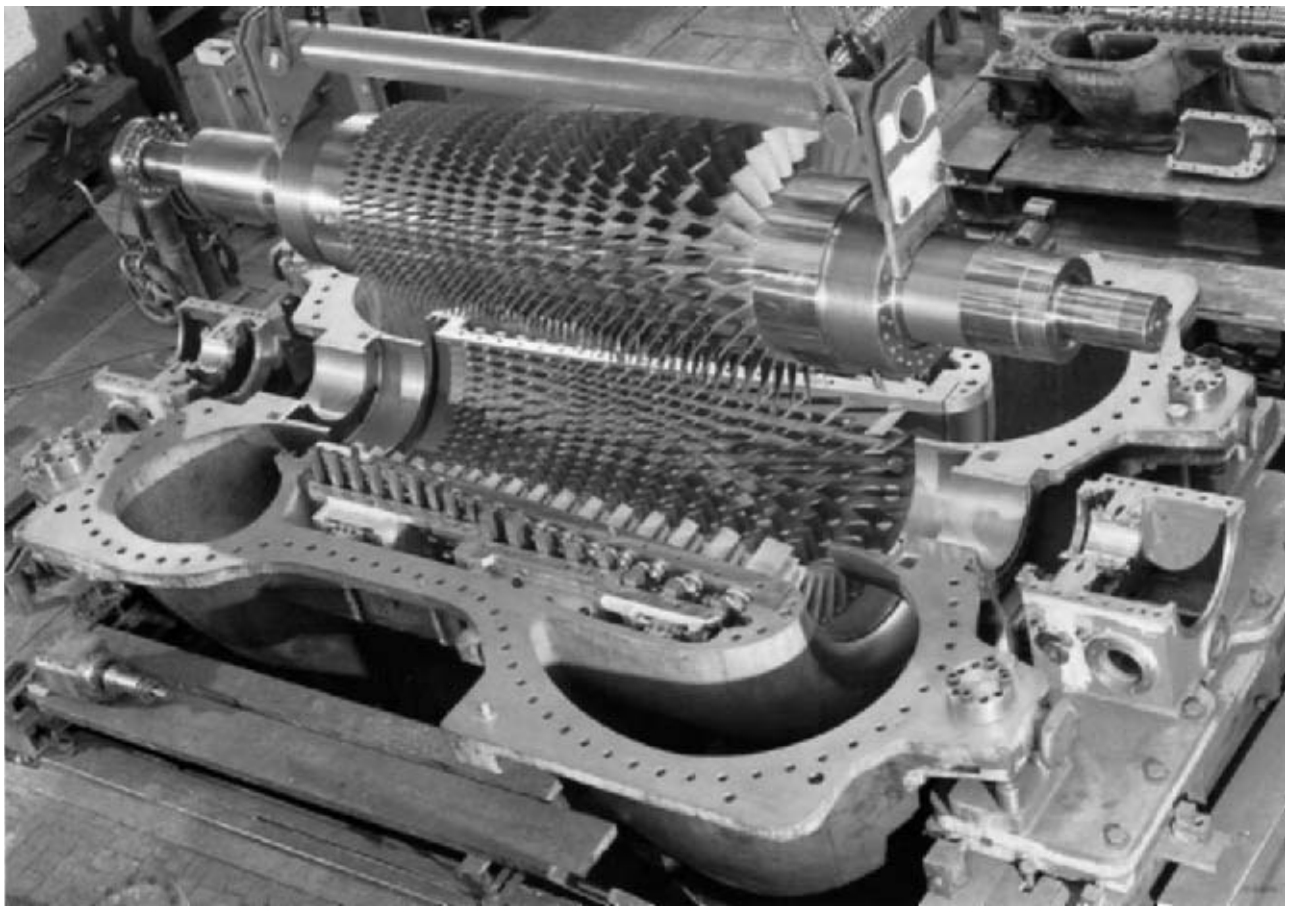


FIG. C-142 Axial compressor, type AV 100–16, during erection. Two identical steam-turbine–driven machines are supplying air to the blast furnace of a British steel works. Suction volume 560,000 Nm^3/h, discharge pressure 6.2 bar, power input 52,000 kW each. (Source: Sulzer-Burckhardt.)

Performance data. Figures C-144 through C-149 (note type designation) facilitate the selection of the

Compressor size	Nominal diameter	D (cm)
	Number of stages	z (–)
	Power input	P (kW)
	Speed	n (rev/min)
	Discharge temperature	T_2 (K)/t_2 (°C)
Using	Capacity	$\dot{m}$ (kg/s)
	Suction pressure	p_1 (bar abs)
	Suction temperature	T_1 (K)/t_1 (°C)
	Relative humidity of the air or gas	φ_1 (%)
	Discharge pressure	p_2 (bar abs)
	Molecular mass	M (kg/kmol)
	Isentropic exponent	k (–)
	Compressibility factor	Z (–)

The following factors and symbols are also used for the calculation:

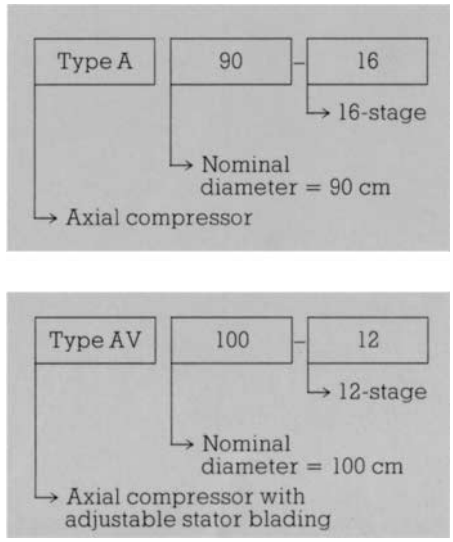


FIG. C-143 Type designation. (Source: Sulzer-Burckhardt.)

	Suction volume (actual)	$\dot{V}_1$ (m³/s)
	Absolute humidity	x (–)
	Polytropic efficiency	η_P (%)
Indices	Suction branch	1
	Discharge branch	2
	Dry	t
	Wet	f

Design features. The basic design offers (see Fig. C-150):

- Cast casing and separate blade carrier
- Casing supported by means of pendulum supports (minimum expansion forces) or feet with keyways for the smaller frame sizes
- Solid or hollow rotors, smooth-running characteristics with integrated balancing pistons
- Blading with optimal aerodynamic characteristics—with good efficiency and specific capacity, low stressing, favorable control characteristics
- Blade vibrations minimized; blade profiles and blade fixation are designed accordingly
- Adjustable stator blades—of standard design—for optimum flow control
- Maintenance-free, oil-free stator blade adjusting mechanism that is also protected against the ingress of contaminants
- Various bearing options
- Various shaft seal options

Casings, bearing pedestals. Depending on the type of gas and the design pressure, the casings are made of gray cast iron, nodular cast iron, or cast steel. The cast design incorporates a rigid construction, effective noise attenuation, and aerodynamically favorable layout of the respective ducting. The suction and delivery branches are usually routed vertically downward. In cases where, due to the composition of the gas and/or the pressure level, steel casings are mandatory, a welded construction

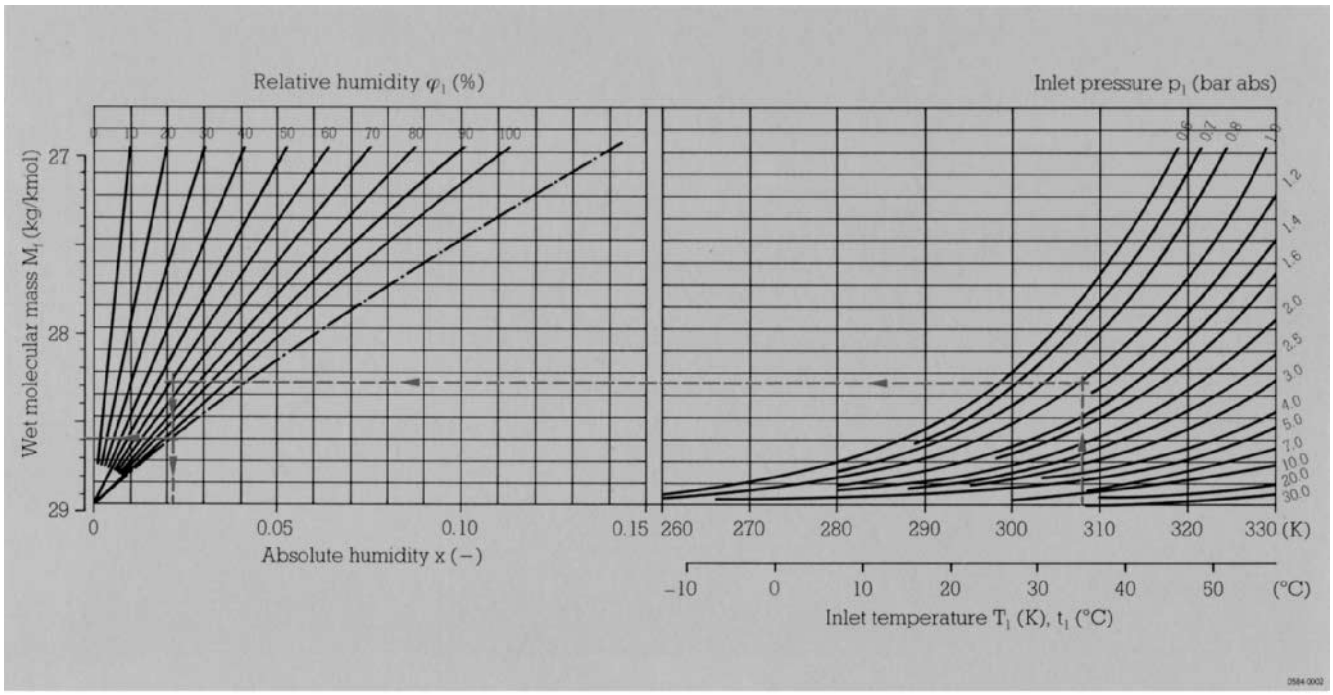


FIG. C-144 Determination of the absolute humidity *x* and the molecular mass M_f of the wet gas. (Source: Sulzer-Burckhardt.)

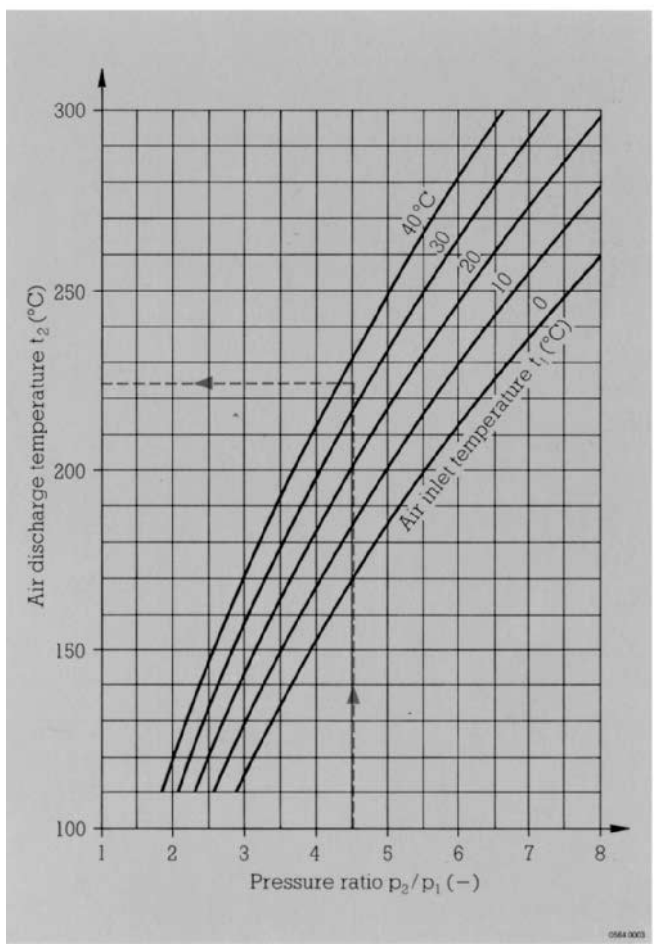
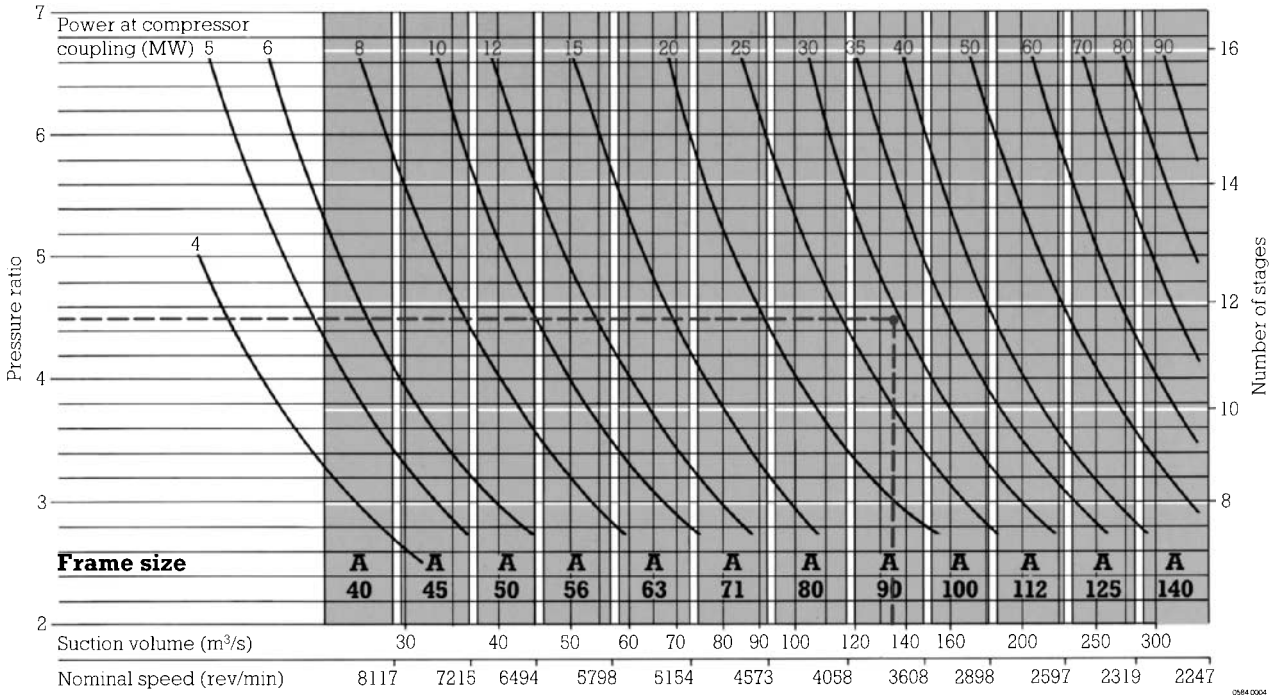


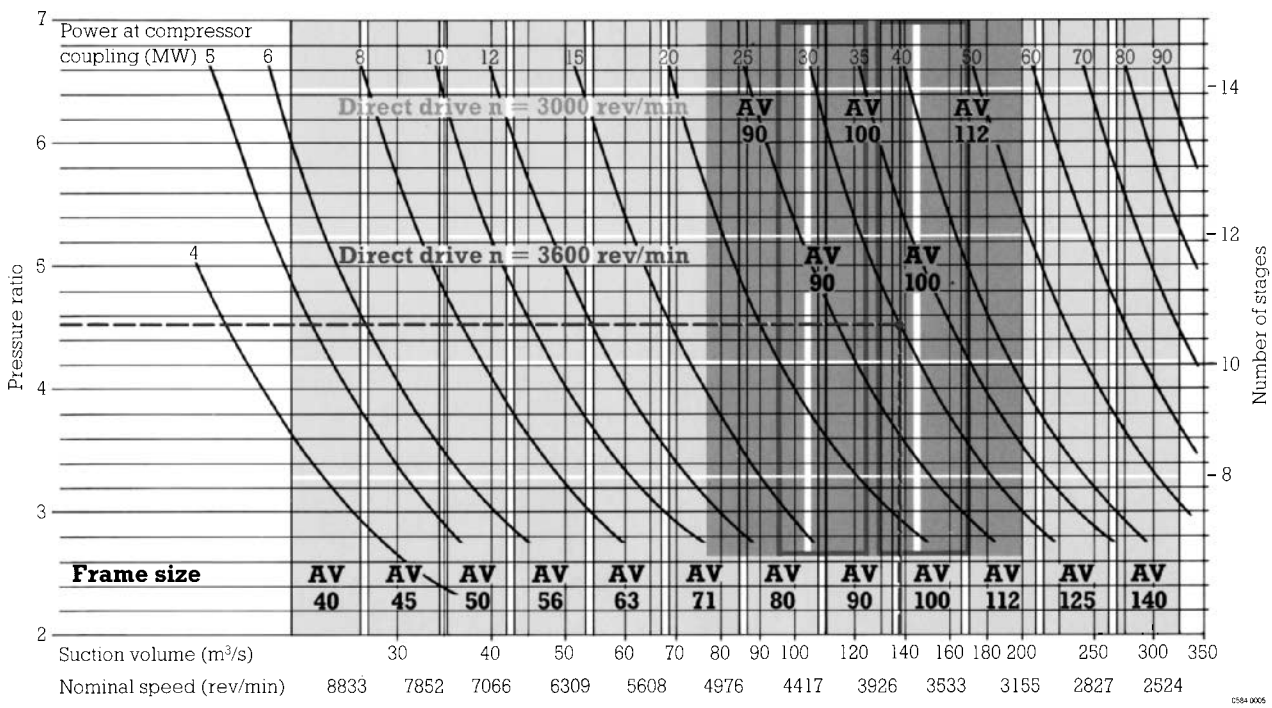
FIG. C-145 Determination of the discharge temperature t_2. (Source: Sulzer-Burckhardt.)

Figs. C-146 and C-147 are valid for air at suction conditions 1 bar, 20°C, 70% relative humidity.



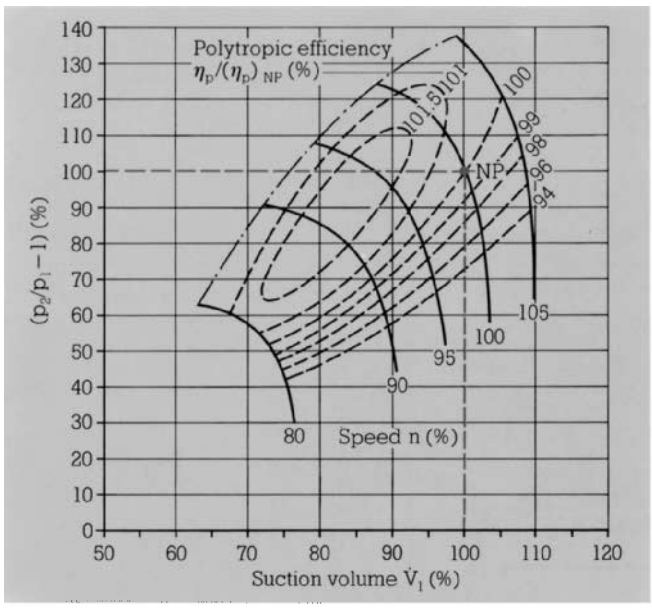
Selection of the compressor size: nominal diameter D (cm) = frame size as a function of the actual wet suction volume flow $\dot{V}_f$ (m³/s). Determination of the approximate number of stages z, nominal speed n (rev/min) and power input P (MW).

FIG. C-146 Performance data for type A (FIXAX). (Source: Sulzer-Burckhardt.)



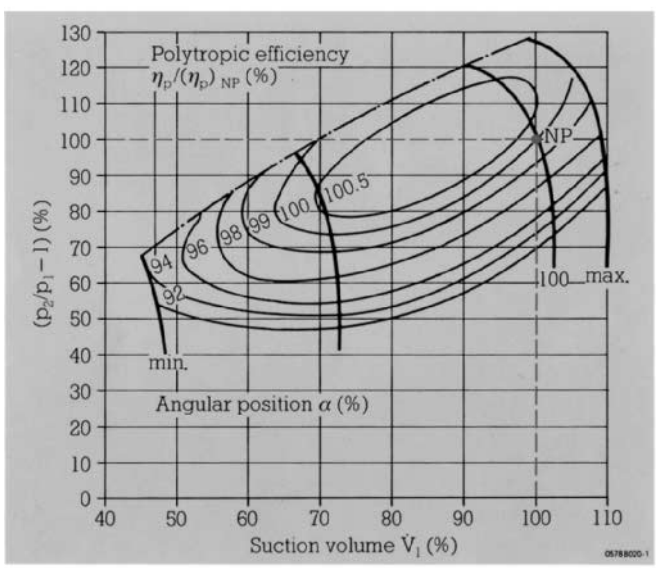
Selection of the compressor size: nominal diameter D (cm) = frame size as a function of the actual wet suction volume flow $\dot{V}_f$ (m³/s). Determination of the approximate number of stages z, nominal speed n (rev/min) and power input P (MW).

FIG. C-147 Performance data for type AV (VARAX). (Source: Sulzer-Burckhardt.)



148

Operating characteristics for a compressor, type A, with fixed stator blades and variable speed:
NP = Reference point (100%) = Design point;
p_2/p_1 = Pressure ratio.
Valid for: constant gas data, constant inlet temperature.



149

Operating characteristics for a compressor, type AV, with adjustable stator blades and constant speed:
NP = Reference point (100%) = Design point;
p_2/p_1 = Pressure ratio;
α = Angular setting of the stator blades.
Valid for: constant gas data, constant inlet temperature.

Depending on the specific process requirements, such as extremely high turn-down, maximum efficiency within a certain range, flat or steep p-V curve, etc., various parameters of the blading may be changed in order to adapt the characteristics to such special conditions.

Selection and performance calculation of an axial compressor

	Calculation example Air compressor, type A
Given	
Capacity	$\dot{m}_t = 146.2$ kg/s
Suction pressure	$p_1 = 1$ bar abs
Suction temperature	$T_1 = 308$ K, $t_1 = 35$ °C
Relative humidity	$\varphi_1 = 60\%$
Discharge pressure	$p_2 = 4.5$ bar abs
Dry molecular mass	$M_t = 28.95$ kg/kmol
Isentropic exponent c_p/c_v	$k = 1.4$
Compressibility factor	$Z = 1$
Calculation instructions	
1. Determination of the absolute humidity x and wet molecular mass M_f (from T_1, p_1, φ_1) with Fig. C-144	$x = 0.021$ $M_f = 28.6$ kg/kmol
2. Calculation of the wet mass flow $\dot{m}_f = \dot{m}_t(1 + x)$	$\dot{m}_f = 146.2\,(1+0.021)$ $= 149.3$ kg/s
3. Determination of the actual suction volume $\dot{V}_1 = \frac{\dot{m}_f \cdot T_1}{F^* \cdot M_f \cdot p_1}$ [m³/s]	$\dot{V}_1 = 133.7$ m³/s
4. Determination of the discharge temperature t_2 with Fig. C-145	$t_2 = 225$ °C
5. Selection of the compressor size (nominal diameter D) and number of stages Z, determination of speed n and power input P with Fig. C-146	D = 90 cm Z = 12 n = 3600 rev/min P = 29 MW
6. Type designation	A 90–12
7. Selection of compressor size AV with Fig. C-147	
7.1 Steam turbine drive, i. e. no speed restriction (speed $n_s = n_n \cdot \sqrt{Z_{th}/Z_s}$) $= 3926 \cdot \sqrt{10.6/11.0}$ Type designation	D = 90 cm Z = 11 n = 3854 rev/min P = 29 MW AV 90–11
7.2 Electric motor direct drive 3600 rev/min Type designation	D = 100 cm Z = 11 n = 3600 rev/min P = 29 MW AV 100–11
7.3 Electric motor direct drive 3000 rev/min $Z_s = Z_{th} \cdot \left(\frac{n_n}{n_s}\right)^2 = 10.6 \cdot \left(\frac{3533}{3000}\right)^2$ $= 14.7$ Type designation	D = 100 cm Z = 15 n = 3000 rev/min P = 29 MW AV 100–15

n_n = Nominal speed
n_s = Selected speed for electric motor direct drive or corrected to arrive at a whole number of stages
Z_{th} = Theoretical number of stages on the basis of Figs. C-146 and C-147
Z_s = Selected number of stages to arrive at a whole number or to account for reduced speed in example 7.3. The reduced rotational speed imposed by the speed of the two-pole motor reduces the tip speed proportionately. This must be compensated by increasing the number of stages to produce the desired pressure ratio.

* Factor F, if mass $\dot{m}$ given in Nm³/h = 33505
SCFM (14.7 psia, 60 F) = 20844
SCFM (14.7 psia, 70 F) = 21245
kg/s = 12.027

Performance data [type A (FIG. C-148), type AV (FIG. C-149)] for selection and performance calculation of an axial compressor. (Source: Sulzer-Burckhardt.)

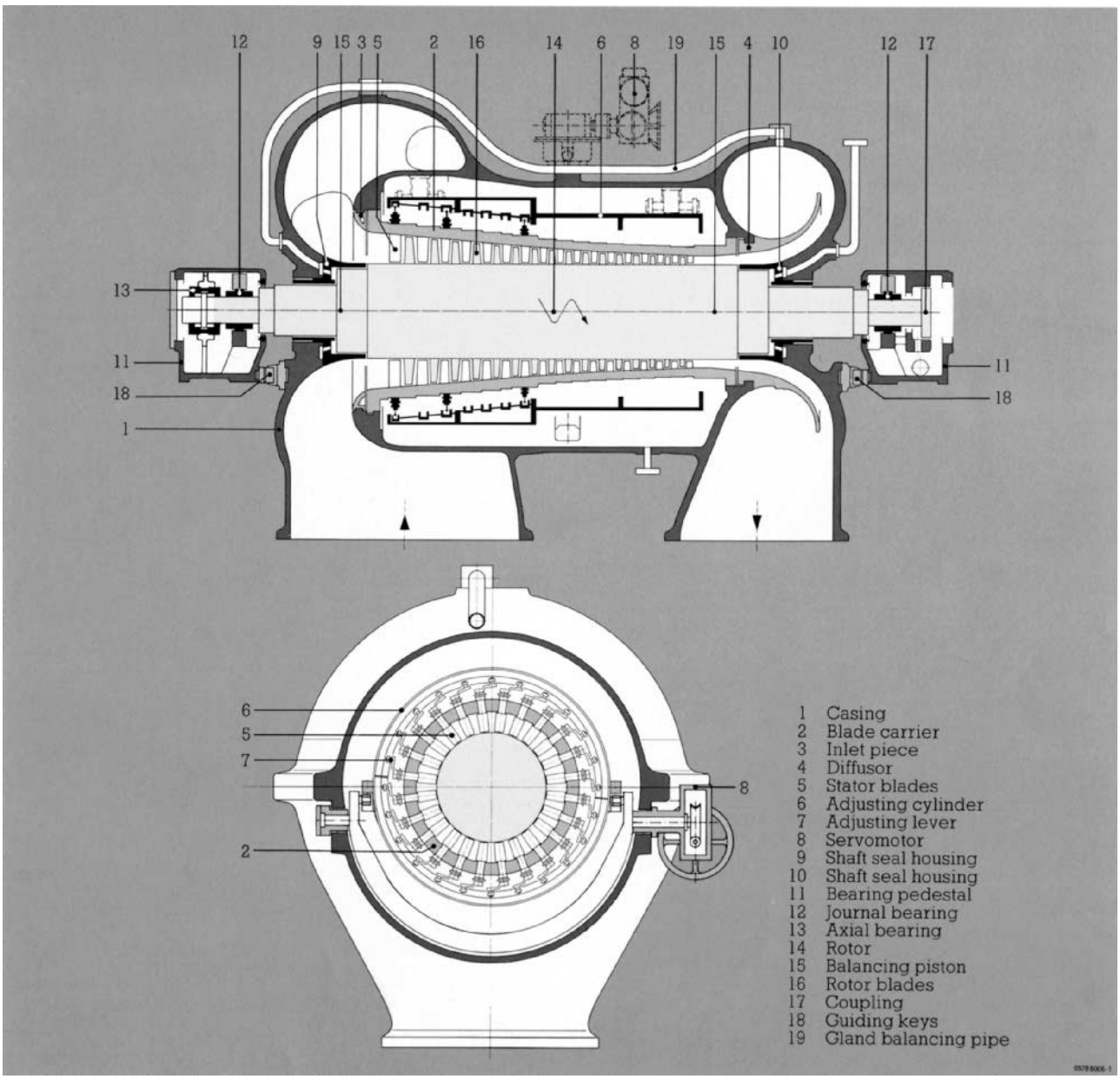


FIG. C-150 Longitudinal and cross-sectional view of an AV compressor. (Source: Sulzer-Burckhardt.)

can be specified. The suction branch may then be axial, or both suction and discharge may be routed upward or downward. In the vertical central plane, the casing is aligned by two keyways; it is equipped with four supporting feet. It is fixed at one end in the axial direction by one pair of feet. The other pair of feet of large frame sizes rests on pendulum supports with spherical contact surfaces. As a result of this, the casing can take up the thermal expansion in both the axial and lateral direction without difficulty. This feature is particularly advantageous in the case of

light steel foundations, as may be found, for instance, on an offshore platform. On small frame sizes the pendulum supports are replaced by sliding keyways.

Blade carrier, casing inserts. The blade carrier inserted in the casing is centered on both the suction and discharge side, and is able to expand freely in the axial direction. The diffusor and the gland inserts are also fitted as separate parts in the casing.

The double-casing design with outer casing and blade carrier offers various advantages:

- Rigid casing construction; the clearances in the blade duct are not influenced directly by external pipe forces.
- Simple fitting of the blades and assembly of the casing parts; the top half of the casing can be raised without dismantling the blade-adjusting mechanism.
- Possibility of fitting different blade carriers, for adapting the blade duct and thus the compressor characteristics to greatly changed operating conditions.
- Optimal protection of the adjusting mechanism in the space between the casing and blade carrier; the space is kept under suction pressure to safeguard the adjusting mechanism against condensation and corrosion attack.

Rotor. The rotors are usually of forged solid design. In the case of larger machines or if the moment of inertia must be minimized to limit the power requirement when running up with an electric motor, welded hollow rotors may be used. Integrated balancing pistons at both ends of the rotor facilitate an equalization of the axial thrust. The careful balancing of the rotor at full speed results in highly smooth running characteristics. If necessary, balancing can also be effected in the casing.

The labyrinth strips are caulked in the rotor.

Blading. Blading with a high degree of reaction, i.e., the increase in pressure takes place exclusively in the impellers, is employed for the compression of lighter gases such as helium or hydrogen.

For all other applications, such as the compression of air, blading with a lower percentage reaction is adopted. The increase in pressure is distributed to the rotor and stator blades. This enables the following design advantages to be realized:

- Higher efficiency with lower aerodynamic loading
- Widest possible control range with high part-load efficiency at constant speed
- Largest possible suction volume at given speed
- Increased reliability of operation due to larger radial blade clearances and the omission of guide vane sealings
- Steeper pressure volume characteristics, especially suitable for capacity control, for the parallel operation of different compressors, for refrigeration processes, and for exact adjustment of the blow-off line

The rotor and stator blades are normally made of 13 percent chrome steel and machined. When handling highly contaminated aggressive air or corrosive gases, alloys with higher chromium and nickel content may be used. The rotor blades have rhomboidal roots and are firmly braced in an exactly defined position in the peripheral grooves of the rotor. This is of particular importance for their vibration-wise design. The fixed stator blades are provided with a rectangular foot. The adjustable stator blades are made in one piece with a cylindrical shaft. The latter is seated in a bearing bush in the blade carrier. The high damping characteristics of this seating arrangement practically eliminate the potential for high vibration amplitudes associated with the stator blades.

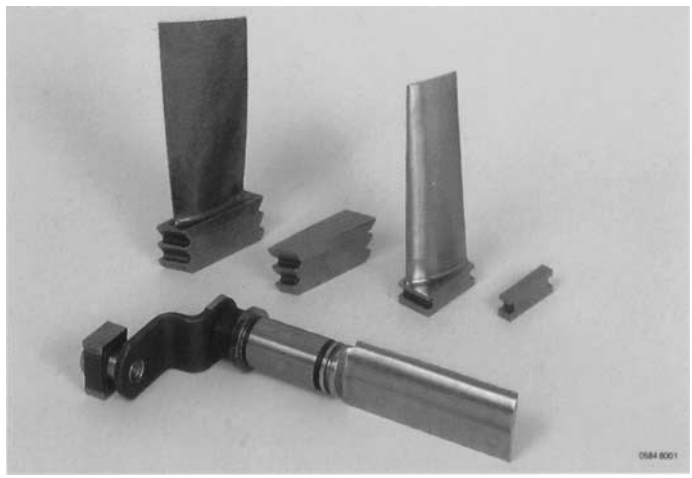


FIG. C-151 Adjustable stator blade, rotor blade, and fixed stator blade with intermediate piece. (Source: Sulzer-Burckhardt.)

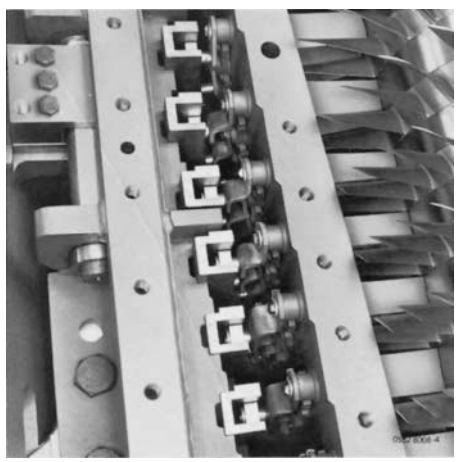


FIG. C-152 Stator blade adjusting mechanism. (Source: Sulzer-Burckhardt.)

Stator blade adjusting mechanism for the AV types. The adjusting mechanism is located in an annular space between casing and blade carrier, and is thus well protected against contaminants and moisture. It is maintenance-free and does not require any lubrication. (See Figs. C-151 and C-152.)

Servomotors. Two types of servomotors are available, electric and hydraulic (Figs. C-153 and C-154).

Automatic mass flow, volume or pressure control. The adjusting mechanism is operated by means of two hydraulic servomotors that are affixed laterally to the casing. One of the servomotors is equipped with a positioning transmitter and the second operates hydraulically in parallel.

The linear movement of the servomotor piston rods is transmitted directly to the adjusting cylinder by way of two ball and socket joints.

Remote setting of reference value. One single electric servomotor is attached laterally to the bottom half of the casing. Its driving shaft actuates a pivoted fork

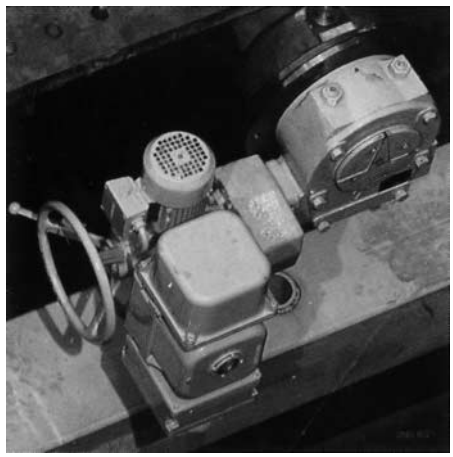


FIG. C-153 Electric servomotor. (Source: Sulzer-Burckhardt.)

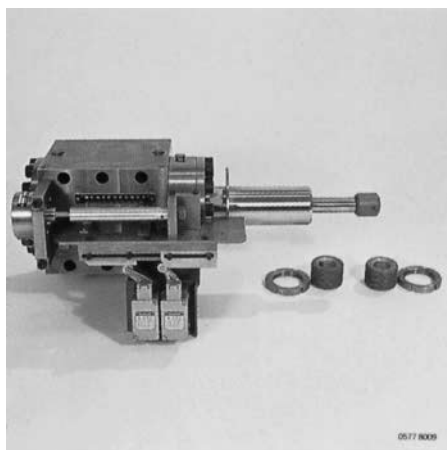


FIG. C-154 Hydraulic servomotor. (Source: Sulzer-Burckhardt.)

positioned on either side of the casing in maintenance-free bearings. This fork in turn moves the adjusting cylinder in the axial direction.

Adjusting cylinder. The adjusting cylinder of welded design can move in the axial direction and is dry-seated. There is no restriction of heat expansion in any direction. U-shaped guide rings are provided on the inner side in which the adjusting levers are engaged.

Stator blades. The adjusting levers provided on the end of each stator blade shaft are connected to the guide rings of the adjusting cylinder by means of pivoting sliders. The axial movement of the cylinder is converted into a rotating movement of the stator blades.

The self-lubricating bearing bushes of the blade shafts are seated in the radial holes of the blade carrier. O-ring packings prevent the ingress of contaminants into the stator blade seating.

Shaft seals. Labyrinth seals are used for the standard models. The stainless steel labyrinth strips are caulked on the rotor and are replaceable. In case of rubbing

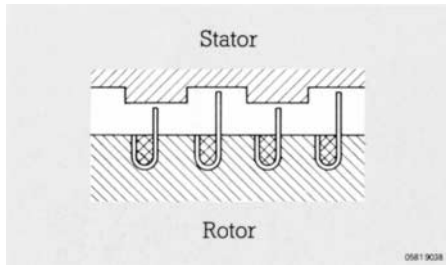


FIG. C-155 Labyrinth shaft seals. (Source: Sulzer-Burckhardt.)

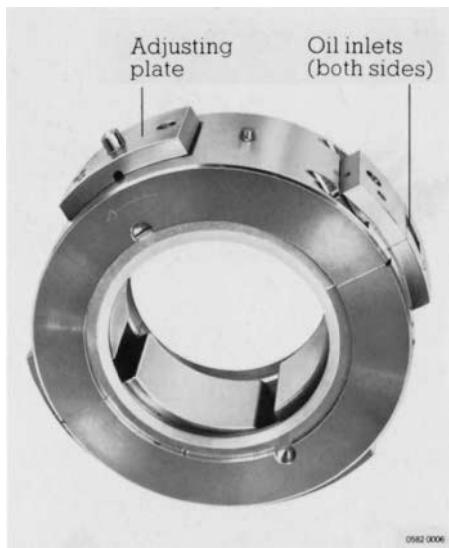


FIG. C-156 Multisegment journal bearing with four tilting pads. (Source: Sulzer-Burckhardt.)

due to unbalance, the friction-induced heat is immediately passed to the massive stator, thus avoiding distortion of the rotor. Gas-tight shaft seals and standstill seals can be fitted for special requirements. See Fig. C-155.

Journal and axial bearings

Journal bearings. In the standard version, i.e., with the compressor rotor solidly coupled and the rotor thrust transferred to the axial thrust bearing of the prime mover or the gear, the bearing housings are equipped only with journal bearings. Two-lobe bearings are provided for the lower speed range; tilting pad journal bearings are generally used for the higher speeds of the smaller frame sizes for reasons of stability. The slight curvature of the adjusting plates allows the bearings to be set accurately on erection. The bearings are firmly held in position by the bearing housing top half. See Figs. C-156 through C-158.

Two-lobe bearings are suitable for both senses of rotation, while tilting pad bearings are essentially for only one direction, although they can tolerate running backward with a somewhat reduced load capacity.

Axial thrust bearings. If a flexible coupling is selected between driver and driven machine, the bearing housing can accommodate the necessary tilting pad thrust bearing. The purpose of this bearing is to absorb the remaining thrust of the machine and any significant axial friction thrust of the coupling due to sharp

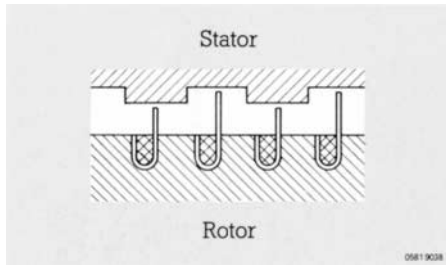


FIG. C-155 Labyrinth shaft seals. (Source: Sulzer-Burckhardt.)

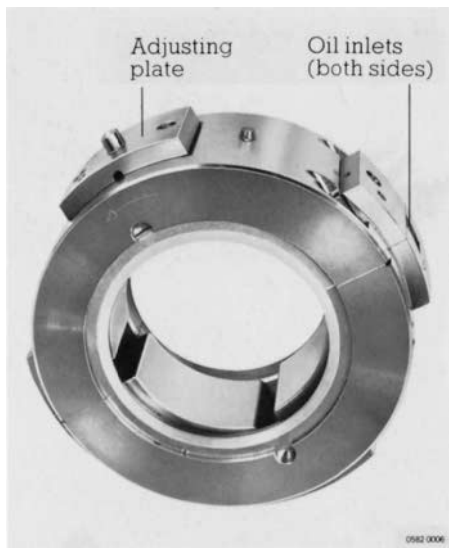


FIG. C-156 Multisegment journal bearing with four tilting pads. (Source: Sulzer-Burckhardt.)

due to unbalance, the friction-induced heat is immediately passed to the massive stator, thus avoiding distortion of the rotor. Gas-tight shaft seals and standstill seals can be fitted for special requirements. See Fig. C-155.

Journal and axial bearings

Journal bearings. In the standard version, i.e., with the compressor rotor solidly coupled and the rotor thrust transferred to the axial thrust bearing of the prime mover or the gear, the bearing housings are equipped only with journal bearings. Two-lobe bearings are provided for the lower speed range; tilting pad journal bearings are generally used for the higher speeds of the smaller frame sizes for reasons of stability. The slight curvature of the adjusting plates allows the bearings to be set accurately on erection. The bearings are firmly held in position by the bearing housing top half. See Figs. C-156 through C-158.

Two-lobe bearings are suitable for both senses of rotation, while tilting pad bearings are essentially for only one direction, although they can tolerate running backward with a somewhat reduced load capacity.

Axial thrust bearings. If a flexible coupling is selected between driver and driven machine, the bearing housing can accommodate the necessary tilting pad thrust bearing. The purpose of this bearing is to absorb the remaining thrust of the machine and any significant axial friction thrust of the coupling due to sharp

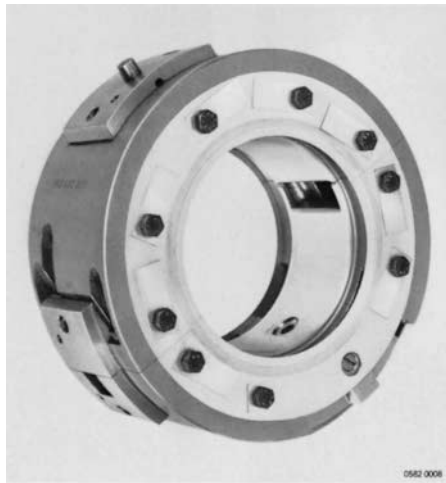


FIG. C-157 Two-lobe journal bearing. (Source: Sulzer-Burckhardt.)

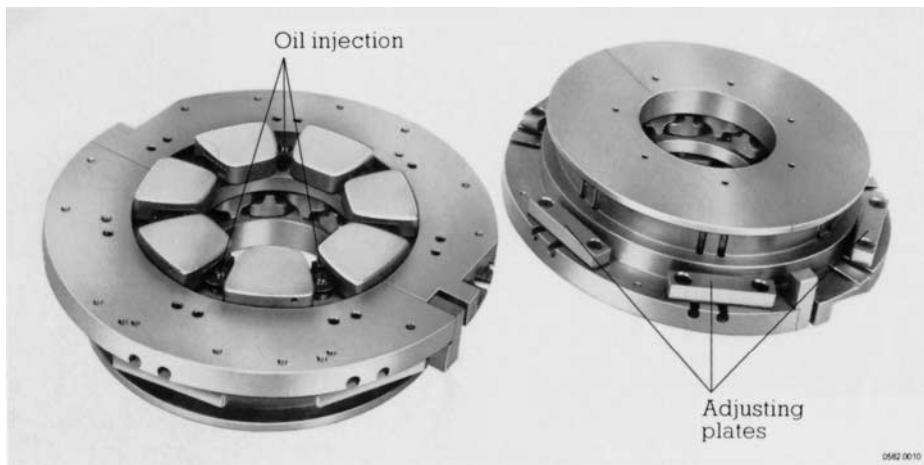


FIG. C-158 Kingsbury-type thrust bearing with self-equalizing pads with directed lubrication. (Source: Sulzer-Burckhardt.)

temporary differential expansion between rotor and casing. To provide easy access and reduce the overhang, it is preferably mounted on the free shaft end.

The tilting pads are supported on load-equalizing segments that allow angularity of the shaft up to 0.3 percent.

Because the tilting pads are supported eccentrically, thrust bearings are suitable for only one direction but tolerate a reversed rotation at a reduced load capacity.

Solid coupling. It is this OEM's normal practice to make extensive use of solid couplings allowing the use of only one axial thrust bearing for single- or multiple-casing arrangements.

An intermediate shaft, flexible enough to allow for considerable misalignment, is inserted between the two shaft ends of the machines to be coupled together (Fig. C-159). In case of motor-driven units, the normal technique is to use single helical gears provided with thrust collars on the pinion shaft, as shown in Figs. C-160 and C-161. The thrust collars not only neutralize the axial thrust created by the meshing of the teeth cut at an angle to the axis of the shaft, but also transmit

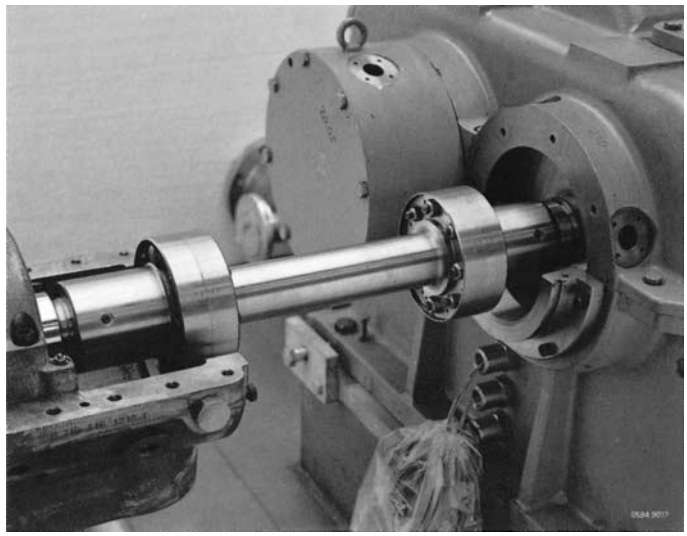


FIG. C-159 The solid quill-shaft coupling conforms to API 671 standard and consists of the quill shaft and the two hubs hydraulically fitted onto the shaft ends of the connected machines. On each coupling side, an equal number of tie bolts for axial fixation and tapered dowel pins for torque transmission and centering ensure a clearly defined connection. Balancing as a complete assembled unit and correlative marking enable removal and remounting of this intermediate shaft with the connected rotors remaining in place, without affecting the balancing quality and vibration behavior of the complete string. (Source: Sulzer-Burckhardt.)

the unbalanced axial thrust of the high-speed rotor train to the thrust bearing on the low-speed wheel shaft. Good gear meshing requires parallelity of gear and pinion shaft and automatically ensures parallelity of the contact surfaces of thrust collar and wheel rim. The slight tapering of the thrust collars is responsible for the formation of a wedge-type oil film creating a pressure zone spread out on an enlarged surface with a pressure distribution very similar to that of a standard oil-lubricated journal bearing.

The relative motion between the two contact surfaces of the thrust collar system is a combination of rolling and sliding and takes place near the pitch circle diameter, resulting in a very small relative velocity. The thrust transmission is therefore conducted with minimal mechanical losses. The considerably reduced losses of the single thrust bearing on the low-speed shaft as compared with the high losses of individual thrust bearings on the high-speed train lead to a substantial power saving. This low-speed bearing can also be more amply dimensioned to provide a much higher overload capacity.

For direct turbine-driven compressor trains, the thrust bearing is usually located in the turbine. Also in this case solid couplings with flexible intermediate shafts are much preferred.

This coupling arrangment avoids heavy overhung gear couplings that are usually responsible for not clearly defined lower critical speeds and for the phenomena of torque lock leading to additional loading of the axial thrust bearing. The resulting axial friction forces can become quite substantial if insufficient attention is given to the cleanliness of the lubricating oil.

Centrifugal compressors

Centrifugal compressors made by this information source are available for the compression of various gases within a flow range of about 0.6 to 60 m^3/s and pressures up to 700 bar. They are used in the most varied applications, mainly in

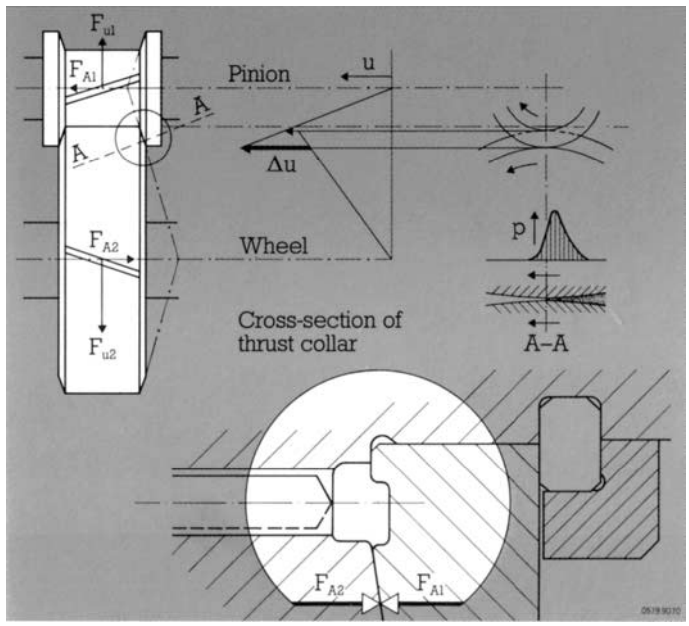


FIG. C-160 Method of axial thrust transfer in a single helical gear with thrust collar. (Source: Sulzer-Burckhardt.)

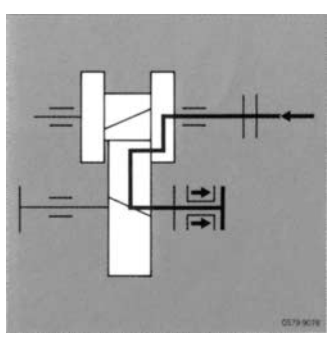


FIG. C-161 Transfer of external forces. (Source: Sulzer-Burckhardt.)

the fields of process engineering, chemical industry, and mineral ore processing, as well as oil and gas production. See Fig. C-127.

Units made by this information source have power inputs varying between a few hundred kilowatts and nearly 40 MW.

To minimize design lead time and ensure economical manufacturing and stocking of the major components, the range of these compressors has been standardized to an optimum extent. The constant geometrical ratio between the frame sizes allows compressors to be assembled to different specifications with the minimum inventory of component parts, diminishing the investment on spare parts. It also permits comparison and forecast of performance and stresses within the series of frame sizes. Figure C-162 shows a section through a centrifugal axially split compressor.

Type range. The standardized range of horizontally split casing types (see Fig. C-131) extends to eight frame sizes for this OEM source including three pressure ranges for each frame size up to 70 bar and is designed for appropriate turndown.

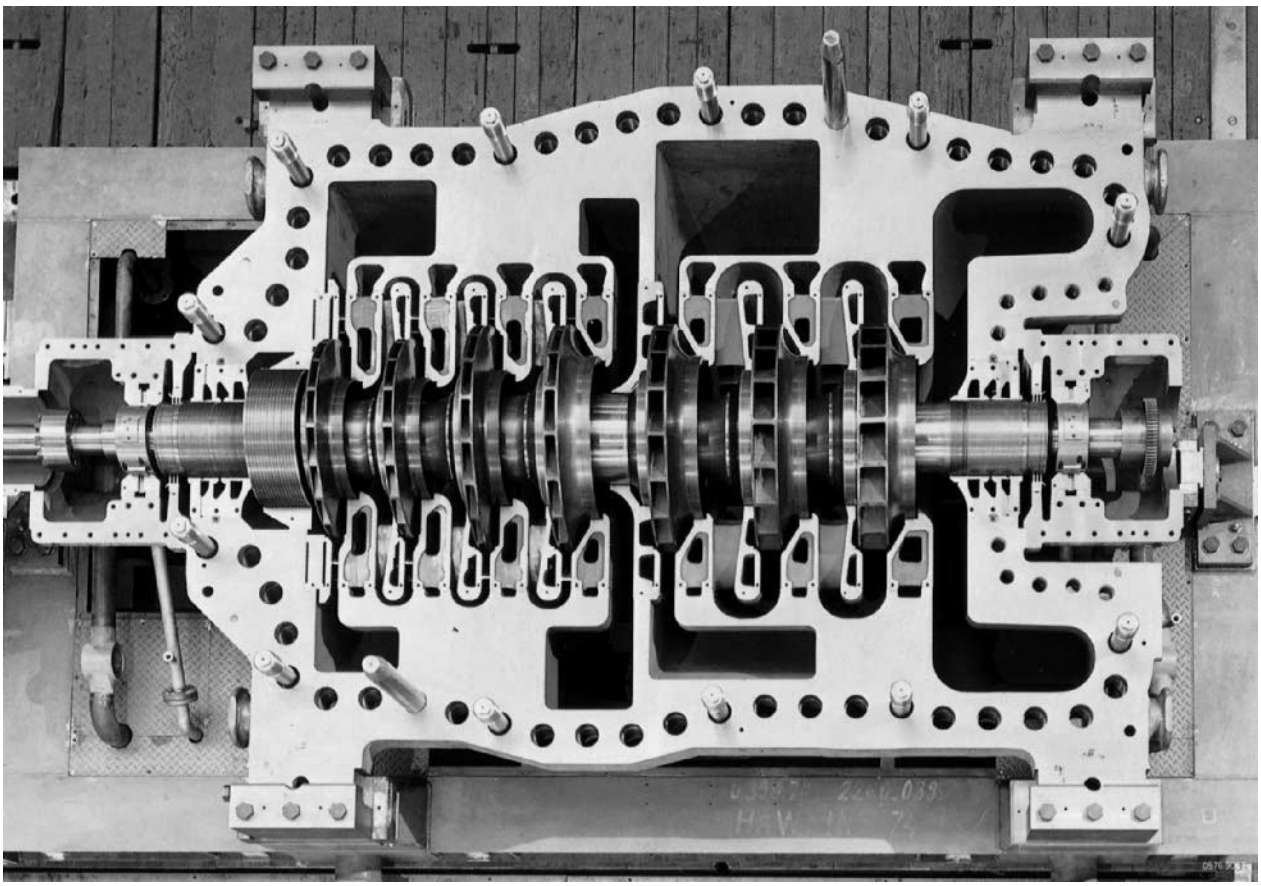


FIG. C-162 Section through a horizontally split axial compressor. (Source: Sulzer-Burckhardt.)

An appropriate selection of impellers and diffusers enables the compressor stages to be matched to any specified operating data. See Figs. C-163 through C-167.

The barrel-type design is suitable for discharge pressures of up to 700 bar. It has vertically split casing with end cover and autoclave cover and horizontally split internal casing for easy assembly and dismantling. Option of intermediate nozzles for connecting to intercoolers or for side-stream intake or extraction. The internal components, including the horizontally split internal casing, autoclave cover, diaphragms, and rotor with bearings and seals, are assembled outside the barrel casing; the internal clearances can then be checked exactly prior to final assembly.

Compressor design: adaptability of standardized product range. The different ways of adapting standard designs to varied operating conditions are described as follows:

Adapting to different specific flow conditions. The basic dimensions of the components and parts of compressors such as casings, impellers, diffusers, and bearings are standardized by using a constant scale factor between the different frame sizes. As can be seen in Fig. C-165, this factor is 1.25 for the casing sizes, 1.12 for impeller diameters, and even smaller for certain dimensions of internal parts. This system permits the assembly of various compressors complying to specified data by using a minimum number of predesigned parts. Moreover, it ensures accurate forecasting of performance interpolated from other frame sizes. The eight frame sizes cover a range between 0.5 and 60 m^3/s. The range of higher

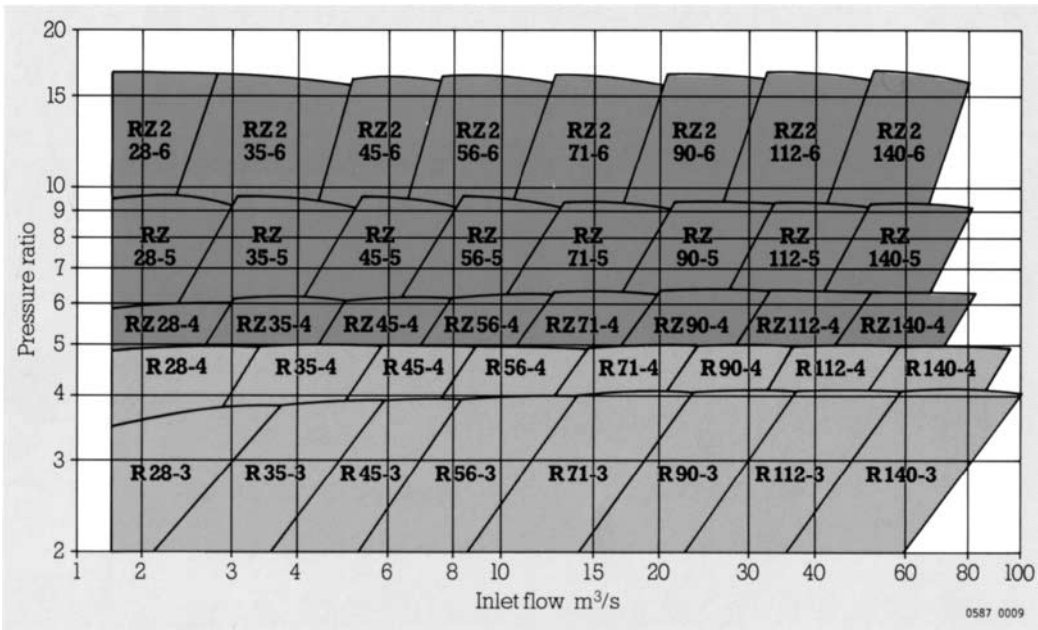


FIG. C-163 Selection chart for centrifugal compressors of the R and RZ series. (Source: Sulzer-Burckhardt.)

suction volumes above about 25 m^3/s and up to 350 m^3/s is covered by 12 frame sizes of axial compressors, where a scale factor of 1.12 is used.

Adapting to different specified pressures. This information source's models (R, RZ, and RS) can be equipped with casings made of gray cast iron, nodular cast iron, or cast steel which, depending on the frame size, makes them suitable for standard design pressure classes of 6, 16, 25, 40, or 64 bar. The RB, RBZ, and RBS casings are invariably made of fabricated or cast steel to cover the standard pressure range of up to 700 bar.

Adapting to different process conditions. The process industry has an ever-changing range of requirements concerning the arrangement of external casing nozzles, either when intercooling is needed to limit temperature during compression or intermediate inlets have to be provided for side-streams. The standard design is very flexible in this respect.

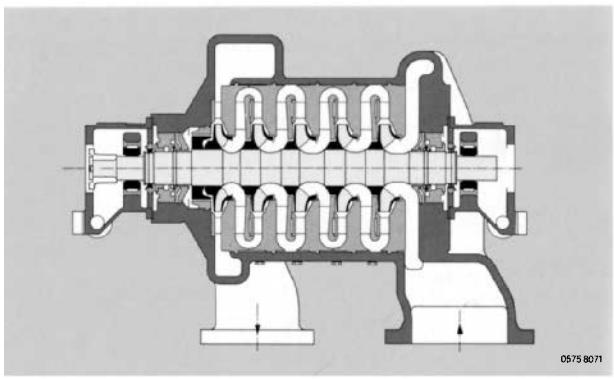
CENTRIFUGALS WITH HORIZONTALLY SPLIT CASINGS

- Series R: straight-through compressor without provision for intercooling
- Series RZ, RZ2: compressors with one or two pairs of intermediate nozzles for connecting to one or two intercoolers
- Series RS, RS2: compressors with one or two intermediate inlet nozzles for one or two side-streams

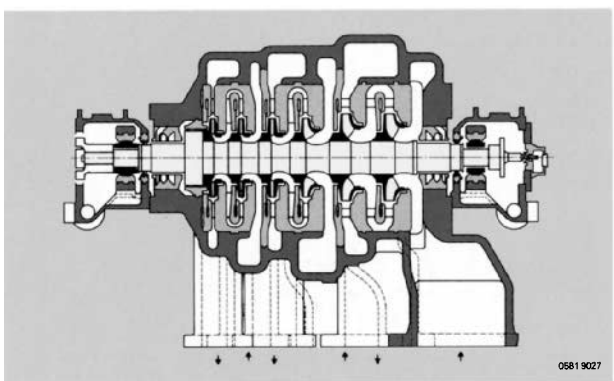
CENTRIFUGALS WITH VERTICALLY SPLIT CASINGS

- Series RB: straight-through compressor without provision for intercooling
- Series RBZ: compressor with one pair of intermediate nozzles for connection to an intercooler
- Series RBS: compressor RB with additional inlet or outlet

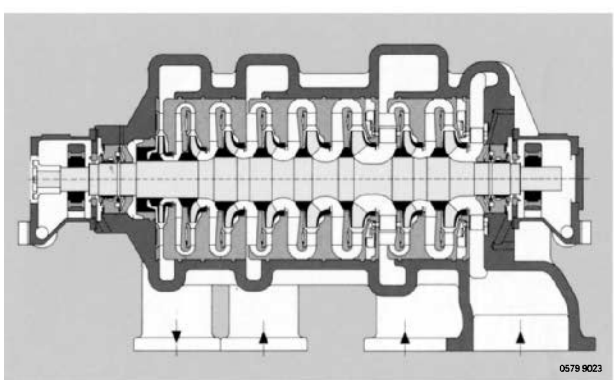
Series R, RZ, RS



Series R:
machines of one to eight stages with horizontally split casing

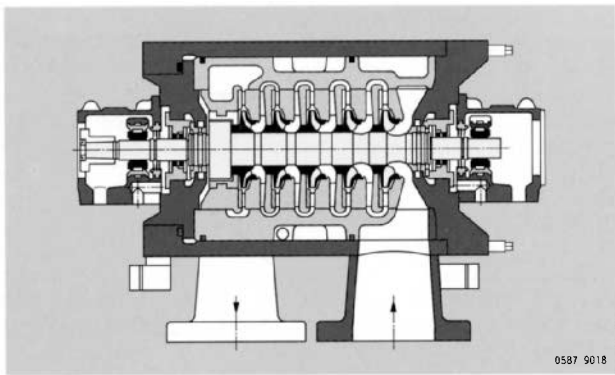


Series RZ:
machines of four to eight stages with horizontally split casing, one to two pairs of intermediate nozzles generally for connecting external intercoolers

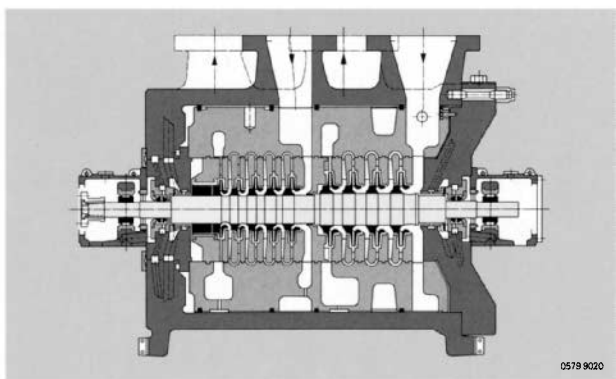


Series RS:
machines of four to eight stages with horizontally split casing and one or two nozzles for side stream intake or extraction

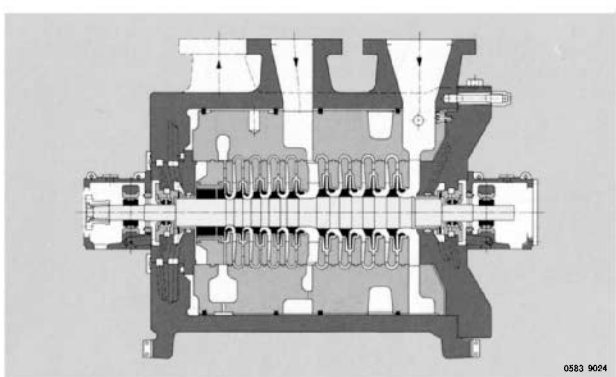
Series RB, RBZ, RBS



Series RB:
machines of two to eight stages with vertically split casing (barrel type)



Series RBZ:
machines of four to nine stages with vertically split casing and one pair of intermediate nozzles



Series RBS:
machines of four to nine stages with vertically split casing and additional inlet or outlet nozzles

FIG. C-164 Typical series designations for centrifugal compressors. (Source: Sulzer-Burckhardt.)

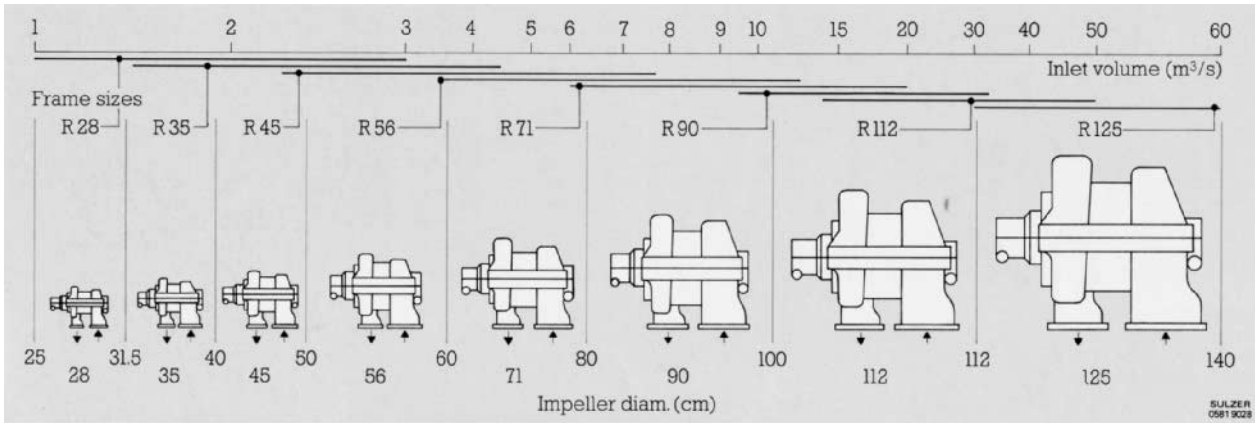


FIG. C-165 Standardized range of centrifugal compressors based on eight geometrically similar frame sizes. Each frame is designed to accommodate three different impeller diameters and different impeller types. (Source: Sulzer-Burckhardt.)

	Compressor types															
	R, RZ, RS								RB, RBZ, RBS							
Frame sizes	28	35	45	56	71	90	112	125	28	31	35	40	45	56	71	90
Pressure classes																
6					▭	▭	▭	▭								
16	▭	▭	▭	▭	▭	▭	▭	▭								
25					▭	▭	▭	▭								▭
40	▭	▭	▭	▭					▭	▭	▭	▭	▭	▭	▭	▭
64	▭	▭			▭				▭	▭	▭	▭	▭	▭	▭	▭
100									▭	▭	▭	▭	▭	▭	▭	▭
160									▭	▭	▭	▭	▭	▭		
250									▭	▭	▭	▭	▭	▭		
320									▭	▭	▭	▭	▭	▭		
400									▭	▭	▭	▭	▭			
500									▭	▭	▭					
700									▭	▭	▭					

FIG. C-166 Standard pressure classes. (Source: Sulzer-Burckhardt.)

Building block system. Each compressor frame size exists in different standardized lengths to accommodate different numbers of stages. With cast casings, adjustments of the pattern are made using a modular technique whereby spacing rings are fitted between standardized pattern parts of the casing to alter the length (Fig. C-167).

When the head required is more than that practicable within one compressor body, two or more casings can be connected in series to form a train.

Design features

Casing. Depending on the required pressure class and the type of gas, the horizontally split casings of the series R, RZ, and RS are made of gray cast iron, nodular cast iron, or cast steel, unalloyed or alloyed. See Fig. C-168 for a section through a horizontally split centrifugal compressor and Fig. C-169 for a section through a barrel centrifugal compressor. All suction, intermediate, and discharge nozzles are normally facing downward to facilitate inspection without disturbing the process pipe connections. Optionally they can be arranged facing upward. In the case of flammable or toxic gases, the horizontal division flange can be provided

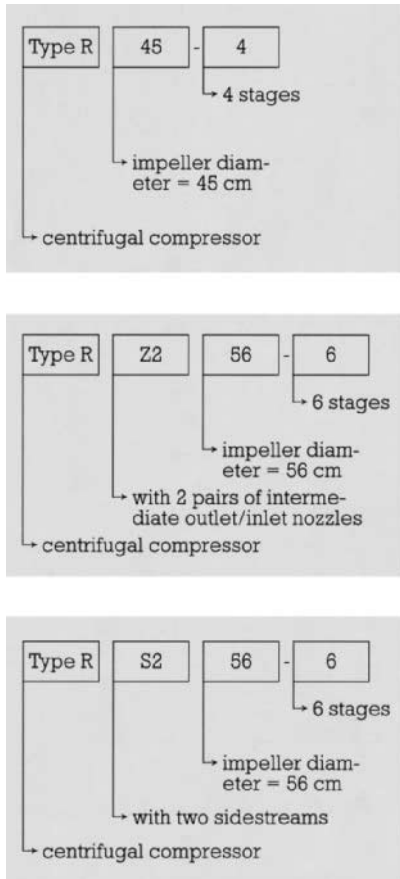


FIG. C-167A This is an illustration of the building block technique for adapting standard casing parts of a given frame size to various process conditions. For the casings of the three series R, RZ, and RS the same basic patterns are used. All casings are made available with nozzles facing upward. (Source: Sulzer-Burckhardt.)

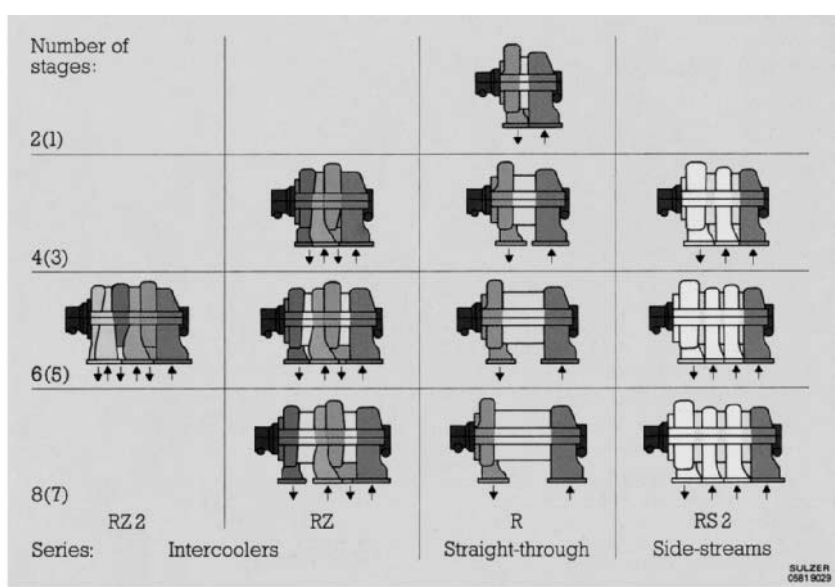


FIG. C-167B For barrel-type compressors the same designations apply as in Fig. 167A although "R" is replaced by "RB." (Source: Sulzer-Burckhardt.)

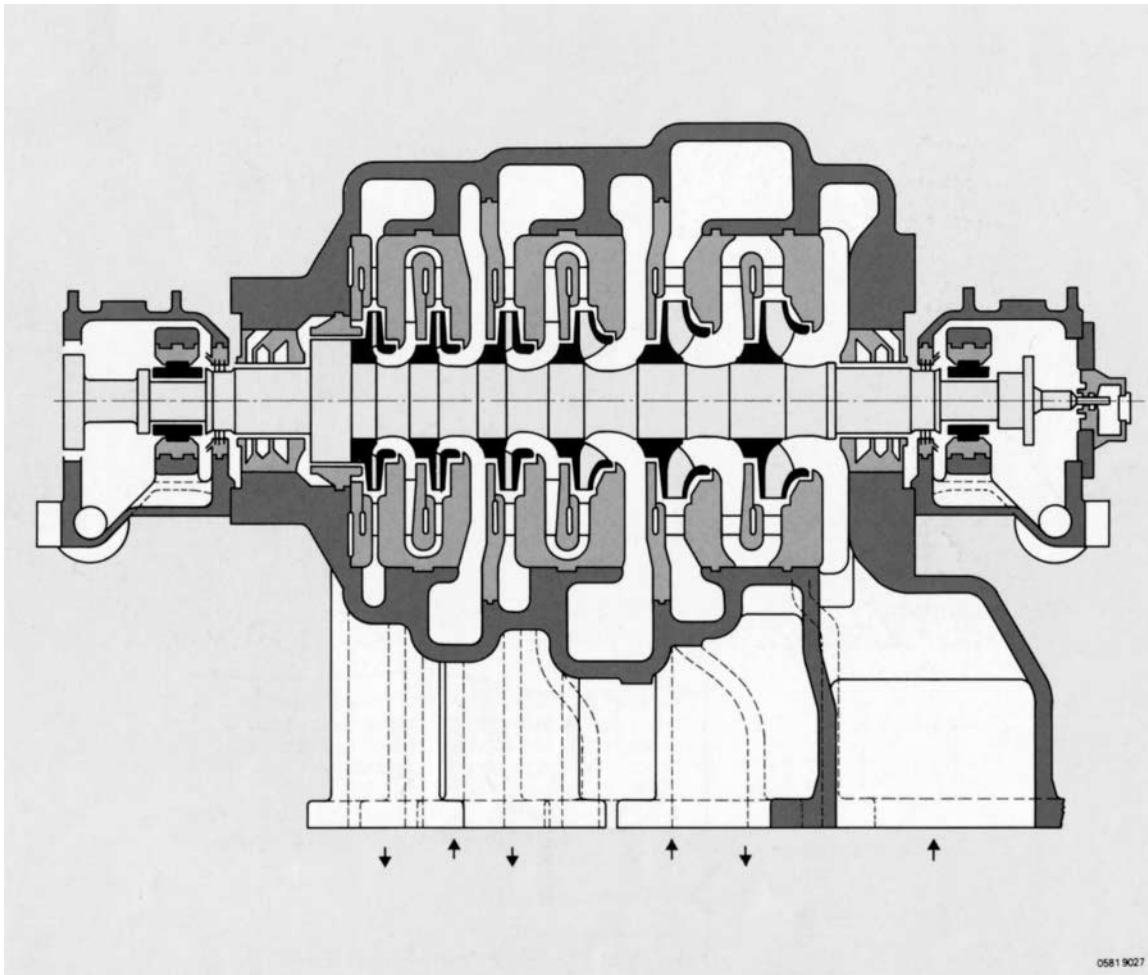


FIG. C-168 Cross-section of horizontally split compressor. (Source: Sulzer-Burckhardt.)

with drainage grooves to allow controlled leakage or inert gas sealing. Vertically split barrel casings of the series RB, RBZ, and RBS are made of cast steel or fabricated steel with the nozzles welded on. The barrel is closed at one end by an internal autoclave-type cover tightly locked by the inner gas pressure toward the shoulder of the outer casing. At the other end the cover is bolted to the casing and sealed by O-rings.

The diaphragms of both the R and RB series are normally made of gray or nodular cast iron, the diffusers of steel with the vanes welded on. The casing is supported by four feet at the horizontal flange to avoid misalignment due to thermal expansion.

The internal parts of the barrel compressor consist of the same standard components as are used for the R series. The horizontally split inner casing with rotor, diaphragms, diffusers, autoclave, end cover, and bolted-on bearing housings is preassembled outside the pressure casing. This subassembly is then inserted into the barrel by means of guide rails. See Figs. C-170 and C-171.

See Fig. C-172 for diagrammatic representation of impellers on rotor assembly, as well as illustration of specific impeller design features. The numbers in the figure correspond with the design features listed in the table on p. C-175.

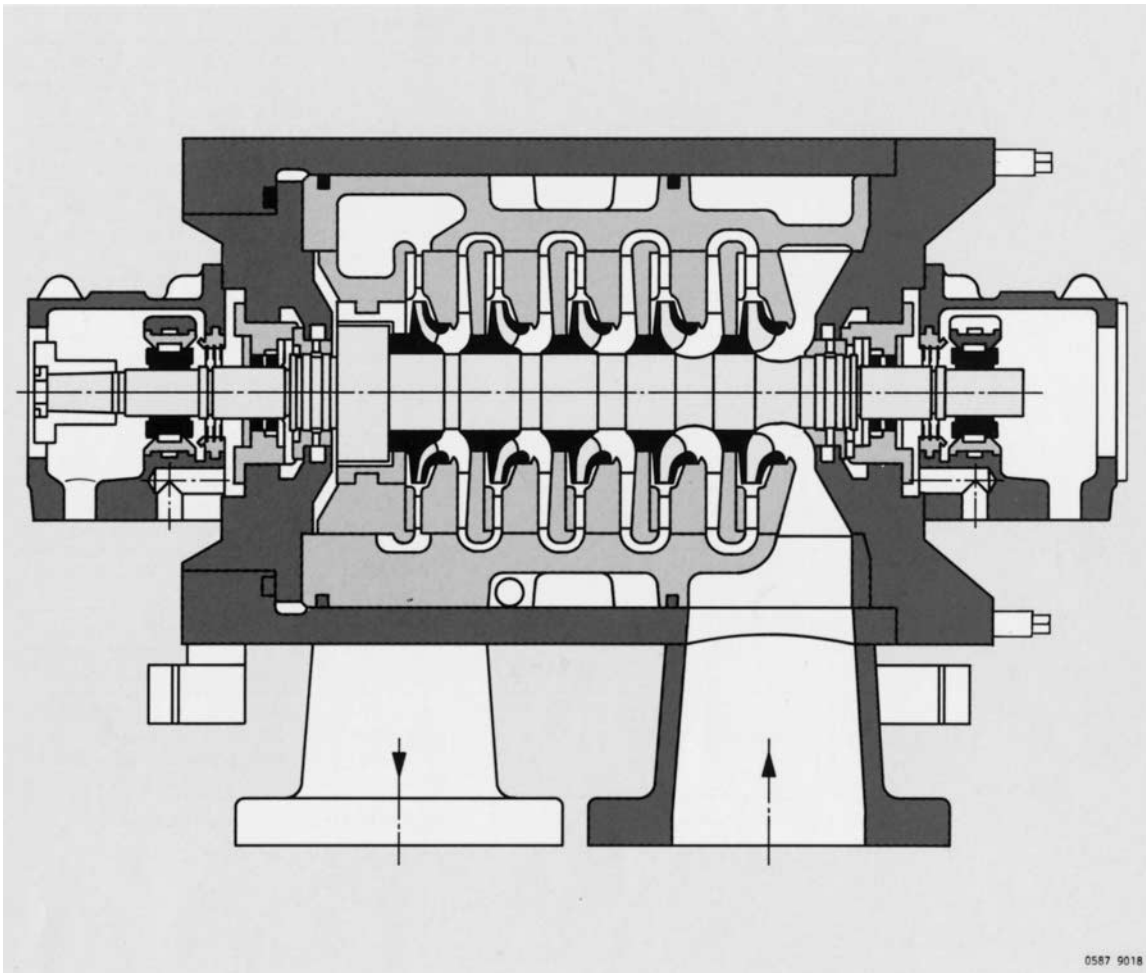


FIG. C-169 Cross-section of barrel compressor. (Source: Sulzer-Burckhardt.)

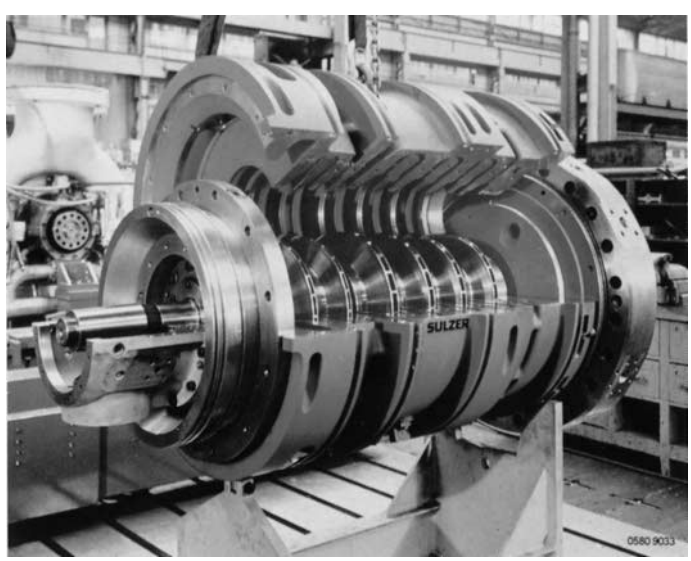


FIG. C-170 Inner subassembly including end covers, seals, and bearing housings. (Source: Sulzer-Burckhardt.)

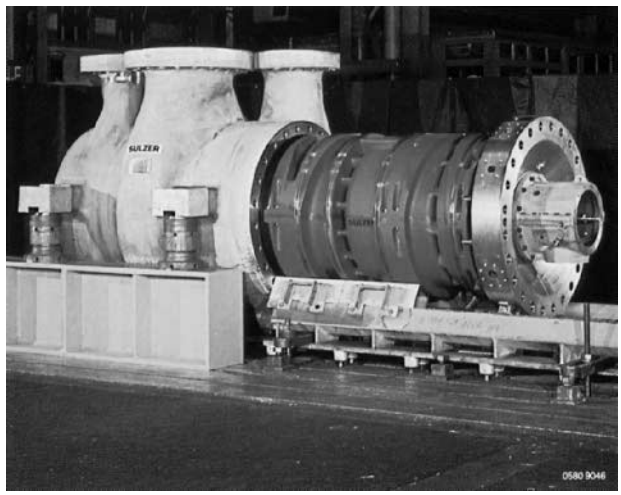


FIG. C-171 The inner subassembly is inserted into the barrel casing by means of guide rails. (Source: Sulzer-Burckhardt.)

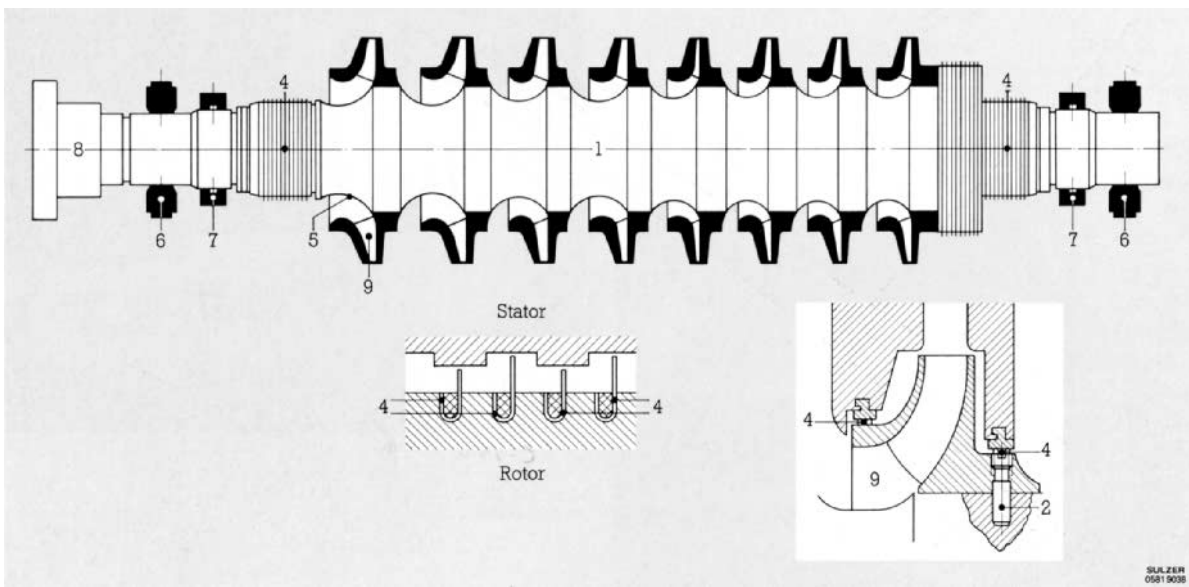


FIG. C-172 Design principles of turbocompressor rotors. (Source: Sulzer-Burckhardt.)

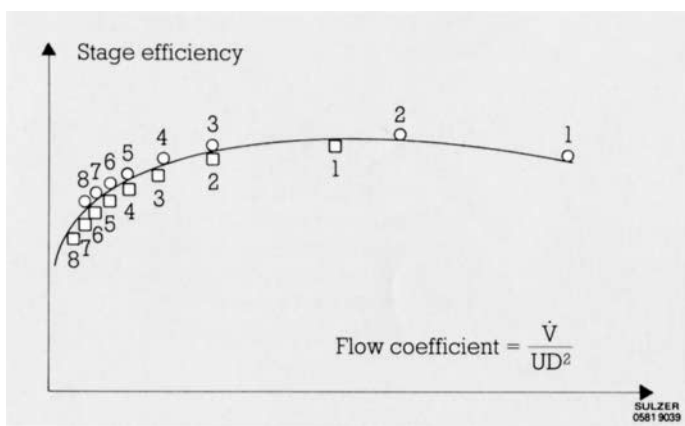


FIG. C-173 Influence of flow coefficient of the first impeller on the efficiency of subsequent stages. ○, efficiency distribution of a compressor with a high-flow impeller at suction. □, efficiency distribution of a compressor with a medium-flow impeller at suction. (Source: Sulzer-Burckhardt.)

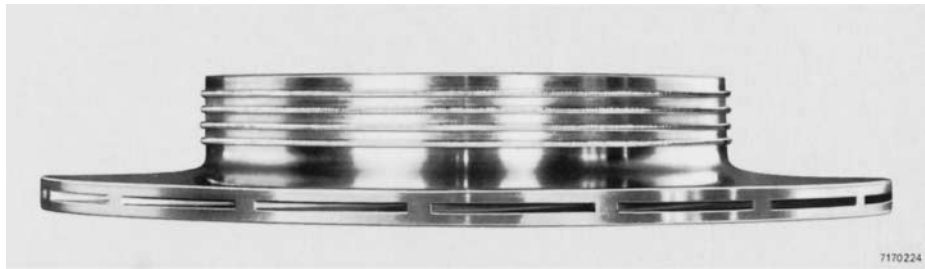


FIG. C-174 Very narrow impeller. Blades milled out of the hub disc and brazed to cover disc. (Source: Sulzer-Burckhardt.)

Design Features	Aimed at Providing
1. Solid sturdy rotor; integral dummy piston for high-speed and high-pressure applications	Minimum sensitivity to critical speeds and unbalance due to higher rotor stability; reduction of rotor internal damping
2. Shrink fit secured by symmetrically arranged radial dowels for impellers	No need for keys and distance bushings; fixation ensures concentricity and perfect balance under extreme operating conditions; allows larger shaft diameters; reduces stress on shaft and impeller
3. No shaft sleeves between stages	Reduces rotor hysteresis and increases running stability
4. Labyrinths always on the rotating element	No distortion of rotor due to local heating up in case of rubbing; labyrinths can be refitted easily
5. Nickel or other plating of shaft portions exposed to corrosion or entirely stainless shafts with mild steel welding or plating at bearings and at floating ring seals	Plating instead of shaft sleeves is a more direct protection; allows larger shaft diameters
6. Tilting pad radial bearings for higher speeds and pressures	Improves running stability; no oil-whip; higher external damping
7. No shaft sleeves for liquid-film seals	Minimum wear; perfect concentricity of shaft and rings with resultant minimum clearances and seal medium losses
8. Solid coupling, tightly bolted to flexible intermediate shaft	Improves reliability due to elimination of high-speed thrust bearing and toothed-type couplings; no gear lock thrust on high-speed thrust bearing
9. High-flow impeller at suction	Improves overall efficiency

Impellers. Impellers with high flow coefficients allow smaller diameters and optimum performance in all stages, resulting in high overall efficiencies (see Fig. C-173). These impellers are of fully welded construction with the blades shaped in three dimensions. A continuous welding technique ensures good aerodynamics and uniform stress distribution.



FIG. C-175 Wide impeller. Blades welded to both hub disc and cover disc. (Source: Sulzer-Burckhardt.)

Very narrow impellers consist of a hub disc with integral blades milled out of the disc material and a hub disc brazed to the blades according to a special manufacturing procedure (see Fig. C-174). For wider impellers, still not allowing welding on hub and cover disc, the blades are welded only to the cover disc (see Fig. C-175), and thereafter the hub disc is brazed to the blades (see Fig. C-176). Both alternatives provide accurate flow passages and a very smooth surface. In those rare cases where brazing is considered incompatible with the nature of the gas, such narrow impellers can be riveted.

Solid coupling. It is normal practice for some manufacturers, such as this information source, to make extensive use of solid couplings allowing the use of only one axial thrust bearing for single- or multiple-casing arrangements.

An intermediate shaft, flexible enough to allow for considerable misalignment, is inserted between the two shaft ends of the machines to be coupled together (Fig. C-177). In case of motor-driven units, the normal technique is to use single helical gears provided with thrust collars on the pinion shaft, as shown in Figs. C-178 and C-179. The thrust collars not only neutralize the axial thrust created by the meshing of the teeth cut at an angle to the axis of the shaft, but also transmit the unbalanced axial thrust of the high-speed rotor train to the thrust bearing on the low-speed wheel shaft.

Good gear meshing requires parallelity of gear and pinion shaft and automatically ensures parallelity of the contact surfaces of thrust collar and wheel rim. The slight tapering of the thrust collars is responsible for the formation of a wedge-type oil film creating a pressure zone spread out on an enlarged surface with a pressure distribution very similar to that of a standard-oil-lubricated journal bearing.

The relative motion between the two contact surfaces of the thrust collar system is a combination of rolling and sliding and takes place near the pitch circle diameter,

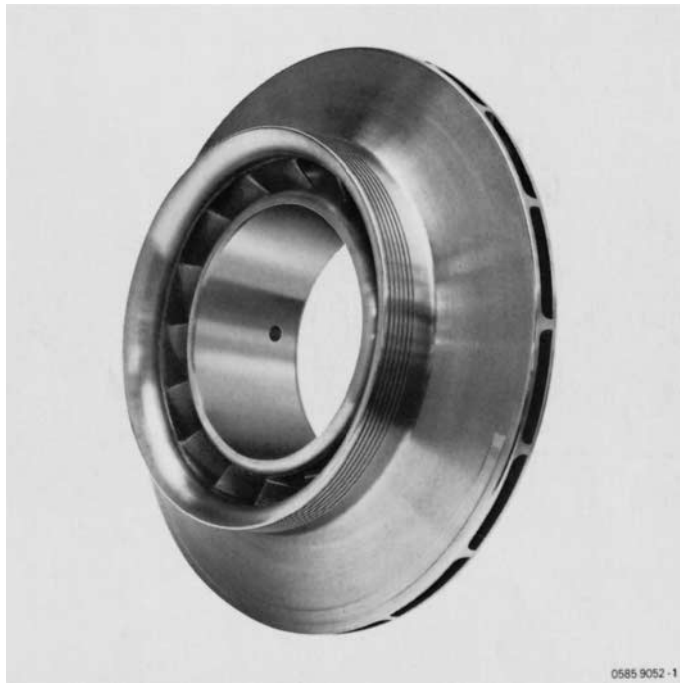


FIG. C-176 Narrow impeller. Blades welded to cover disc and brazed to hub disc. (Source: Sulzer-Burckhardt.)

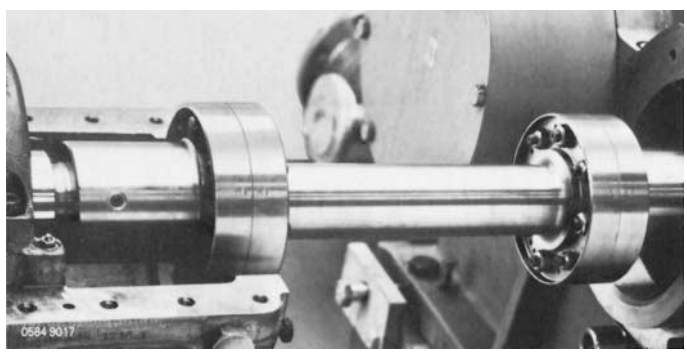


FIG. C-177 The solid quill-shaft coupling conforms to API 671 standard and consists of the quill shaft and the two hubs hydraulically fitted onto the shaft ends of the connected machines. On each coupling side, an equal number of tie bolts for axial fixation and tapered dowel pins for torque transmission and centering ensure a clearly defined connection. Balancing as a complete assembled unit and correlative marking enable removal and remounting of this intermediate shaft with the connected rotors remaining in place, without affecting the balancing quality and vibration behavior of the complete string. (Source: Sulzer-Burckhardt.)

resulting in a small relative velocity. The thrust transmission is therefore effected with almost no mechanical losses. The considerably reduced losses of the single thrust bearing on the low-speed shaft as compared with the high losses of individual thrust bearings on the high-speed train lead to a substantial power saving. Moreover, this low-speed bearing can be more amply dimensioned to provide a much higher overload capacity.

For direct turbine-driven compressor trains, the thrust bearing is usually located in the turbine. Also in this case solid couplings with flexible intermediate shafts are much preferred.

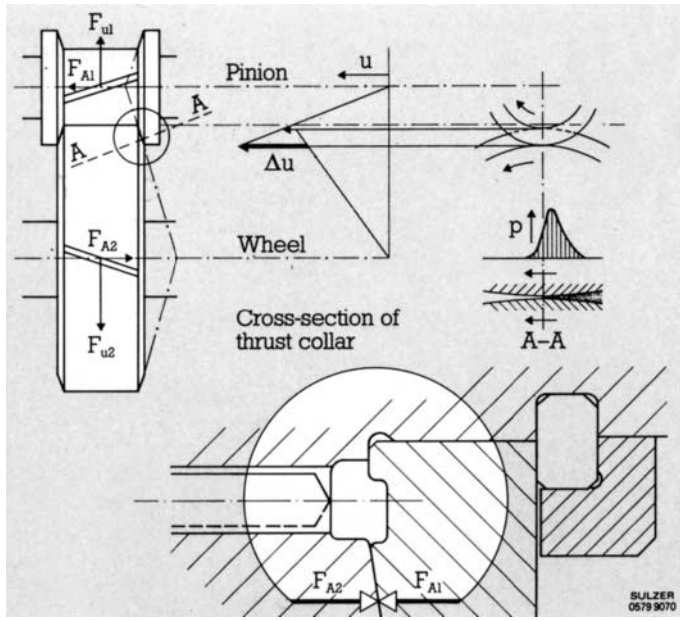


FIG. C-178 Method of axial thrust transfer in a single helical gear with thrust collar. F_u = peripheral force, F_A = axial force, u = peripheral speed, p = pressure. (Source: Sulzer-Burckhardt.)

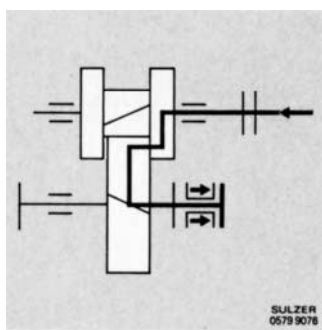


FIG. C-179 Transfer of external forces in bearing. (Source: Sulzer-Burckhardt.)

This coupling arrangement avoids heavy overhung gear couplings that are usually responsible for not clearly defined lower critical speeds and for the phenomena of torque lock leading to additional loading of the axial thrust bearing. As with axial compressors, the resulting axial friction forces can become quite substantial if insufficient attention is given to the cleanliness of the lubricating oil.

Journal and axial bearings

Journal bearings. See Fig. C-180. In the normal version, i.e., with the compressor rotor solidly coupled and the rotor thrust transferred to the axial thrust bearing of the prime mover or the gear, the bearing housings are equipped only with journal bearings. Two-lobe bearings are provided for speeds up to about 7000 rpm, tilting pad journal bearings are generally used for higher speeds and for types RB, RBZ, and RBS for reasons of stability. The slight curvature of the adjusting plates allows the bearings to be set accurately on installation. The bearings are firmly held in position by the bearing housing top half.



FIG. C-180 Multisegment journal bearing with four tilting pads. (Source: Sulzer-Burckhardt.)

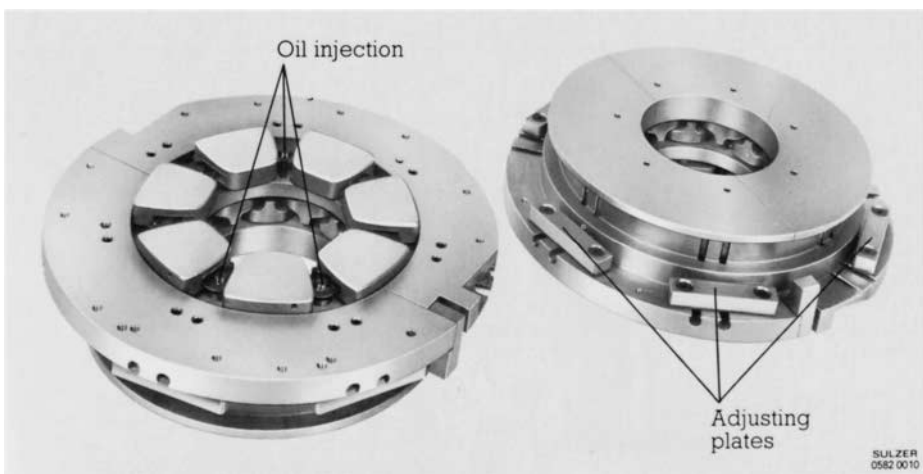


FIG. C-181 Kingsbury-type thrust bearing with self-equalizing pads with directed lubrication. (Source: Sulzer-Burckhardt.)

Two-lobe bearings are suitable for both senses of rotation, while tilting pad bearings are essentially for only one direction, although they can tolerate running backward with a somewhat reduced load capacity.

Axial thrust bearings. See Fig. C-181. Compressors driven through toothed or flexible couplings have a special bearing housing that can accommodate the necessary additional tilting pad thrust bearing. The purpose of this bearing is to absorb the remaining thrust of the machine and any significant axial friction thrust of the coupling due to sharp temporary differential expansion between rotor and casing. To provide easy access and reduce the overhang, it is preferable to mount it on the free shaft end.

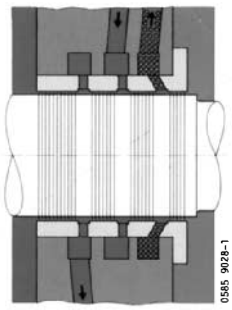
The tilting pads are supported on load-equalizing segments that allow angularity of the shaft up to 0.3 percent.

Because the tilting pads are supported eccentrically, thrust bearings are suitable for only one direction but tolerate a reversed rotation at a somewhat reduced load capacity.

Labyrinth seal

For low pressures or for applications, where either a constant loss of the process gas to atmosphere or a constant slight contamination of the process gas with buffer gas can be tolerated, or where even at high pressures the oilseal must be excluded for reasons of incompatibility of oil with the process gas, the appropriate seal is the labyrinth type. The simplicity of its design and associated system make it a reliable element with easy maintenance.

By interchanging the connection for buffer gas and ejector, it either prevents leakage to atmosphere (picture) or contamination of process gas.



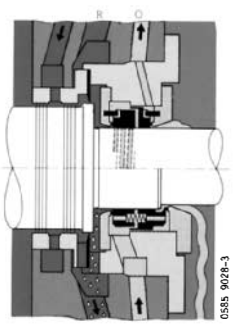
The labyrinths consist of thin stainless J-shaped steel strips on the rotating part and are kept in place by a caulked in steel wire. All strips are easily replaceable.

The labyrinth strips are caulked into the rotating part because, if rubbing occurs, the thin metal is then abraded without the casing and the rotor becoming unacceptably warm, avoiding rotor distortion.

Liquid film seal

Gas-tight shaft seals of the non-contact type are SULZER's basic recommendation for compressors delivering inflammable, toxic or costly gases.
Stationary floating rings operate satisfactorily at peripheral speeds as high as 100 m/s and are therefore preferably fitted on high-speed machines or compressors with large shaft diameters.

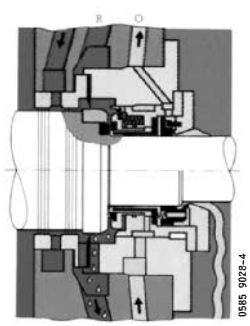
Liquid film seals consist of a pair of seal rings fitted in such a way that they are floating to follow radial movements of the shaft relative to the casing.
One ring of a pair is on the atmospheric side, and the other on the gas side. The latter is working with a relatively small pressure difference of 0.5 to 1.0 bar between the seal oil pressure and the gas pressure, in order to keep the oil leakage towards the gas side small.
The leakage of sealing oil towards the gas side is acceptable in most cases, since the oil can easily be recovered after degasing. If required this oil leakage can be further reduced by providing the inner side of the gas side ring with spiral wind-back grooves. This special feature combines the advantages of floating ring seals (no wear, long life) with those of mechanical seals (minimum oil leakage to the gas side).



Mechanical seal

Where extremely low seal oil flow towards the gas side is required, a single or double mechanical seal may be installed. These seals can be used without reservation up to 90 m/s sliding velocity. The mechanical seal can also serve as stand-still seal.

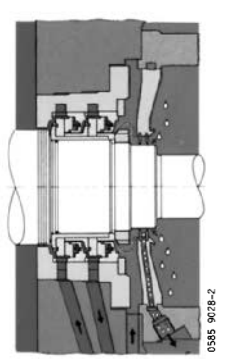
Although mechanical seals should primarily be used for applications where the ring can be periodically checked for wear, they have also been installed for services where minimum intervals between overhauls are well in excess of 10 000 operating hours.



The mechanical seal consists of a stationary carbon face mounted in a sealing ring support. It is moderatly pressed against the face of a shaft collar by several springs as well as by the sealing oil pressure, which is about 1 to 3 bar higher than the gas pressure. Symmetrically placed double mechanical seals are available. At standstill of the compressor and the seal oil pump, the negligible quantity of sealing oil required is supplied from the overhead tank to which the reference gas pressure is connected.

Gastight dry gas seal

For flammable, toxic and pure gases up to temperatures of 200 °C. One single seal is capable of sealing a pressure differential of 100 bar. For higher levels the number of seals is increased accordingly. Process gas tapped from the compressor discharge is fed to the seal through a fine (2 µ) filter. The small leakage is led to the stack. A fan ensures that an ample air flow dilutes at all times any accumulation of gas/oil vapor behind the seal to a safe level, well below an explosive mixture. An additional seal serving as standby can be installed (tandem arrangement). Installation of a labyrinth seal with inert gas injection between two dry gas seals is possible.



R Reference pressure (connected to the seal at the compressor discharge side)
O Sealant overflow for cooling of seal

Atmospheric or air side
Process gas, also sealing or buffer gas
Sealant liquid, seal oil
Sealing gas and sealant liquid mixture: seal gas to flare or back to suction via separator and/or active carbon filter; contaminated sealant to separator or collector tank to waste or to oil reclamation unit
Gas/oil vapor mixture diluted with air

FIG. C-182 Shaft seal arrangements. (Source: Sulzer-Burckhardt.)



FIG. C-183 Inlet guide vane assembly. (Source: Sulzer-Burckhardt.)

Emergency axial thrust ring. Compressors without their own thrust bearing can be fitted with an emergency axial thrust ring so that if the coupling between compressor and driver fails, the rotor remains in position as it slows down and does not rub on the casing. A relatively large clearance (approx. 1.0 mm) is provided so that the thrust ring does not rub during normal operation.

Oxygen compressors are always provided with an axial thrust ring.

Shaft seals. See Fig. C-182.

Inlet guide vanes. Figure C-183 illustrates an inlet guide vane assembly. On centrifugal compressors running at constant speed and mainly full load, a suction throttle valve is the most appropriate way to reduce the starting torque and for part-load operation. In cases where part-load occurs frequently and power is highly evaluated, inlet guide vanes achieve higher part-load efficiencies and a somewhat larger stable operating range. Inlet guide vanes before the first stage can be accommodated as a standard option of a horizontally split or the recycle stage of a vertically split compressor. They are located in the inlet channel and are adjusted through linkages by a ring that in turn can be operated by a hand wheel or connected to a pneumatic or electric actuator allowing for automatic volume or pressure control or alternatively remote setting. The vanes are provided with shafts pivoting in self-lubricated bushings. These are completely maintenance-free and due to the absence of lubricant, there is no contamination of the process gas. See Fig. C-184.

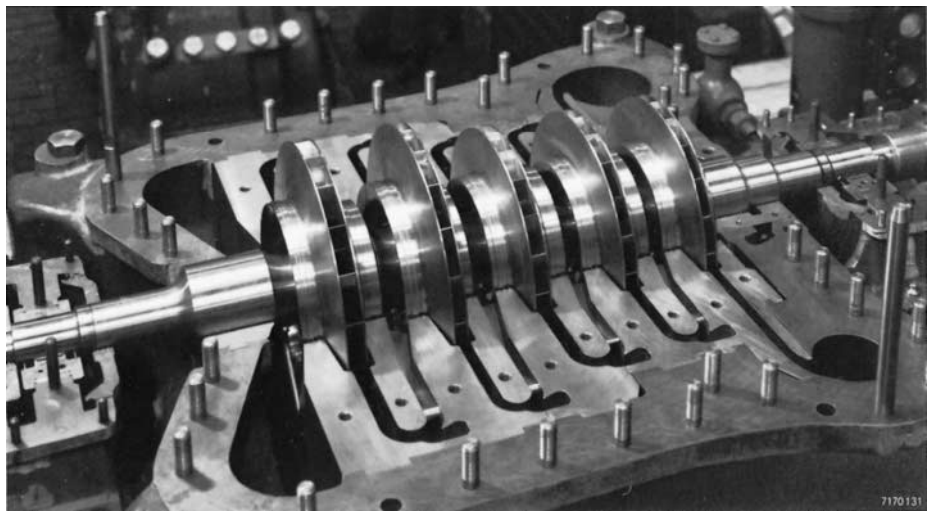


FIG. C-184 Centrifugal compressor with injection and drainage system for the compression of dirty gases. (Source: Sulzer-Burckhardt.)

Fluid injection devices. When compressing dirty gases or fluids that can cause crystal formation or polymerization, it is possible that some of these impurities might settle on the inside of the compressor channels and clog the internal passages. Injection devices have been developed for cleaning the insides of compressors, either periodically or continuously, so as to maintain the original performance.

When necessary, injection nozzles are located in the flow channels and the washing fluid is injected as close as possible to the deposits. In centrifugal machines injection is effected in all stages. Nozzles may also be provided to wash the very narrow leakage paths at the rotor seals.

The amount of fluid injected is controlled with dosemeters.

To prevent corrosion of those parts that come in contact with the fluid, conditions are controlled to avoid high temperatures, high water content due to evaporation of the water, or saturation of the process gas.

By using specially adapted materials for the internal components, gases can also be compressed in fully wet condition. Means are provided in the compressor casing for the drainage of excess fluid and sludge.

Compressor materials. Typical metallurgical selections for compressor materials are listed in Fig. C-185.

Application example: Retrofit design modification case history: Mopico™ compressor for gas pipeline stations

Updating requirements of existing installations. There are approximately 6400 gas compressor units of all types in the United States, with a total rating of 10 million kW (13 million hp) installed in over 1050 stations on the U.S. gas transmission system. These mainline stations are spaced mainly 50–70 miles apart and include 2 to 30 units. Most of these stations were built more than 30 years ago.

Such gas pipeline right-of-ways often consist of three or four pipes, 24 to 42 inches in diameter and were originally rated for 60 to 70 bar maximum operating pressure, a value that has been reduced to 50 to 60 bar due to the age of the installations.

Element	Material designation	DIN No.	ASTM or AISI equivalent
Casings	GG-20	0.6020	A 278, Class 30
	GGG-40	0.7040	A 395
	GGG-Ni 22	0.7670	A 439, Type D-2C
	GS-C25	1.0619	A 216, Grade WCB
	GS-17 CrMo 5 5	1.7357	A 217, Grade WC6
	GS-17 CrMo 9 10	1.7377	A 757, Grade D1 Q1
	G-X 5 CrNi 13 4	1.4313	A 487, Class CA6NM
	G-X 5 CrNi 17 4	–	Cast 17-4
	25 CrMo 4	1.7218	A 336, Class F12
	StE 355	1.0562	A 537, Class 1
Diaphragms	HI	1.0345	A 515, Grade 55
	GG-20	0.6020	A 278, Class 30
	GGG-40	0.7040	A 536 or A 395
	GGG-Ni 22	0.7670	A 439, Type D-2C
	GS-C25	1.0619	A 216, Grade WCB
	G-X 5 CrNi 13 4	1.4313	A 487, Grade CA 6NM
	G-X 5 CrNi 17 4	–	Cast 17-4
	X 5 CrNi 18 9	1.4301	A 276, Type 304
	X 20 Cr 13	1.4021	A 276, Type 420
	X 20 CrNi 17 2	1.4057	A 276, Type 431
Shafts	28 NiCrMoV 8 5	1.6932	A 470
	20 CrMoNiV 4 7	1.6979	A 470
	25 CrMo 4	1.7218	AISI 4130
	34 CrNiMo 6	1.6582	AISI 4340
	X 20 CrMoV 12 1	1.4922	A 565, Grade 616 HT
	X 20 CrNi 17 2	1.4057	AISI 431
	NiCu30Al	2.4375	Monel K-500
Impellers	25 CrMo 4	1.7218	AISI 4130
	23 CrNiMo 7 4 7	1.6749	A 471
	20 CrMoNiV 4 7	1.6979	A 471, Class 10
	X 2 CrNi 13 4	–	A 182, Grade F3NM
	X 5 CrNi 13 4	1.4313	A 182, Grade F6NM
	X 6 CrNiMoCu 14 5	–	A 705, S45000 (14-5PH)
	X 20 CrMoV 12 1	1.4922	A 565, Grade 616 HT
	X 20 CrNi 17 2	1.4057	AISI 431
	X 2 CrNiMoN 22 5	1.4462	A 182, Grade F51
	NiCu30Al	2.4375	Monel K-500
	TiAl6V4	3.7165	B 381, Grade F-5
Labyrinths	X 6 CrMo 17 1	1.4113	AISI 434
	X 6 CrNiTi 18 10	1.4541	AISI 321
	NiCu30Fe	2.4360	Monel 400
	Titanium	3.7055	Titanium

FIG. C-185 Typical compressor material selection. (Source: Sulzer-Burckhardt.)

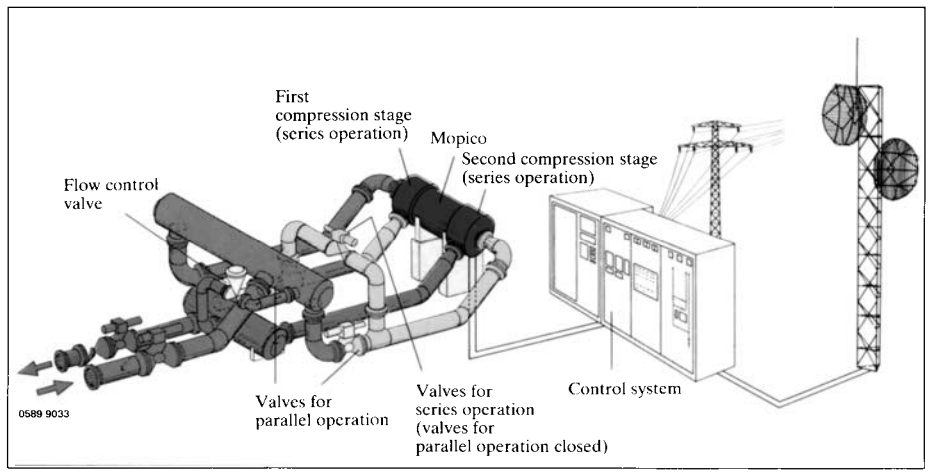


FIG. C-186 Schematic layout of a compressor station. (Source: Sulzer-Burckhardt.)

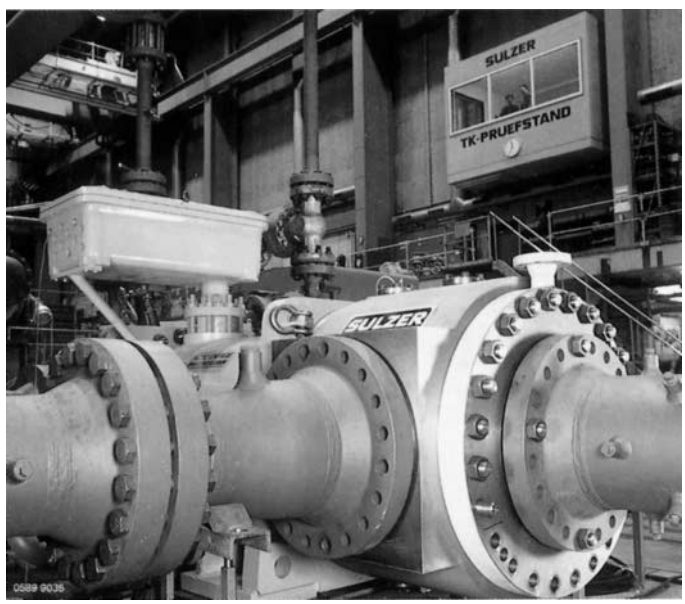


FIG. C-187 Mopico compressor on a test bed. (Source: Sulzer-Burckhardt.)

These antiquated systems are very limited in their delivery capacity compared to modern systems. As an example, a 55-mile pipeline section between stations, having one 30-, two 36-, and one 42-in conduits operating at a maximum line pressure of 55 bar can transport about 100 million Nm^3 gas per day. Some 15 to 20 individual integral gas engines, drive rating 33,000 to 37,000 kW are required for this purpose.

Such installations compare poorly with modern systems: a typical Russian gas pipeline built in the 1980s can transmit the same 100 million Nm^3 per day through 55 mi of a single 56-in conduit at 74.5 bar maximum line pressure using two gas turbines producing the same total output.

The Russian compressor stations are standardized in design and rating and are located approximately every 65 mi and include over 75,000 kW in gas turbine power. Because the Russian system was developed 30 years after the American, the gas

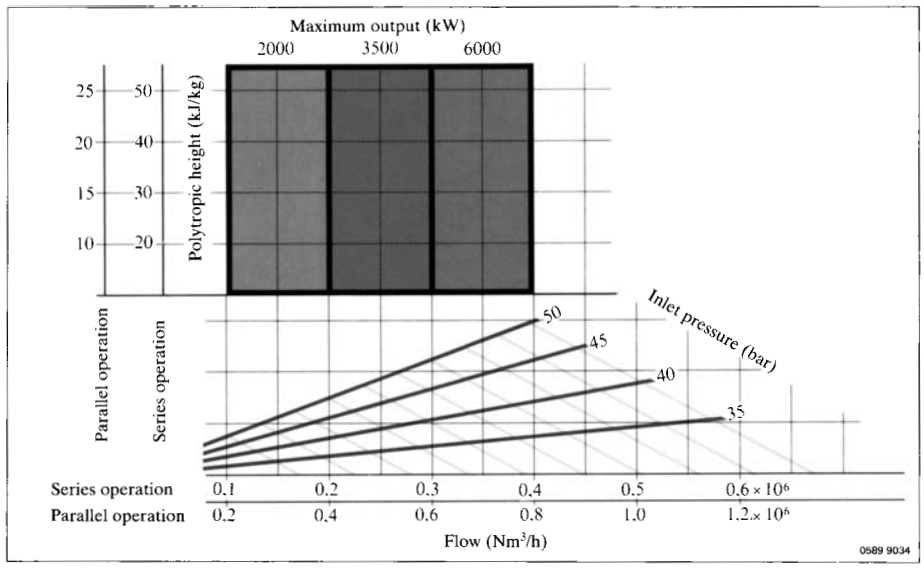


FIG. C-188 Operating ranges of the Mopico compressor. (Source: Sulzer-Burckhardt.)

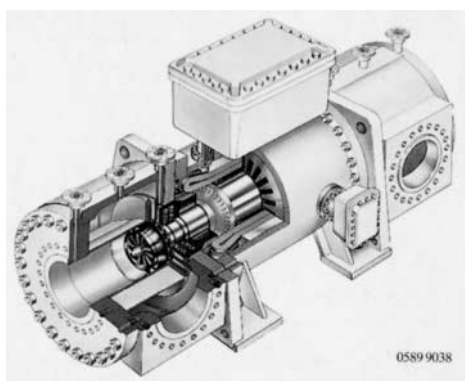


FIG. C-189 Cutaway section through the new Mopico gas pipeline compressor. Motor and compressor are fitted into a hermetically sealed casing. (Source: Sulzer-Burckhardt.)

turbine technology of the 1970s was available to them, as well as the large diameter, high pressure pipe produced in Japan and Europe.

The European pipeline network that was built around 1970 is on a par with other systems built in the same period, i.e., the Argentinian, Australian, and Canadian systems. However, during the past few years, the Canadian gas pipeline companies have carried out tests with new gas turbine units and combined cycle systems. Moreover, the first high-power motor compressors using variable-frequency drives (VFDs) have been installed. These 6-MW and 18-MW systems use German and Swedish synchronous motors and the load commutated inverter (LCI) drive technology.

In contrast, the gas pipeline networks in the United States have had virtually no important new pipeline technology applied to them. The new Mopico compressor, with its high-speed induction motor, magnetic bearings, and variable-frequency drive, is useful for the conditions in older installations.

See Fig. C-186 for a typical compressor station layout, Fig. C-187 shows one of these compressors on a test bed, and Fig. C-188 shows the operating ranges of the "Mopics."



FIG. C-190 Rotor shaft with radial impellers. (Source: Sulzer-Burckhardt.)

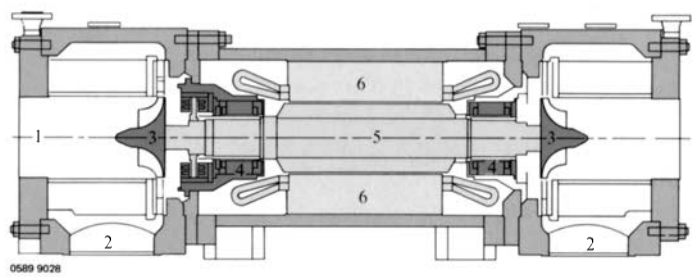


FIG. C-191 Section through the simply constructed Mopico compressor. (Source: Sulzer-Burckhardt.)

Design. The Mopico gas pipeline compressor (see Figs. C-189 and C-190) features a high-speed, two-pole, squirrel-cage induction motor. Motor and compressor are housed in a hermetically sealed, vertically split, forged steel casing (Figs. C-189 and C-191). The center section contains the motor and bearings, and each of the end casing sections houses a compressor wheel, a fixed vane diffusor, and the inlet and discharge flanges.

Mopico compressors can be operated in series or parallel. Magnetic radial bearings and a double-acting magnetic bearing maintain the runner in position. Figure C-190 shows the rotor shaft with radial impellers. The motor is cooled by gas metered from the high-pressure plenum of one of the compressor housings. Hence the Mopico runs completely oil-free.

The speed and thus the discharge rate of the Mopico unit is controlled by a thyristorized, variable-frequency drive. This drive uses thyristors that can be switched out. These enable pulse-free runup without current peaks and an operating speed range of 70 to about 105 percent.

Design criteria. The following conditions can be complied with through the new combination of elements:

- Low installation, maintenance, and energy consumption costs
- Broad operating range at high economic performance
- Compatibility with existing compressors
- Unattended remote control

- Emissionless and oil-free
- Can be installed outdoors

Based on cost per installed hp, the cost of the Mopico compressor is only about two-thirds that of a gas turbine unit and less than half that of a low-speed reciprocating compressor. It is some 10 percent less than a "conventional" centrifugal with dry seals, magnetic bearings, and a direct-drive, high-speed induction motor with variable-frequency drive.

On the other hand, a conventional centrifugal with motor-gear drive and IGV is less expensive. This system is, however, unacceptable for pipeline application because of poor efficiency at low pressure ratio conditions.

The overall energy consumption costs of a Mopico system using energy produced in a base load power station is considerably lower than the cost of either a gas engine or a gas turbine burning natural gas.

Gas pipelines impose the most stringent operating requirements. A typical main line station has to accommodate flow differentials of 50 percent and more between winter and summer. In order to handle the wide range of part-load conditions most efficiently, a main line station should include multiple individual units, each with a broad operating range at high efficiency. Both prime mover and compressor have to be taken into account when evaluating part-load efficiency.

Most of the competitive systems offer good design point efficiencies, but show rapid efficiency deterioration below 70 percent load. With Mopico, however, the motor-VFD system operates within 3–5 percent of its design efficiency over the full operating range of the pipeline. Motor speed never drops below 70 percent, since the operating mode of the system changes from series to parallel at that point. A mainline station with six Mopico units would typically operate six months of the year with three units in the parallel mode. For the rest of the year, the station would operate with four to six units in the series mode.

Compatibility. There is no problem associated in paralleling Mopico units with existing recips or centrifugal compressors. The speed of the Mopico is adjusted to attain the desired level of loading.

Unattended remote control. The system is designed to be operated by remote signals much the same as motor pumps on oil pipelines.

Low maintenance cost. A Mopico compressor has no wearing parts and thus requires practically no regular maintenance. Planned maintenance is limited to the replacement of the water cooling pump's mechanical seal every few years, the periodic cleaning of the filters for the control room air-conditioning system and the normal verification and adjustment work on the electronic control equipment of the drive system and of the magnetic bearings.

Oil-free, no emissions, intrinsically safe. The Mopico has no shaft seals, because it runs on magnetic bearings. Therefore there is no oil requirement in the system. Moreover, since there is no combustion, there are no emissions. Intrinsic safety of the system arises from the hermetic sealing and the fact that the motor is fully pressurized with cooling gas (no air).

Installation. The Mopico compressor can be installed out- or indoors. The VFD system, magnetic bearing controls, other unit controls and switchgears are designed for installation in a weather-protected building outside of the hazardous area.

Design features

Motor. The electric motor manufacturer selected an asynchronous motor for the Mopico system. The advantages of such high-speed, squirrel-cage motors are:

- Good efficiency
- Low maintenance requirements
- Ability to comply with the safety regulations for hazardous areas
- Axisymmetric rotor
- Simple construction

Additional specific design objectives for this particular application were:

- Ability to function in a gas-filled environment
- Compatibility with the variable-frequency drive
- Compatibility with the magnetic bearings

The main terminal box is located on the top of the motor housing. Two smaller terminal boxes on both sides are used for controlling and feeding the magnetic bearings. Both terminal box types meet the requirements of the various safety regulations for hazardous areas (CSA, NEC, EuroNorm).

Because the motor is cooled by gas, the stator windings are built to Class H standards, with mica and glass-based insulation materials that withstand temperatures to 200°C. The impregnation technique uses a silicone resin to ensure that the stator can completely withstand mechanical stresses and remain impervious to dampness and corrosive environments. Special shields in nonmagnetic materials are then adjusted inside the housing to avoid eddy-current losses in the steel.

Compressor. The compressor module consists of three main parts: the outer, pressure-bearing discharge casing, the inlet insert and the radial compressor impeller. The outer casing is machined from a single steel forging with an integral discharge flange. The symmetry of this and the motor casing allows the discharge flange to be arranged at any angle. There are no lining-up problems, since the solid casing makes the unit impervious to forces associated with the external pipeline. The insert is a simple welded structure, which combines the functions of axial inlet to the impeller, inlet labyrinth carrier, and diffusor carrier. Attachment of the radial impeller to the high-speed motor shaft end is by means of a central tie bolt; torque is transmitted by means of two drive pins.

The impellers are of the shrouded type and belong to the family of modern impellers designed in recent years using the latest numerical design and testing techniques. A wide operating range at high efficiency is a feature of such impellers.

The diffusor was developed especially for the Mopico unit. It has a rather short radial section containing fixed vanes in a tandem arrangement. At the exit to the vanes, the now purely radial flow is dumped into an annular space. In comparison to more conventional diffusors, the Mopico type exhibits a flat loss characteristic.

Drive. The frequency control equipment is located in a separate building, outside the area classified as hazardous. Also housed inside the building are the motor starter-breaker, the magnetic bearings control system (including an auxiliary power supply), the Mopico unit controls, including valve sequencing, auto start/stop, surge control, real-time performance monitoring, and the various system monitoring, alarm, and shutdown devices.

Testing of the Mopico system. Already at the beginning of the design phase of the motor, tests were carried out with a small test rotor having the same diameter as the actual one, but reduced in length. The purposes of these tests were to study the mechanical stresses in the rotor and to evaluate the windage losses. This rotor was mechanically driven up to 13,000 rpm, without any permanent deformation occurring.

TABLE C-13 Test Results, Design, and Maximum Values

	Test	Design	Maximum
Speed (rpm)	9,450	9,850	9,850
Shaft power (MW)	5.63	5.74	8.5
Line voltage (kV)	5.74	5.94	5.94
Current (A)	688	677	979
Power factor	0.87	0.87	0.88
Motor efficiency (%)	94.8	95.1	95.6
Losses (kW)	309	293	387
Cooling gas	N_2/He	Methane	Methane
Inlet temperature (°C)	57	33	33
Outlet temperature (°C)	103	60	69
Heat capacity (J/kg°C)	1,207	2,500	2,500
Flow (% of compressor flow)	2.6	2.0	1.3
Motor temperature rise			
Stator winding mean T (°C)	71	57	122
Stator winding mean T (°C)	128	90	155
Stator winding max. T (°C)	140	100	175
Rotor cage mean T (°C)	120	75	117

During the manufacturing phase, specific tests were performed on some components: motor casing, compressor casing, and pass-throughs were tested under pressure up to 150 bar by INIEX, the Belgian control laboratory, or by equivalent laboratories in the U.S., and accepted. Insulation and winding components were tested in a natural gas environment.

The Mopico compressor on the test bed is shown in Fig. C-187. Dummy impellers, having the same mechanical characteristics as the real ones, were used for the no-load conditions. It was therefore possible to tune the magnetic bearings in the full-speed range, to measure the no-load losses (mechanical and iron losses), and to have better knowledge of the electrical parameters necessary to compute the performances of the motor.

Finally, the whole prototype was tested on-load. Several test runs were performed in order to obtain thermal stabilization of the whole circuit and particularly of the motor. These test runs were made under various conditions of speed, pressure, load, nature of gas, level of cooling flow in the motor, and operation mode of the control loop.

A test performed at 9500 rpm and 5.63 MW shaft power in series mode with a mixture of nitrogen and helium was very close to the design nominal conditions (9850 rpm, 5.74 MW). Table C-13 summarizes the test results. Comparing the first two columns shows the influence of the nature of the gas and of an efficient cooling on the motor temperature rise. With 23 percent less relative cooling flow, the temperature rise is reduced by 20 percent. By virtue of the large safety margin in temperature, it could be possible either to increase the output power or to decrease the cooling flow.

The last column shows a theoretically available shaft power of 8.5 MW without excessive motor temperature rise and with a constant absolute cooling flow.

Interaction of magnetic bearings and rotor. Magnetic bearings have, besides the well-known advantages of no wear, no lubrication, and low power consumption, also the reputation of being able to solve any rotordynamic problem. This is not true. On the contrary, they can cause a variety of new problems.

The Mopico compressor has, at present, the heaviest rotor running at a speed of 10,000 rpm on magnetic bearings. The stiffness and damping coefficients of

magnetic bearings depend on the vibration frequency, whereas for oil bearings they depend on the rotating speed. Compared to oil bearings, magnetic bearings have a lower stiffness. In the frequency range of 50 to 200 Hz, the stiffness of the magnetic bearings is only about one sixth of the stiffness that an oil bearing for Mopico would have at a speed of 10,000 rpm. In order to prevent large static deflections due to the low stiffness, the controller of the magnetic bearing has an integration term.

Phase lead cells in the controller provide the damping of the magnetic bearing—however, only in a limited frequency range. At very low frequencies (below about 30 Hz) and at very high frequencies (above 1500 Hz), the damping is negative. In the frequency range of 50 to 200 Hz, the damping coefficient of the magnetic bearing is of the same order of magnitude as that of an oil bearing.

The theoretical model of the rotor bearing system is well proven by measured closed loop transfer functions (relation of the displacement at the sensor and a magnetic excitation force at the bearing).

The rotor can be safely run up to the design maximum speed of 10,000 rpm. The vibration level at this speed is not more than about 50 percent of the limit due to amplifier saturation. If the cold rotor is run up slowly to this speed (within 10 to 15 min), the level is even lower.

Isotherm turbocompressors

Turbocompressors with the lowest power consumption. The word *isotherm* describes the principal feature of these machines; the flow medium is cooled intensively during the compression process in order to come as close as possible to ideal isothermal compression, giving maximum efficiency and therefore minimum power requirement.

Isotherm compressors are particularly suited to oil-free compression of air, oxygen, or nitrogen to discharge pressure up to 13 bar. For higher pressure ratios a booster may be added to the compressor train. The isotherm compressors are widely used in air separation plants, fertilizer plants, iron and steelworks, and for compressed-air supplies to mines. See Figs. C-192 and C-193 for illustration of some typical applications. The design has been steadily improved since they were first introduced in 1913. An important factor for their great success is their low specific power requirement, a result of flow path and intensive intercooling. Compact, standardized construction and high availability are required of them. More than 1000 isotherm compressors are in service throughout the world, many with up to 200,000 h to their credit and a time between overhauls of three to six years.

Operating range. See Fig. C-193.

Design features. Design features are incorporated to suit an end user's application. Typically they include:

1. Low power consumption
2. Resistance against corrosion
3. High rotor stability = low vibration level
4. Low noise level
5. Shaft-string configuration
6. Ease of installation
7. Simple maintenance programs
8. Minimized space requirement
9. High reliability

magnetic bearings depend on the vibration frequency, whereas for oil bearings they depend on the rotating speed. Compared to oil bearings, magnetic bearings have a lower stiffness. In the frequency range of 50 to 200 Hz, the stiffness of the magnetic bearings is only about one sixth of the stiffness that an oil bearing for Mopico would have at a speed of 10,000 rpm. In order to prevent large static deflections due to the low stiffness, the controller of the magnetic bearing has an integration term.

Phase lead cells in the controller provide the damping of the magnetic bearing—however, only in a limited frequency range. At very low frequencies (below about 30 Hz) and at very high frequencies (above 1500 Hz), the damping is negative. In the frequency range of 50 to 200 Hz, the damping coefficient of the magnetic bearing is of the same order of magnitude as that of an oil bearing.

The theoretical model of the rotor bearing system is well proven by measured closed loop transfer functions (relation of the displacement at the sensor and a magnetic excitation force at the bearing).

The rotor can be safely run up to the design maximum speed of 10,000 rpm. The vibration level at this speed is not more than about 50 percent of the limit due to amplifier saturation. If the cold rotor is run up slowly to this speed (within 10 to 15 min), the level is even lower.

Isotherm turbocompressors

Turbocompressors with the lowest power consumption. The word *isotherm* describes the principal feature of these machines; the flow medium is cooled intensively during the compression process in order to come as close as possible to ideal isothermal compression, giving maximum efficiency and therefore minimum power requirement.

Isotherm compressors are particularly suited to oil-free compression of air, oxygen, or nitrogen to discharge pressure up to 13 bar. For higher pressure ratios a booster may be added to the compressor train. The isotherm compressors are widely used in air separation plants, fertilizer plants, iron and steelworks, and for compressed-air supplies to mines. See Figs. C-192 and C-193 for illustration of some typical applications. The design has been steadily improved since they were first introduced in 1913. An important factor for their great success is their low specific power requirement, a result of flow path and intensive intercooling. Compact, standardized construction and high availability are required of them. More than 1000 isotherm compressors are in service throughout the world, many with up to 200,000 h to their credit and a time between overhauls of three to six years.

Operating range. See Fig. C-193.

Design features. Design features are incorporated to suit an end user's application. Typically they include:

1. Low power consumption
2. Resistance against corrosion
3. High rotor stability = low vibration level
4. Low noise level
5. Shaft-string configuration
6. Ease of installation
7. Simple maintenance programs
8. Minimized space requirement
9. High reliability

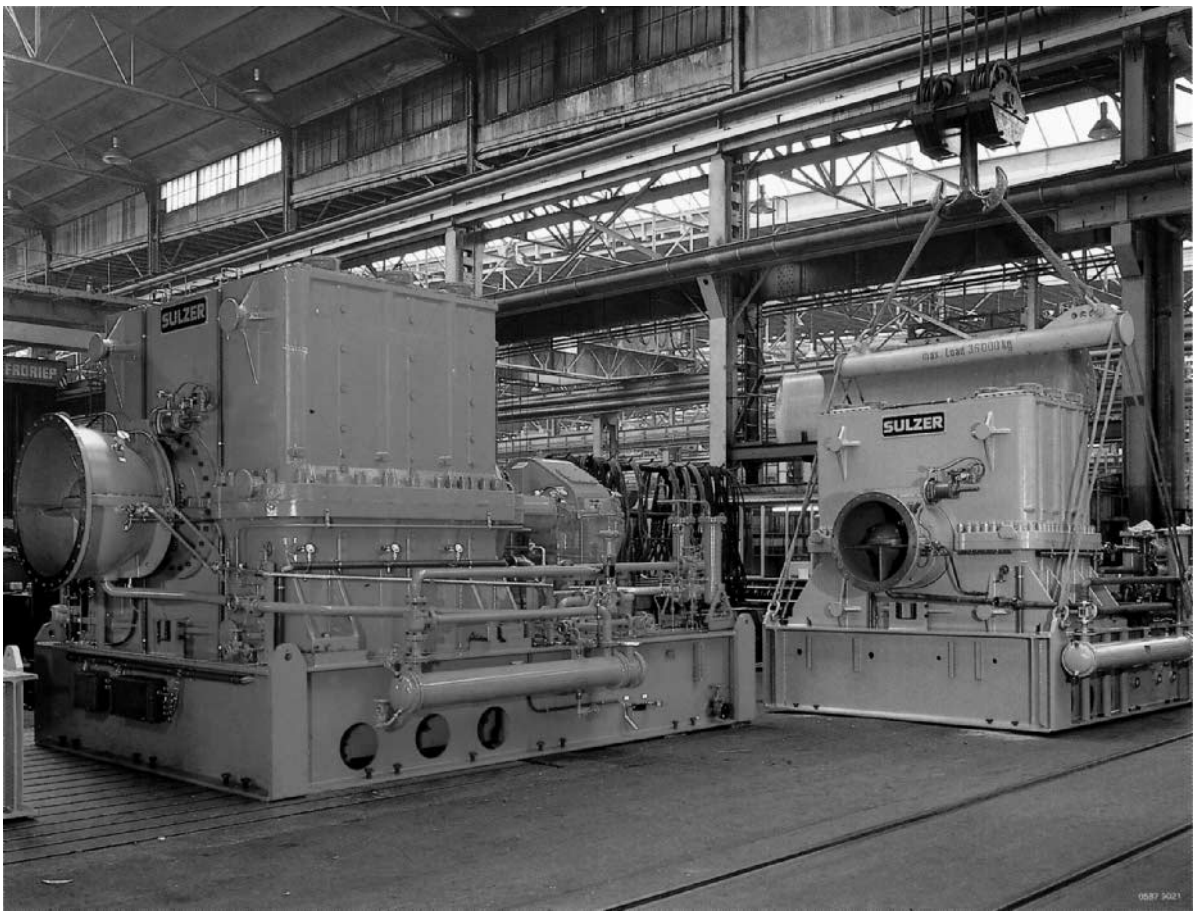
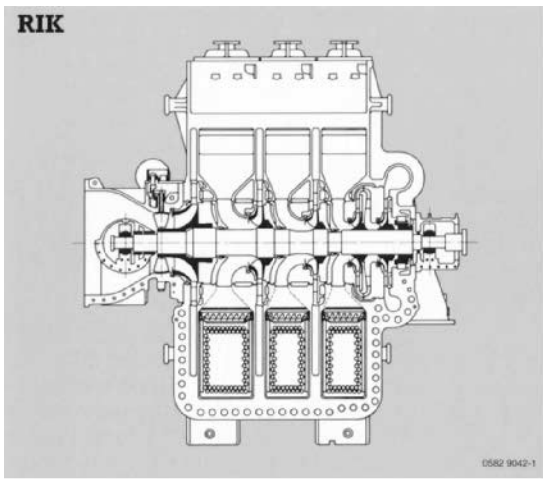
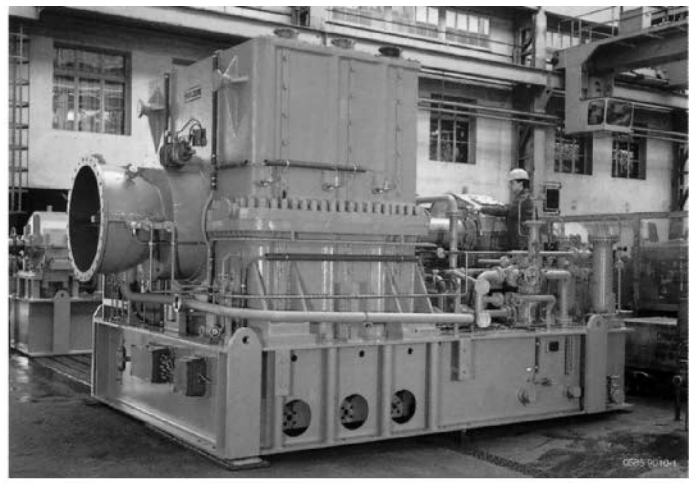


FIG. C-192 Air compressors, type RIK 80 and RIK 56. Transportation as a single-lift package. (Source: Sulzer-Burckhardt.)

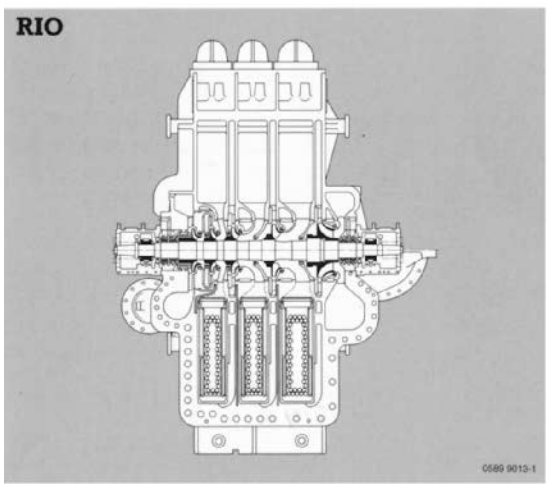
1. *Low power consumption*
   - Intercooling of the gas reduces the inlet temperature into the subsequent stage and therefore its power requirement.
   - Optimization of the distribution of the total cooling surface within the individual cooling stages with respect to heat load, cooling effect, and air-side pressure drop adds to overall efficiency.
   - The short flow path achieved by the single-shaft in-line design with cooler tube bundles integrated in the casing reduces the pressure losses on the gas side as there is no external piping.
   - The staggered high-flow impellers ensure optimal combined performance of all stages, avoiding the lower range of flow coefficients that exhibit a drop in efficiency (Fig. C-194). All impellers are of fully welded or welded and brazed construction (Fig. C-195).
   - In case of dirty cooling water, an automatic cleaning system for the cooler tubes can be installed. This would extend time between overhauls without impairing long-term efficiency.
2. *Resistance against corrosion*
   - Most of the carefully designed flow path is handling hot superheated air; the not-quite-saturated air after the cooler is taken by the shortest way to the next impeller (Fig. C-196).



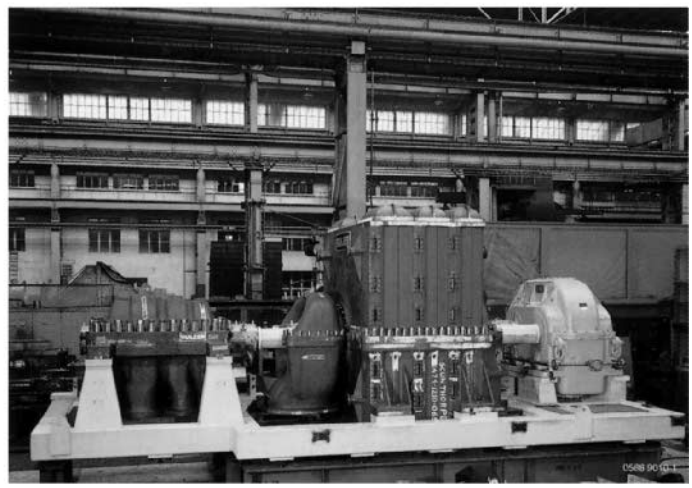
Type RIK
Flow range from 35 000 to 400 000 m³/h.
Nominal discharge pressure up to 13 bar.



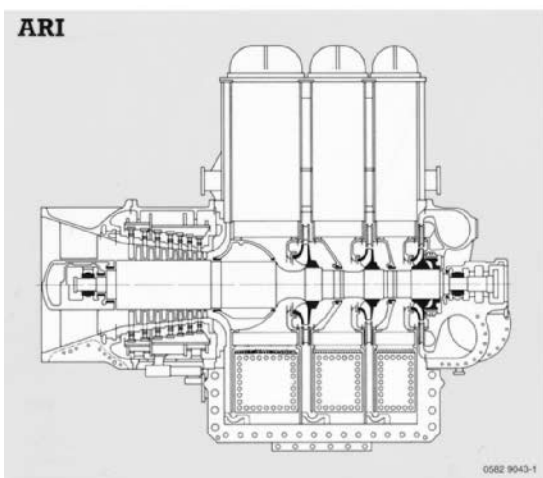
Air compressor, type RIK 71, for an air separation plant in the USA.



Type RIO
For oxygen; flow range from 12 000 m³/h to 90 000 m³/h.
Nominal discharge pressure up to 20 bar.



Oxygen compressor group with LP compressor, type RIO 35, for Europes largest air separation plant



Type ARI
Flow range up to 600 000 m³/h.
Nominal discharge pressure up to 11 bar.



Air compressor, type ARI 56, in a Chinese air separation plant.

FIG. C-193 Operating ranges and applications of isotherm compressors. (Source: Sulzer-Burckhardt.)

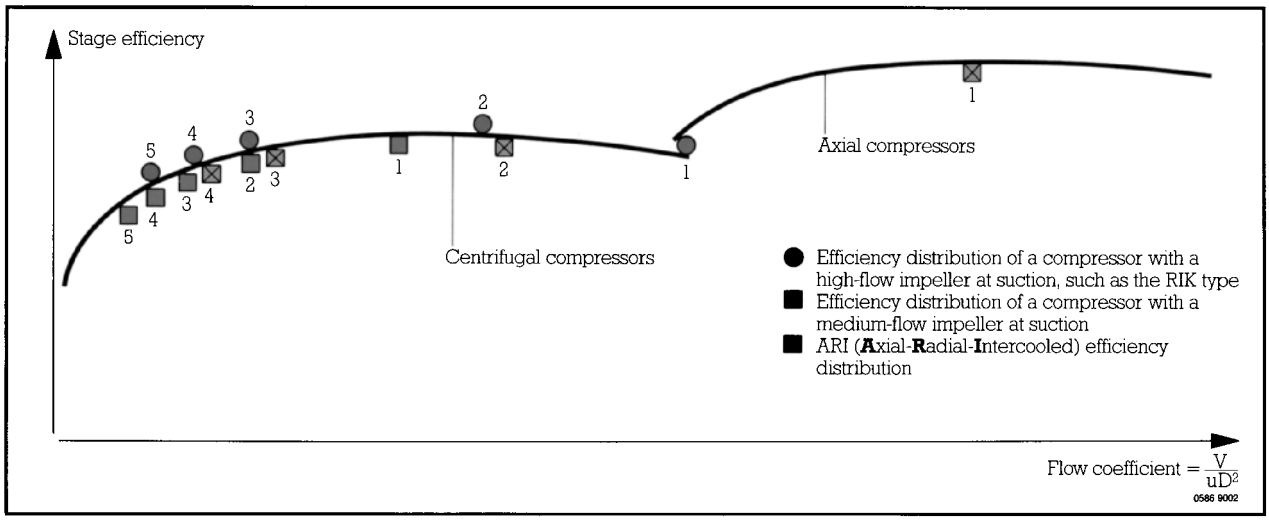


FIG. C-194 Influence of flow coefficient of the first impeller on the efficiency of subsequent stages. (Source: Sulzer-Burckhardt.)



FIG. C-195 Welded impeller of high flow coefficient. (Source: Sulzer-Burckhardt.)

- The vertical position of the built-in lateral cooler tube bundles the inertia-type water separators fitted at the outlet of the coolers (except for the RIO types) to show high separation efficiency enhanced by the effective condensate removal by gravity (Figs. C-197 and C-198). Due to this and the subcooling effect along the tube fins, the air entering the following stage has a mean temperature just slightly above the dew point, which again reduces erosion and corrosion. The condensate is drained by automatic traps.

3. Rotor stability. Radial bearings and special coupling techniques help turbocompressor rotors achieve high stability under all practical operating conditions. This is achieved by the main features illustrated in Table C-14 (Fig. C-199); Figs. C-200 and C-201 show typical rotor assembly layouts.

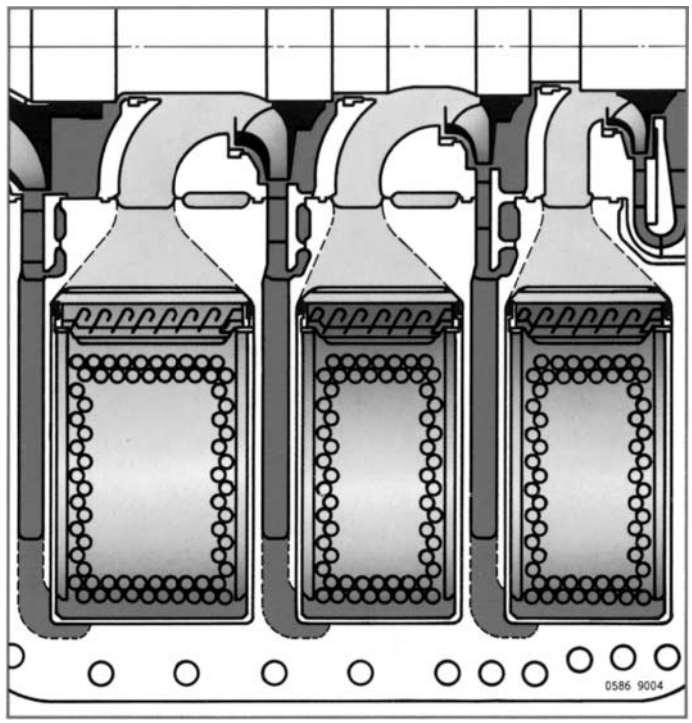


FIG. C-196 Temperature and humidity conditions: hot cool, but well above dew point; cold, but not yet saturated; cold and condensing. (Source: Sulzer-Burckhardt.)

4. Low noise level. The radial casing with the built-in coolers has an attenuating effect on the noise generated by the active high-velocity parts embedded in this compact outer package. The same applies to the double-casing axial part of the ARI series. The noise level is therefore lower than that of a centrifugal compressor with separate external coolers and the necessary interconnecting piping. In case of severe noise level restrictions, a noise hood covers compressor and gears.

5. Simple shaft-string configuration. The single-shaft in-line concept allows a simple configuration of a complete motor or steam turbine driven compressor train with the least number of shafts and bearings. See Figs. C-202 and C-203. The common choices generally are:

a. Steam tubine direct drive with one common axial thrust bearing in the steam turbine and solid coupling with flexible intermediate shaft between turbine and compressor. Four journal bearings. Axial thrust compensated. No additional load on thrust bearing due to torque lock. Standard for ARI and semipackaged RIK types.
b. Steam turbine direct drive with individual thrust bearings. Four journal bearings. Axial thrust not compensated. Additional load on thrust bearing due to torque lock caused by thermal expansion of shafts taken up by conventional gear coupling. Depending on magnitude of transient thermal expansion of compressor and turbine rotor, a diaphragm-type coupling can be used, reducing the additional axial thrust and requiring no lubrication. Alternative for ARI and semipackaged RIK types.

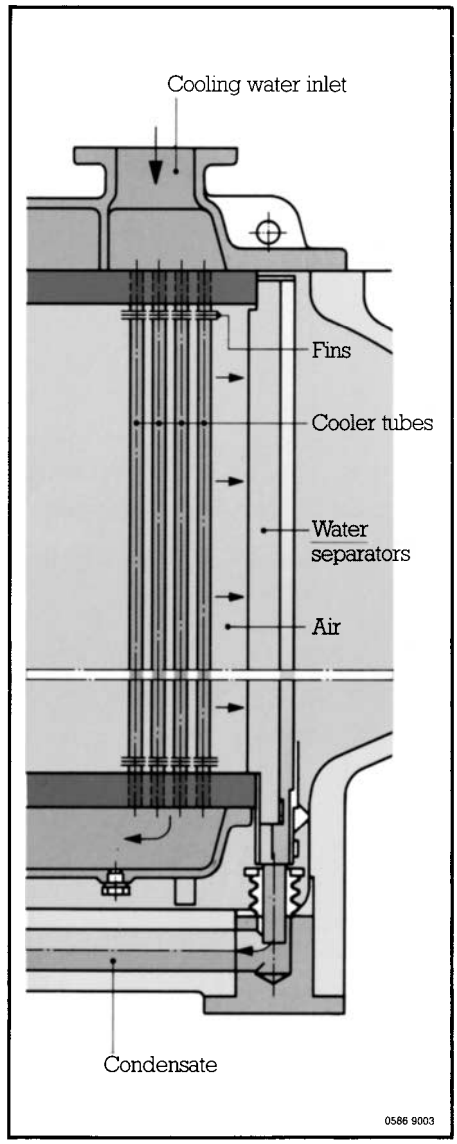


FIG. C-197 Intercooler outlet with water separators. (Source: Sulzer-Burckhardt.)

c. Motor drive with speed increasing gear of conventional design. Axial thrust bearing on compressor and gearwheel shaft. Eight journal bearings including gear and motor. Solid coupling and flexible shaft between compressor and gear; no torque lock. Solution suitable for high-power, low-speed compressors. Preferred alternative for the large ARI types.
d. For higher speeds with motor drive, the OEM applies a solution described in Figs. C-204, C-205A, and C-205B.

Solid couplings with thrust collar gears. Particularly for electric motor driven high-speed compressors of medium to high power and/or pressure, it is normal practice to make extensive use of solid couplings allowing the use of only one axial thrust bearing for single- or multiple-casing arrangements.

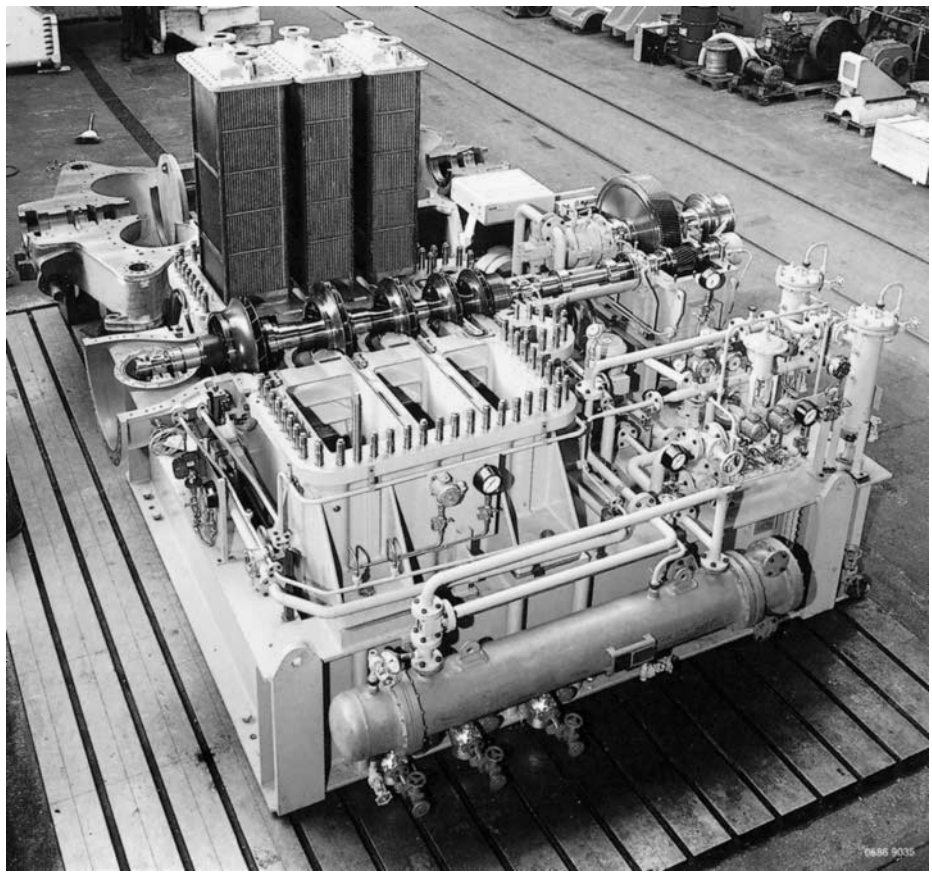


FIG. C-198 RIK model with intercooler tube bundles in the casing bottom half. The vertical water separators at the cooler outlet ensure effective drainage of the condensate. (Source: Sulzer-Burckhardt.)

TABLE C-14 Design Features Illustrated in Figs. C-199 through C-201

Design	Advantages
1 Solid, sturdy rotor with shrunk-on dummy piston	Minimum sensitivity to critical speeds and unbalance due to higher rotor stability; reduction of rotor internal damping
2 Shrink fit secured by symmetrically arranged radial dowels for impellers	No need for keys and distance bushings; fixation ensures concentricity and perfect balance under extreme operating conditions; allows larger shaft diameters; reduces stress on shaft and impeller
3 No shaft sleeves between stages	Reduces rotor hysteresis and increases running stability
4 Labyrinths always on the rotating element. Stainless steel strips caulked into the shaft and impeller grooves	No distortion of rotor due to local heating up in case of rubbing; labyrinths can be refitted easily
5 Nickel plating or other coating of shaft portions exposed to corrosion, if necessary	Plating instead of shaft sleeves is a more direct protection; allows larger shaft diameters
6 Tilting-pad radial bearings for higher speeds	Improves running stability; no oil-whip; higher external damping
7 Solid coupling, tightly bolted to flexible intermediate shaft	Improves reliability due to elimination of high-speed thrust bearing and toothed-type couplings; no torque lock thrust on high-speed thrust bearing
8 High-flow impeller at suction	Improves overall efficiency

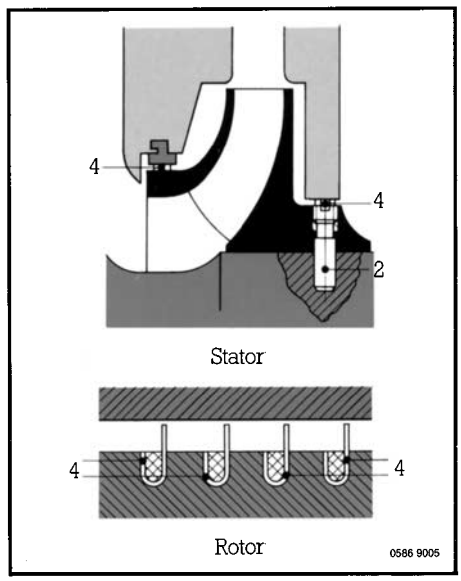


FIG. C-199 Design principles of isotherm turbocompressor rotors. (Match numbers on the figure with features in Table C-14.) (Source: Sulzer-Burckhardt.)

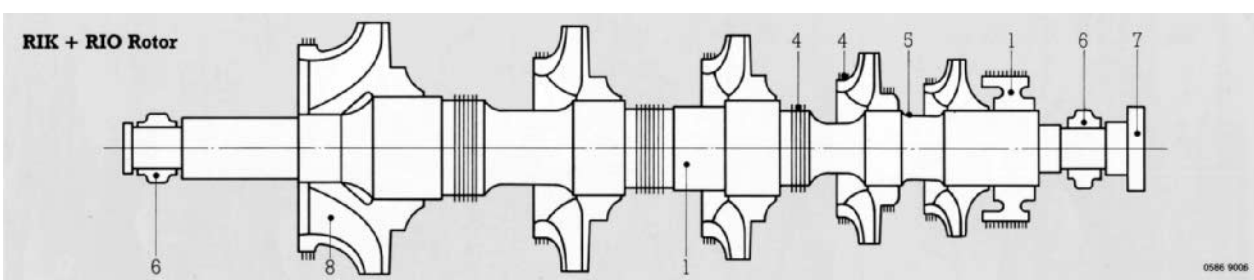


FIG. C-200 Typical rotor assembly layout: RIK and RIO models. (Source: Sulzer-Burckhardt.)

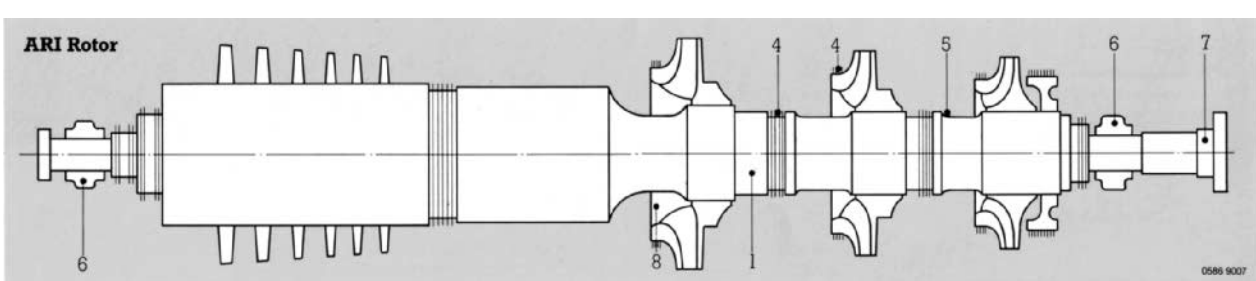


FIG. C-201 Typical rotor assembly layout: ARI model. (Source: Sulzer-Burckhardt.)

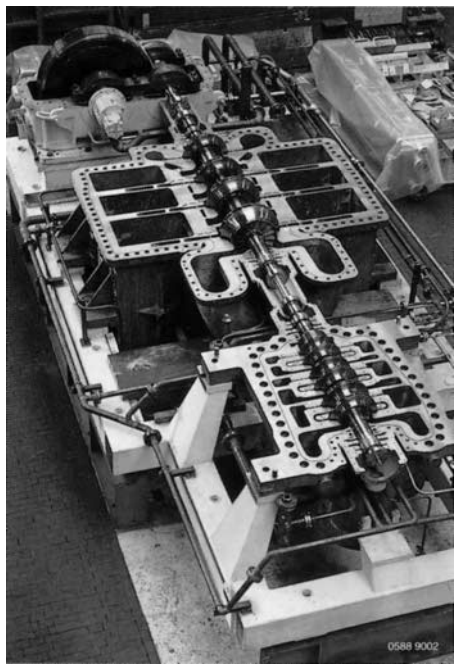


FIG. C-202 Typical shaft-string configuration of a motor-driven isotherm compressor with booster. Axial thrust transmission according to Figs. C-204, C-205A, and C-205B with one single thrust bearing on the low-speed side of main gear. (Source: Sulzer-Burckhardt.)

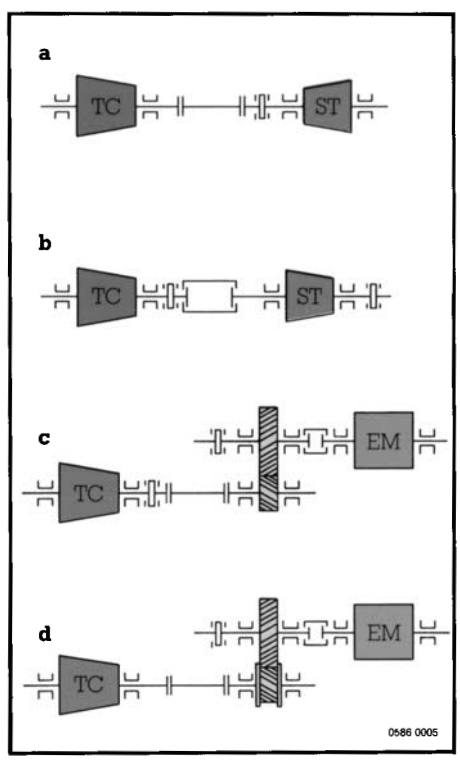


FIG. C-203 Shaft-string configurations. (Source: Sulzer-Burckhardt.)

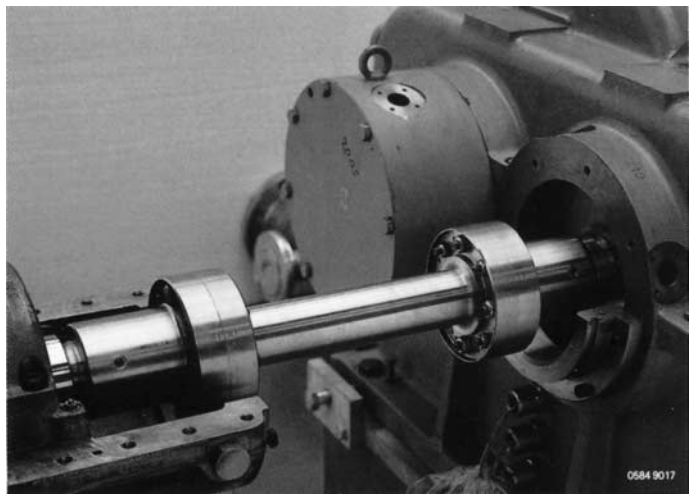


FIG. C-204 The solid quill-shaft coupling conforms to API 671 standard and consists of the quill shaft and the two hubs hydraulically fitted onto the shaft ends of the connected machines. On each coupling side, an equal number of tie bolts for axial fixation and tapered dowel pins for torque transmission and centering ensure a clearly defined connection. Balancing as a complete assembled unit and correlative marking enable removal and remounting of this intermediate shaft with the connected rotors remaining in place, without affecting the balancing quality and vibration behavior of the complete string. (Source: Sulzer-Burckhardt.)

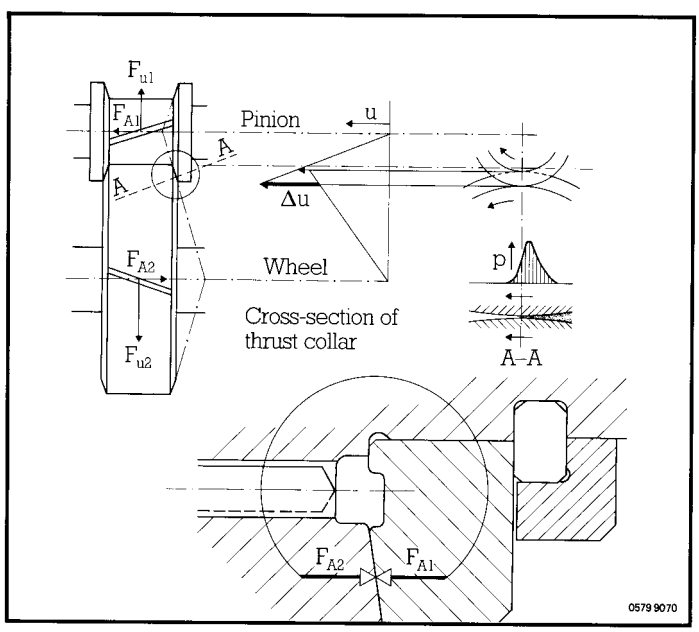


FIG. C-205A Method of axial thrust transfer in a single helical gear with thrust collar. F_u = peripheral force, F_A = axial force, u = peripheral speed, p = pressure. (Source: Sulzer-Burckhardt.)

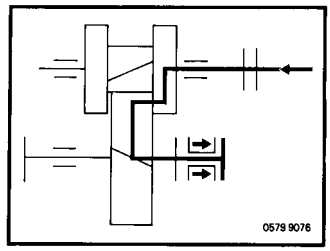


FIG. C-205B Transfer of external forces. (Source: Sulzer-Burckhardt.)

An intermediate shaft, flexible enough to allow for considerable misalignment, is inserted between the two shaft ends of the machines to be coupled together (Fig. C-204). In case of motor-driven units, the normal technique is to use single helical gears provided with thrust collars on the pinion shaft, as shown in Figs. C-205A and C-205B. The thrust collars not only neutralize the axial thrust created by the meshing of the teeth cut at an angle to the axis of the shaft, but also transmit the residual axial thrust of the high-speed rotor train to the thrust bearing on the low-speed wheel shaft.

Good gear meshing requires parallelity of gear and pinion shaft and automatically ensures parallelity of the contact surfaces of thrust collar and wheel rim. The slight tapering of the thrust collars is responsible for the formation of a wedge-type oil film creating a pressure zone spread out on an enlarged surface with a pressure distribution very similar to that of a standard-oil-lubricated journal bearing.

The relative motion between the two contact surfaces of the thrust collar system is a combination of rolling and sliding and takes place near the pitch circle diameter, resulting in a very small relative velocity. The thrust transmission is therefore effected with almost no mechanical losses. The considerably reduced losses of the single thrust bearing on the low-speed shaft as compared with the high losses of individual thrust bearings on the high-speed train lead to a substantial power saving. Moreover, this low-speed bearing can be more robustly dimensioned to provide a much higher overload capacity.

This coupling arrangement avoids heavy overhung gear couplings that are usually responsible for not clearly defined lower critical speeds and for the phenomena of torque lock leading to additional loading of the axial thrust bearing. The resulting axial friction forces can become quite substantial if insufficient attention is given to the cleanliness of the lubricating oil. This arrangement is, therefore, the preferred solution. Its strict application is clearly visible on the air compressor train (Fig. C-202).

6. Design features for erection on site and dismantling for inspection include:
- Package construction
- One single horizontal plane of the axis
- Vertical cooler bundles easily withdrawable
- No heavy and cumbersome crossover piping between compressor and external intercoolers

7. Reduced maintenance, because all components are easily accessible

8. Minimum space requirement through compact single-shaft design with integrated coolers. Low elevation of operating floor for ARI types; skid-mounted

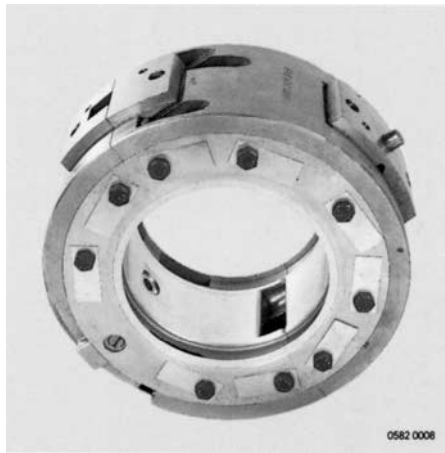


FIG. C-206 Two-lobe journal bearing. (Source: Sulzer-Burckhardt.)

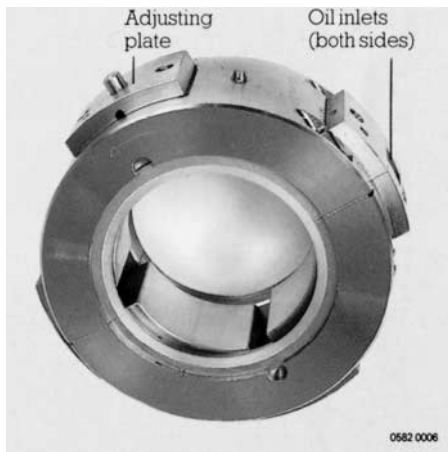


FIG. C-207 Multisegment journal bearing with four tilting pads. (Source: Sulzer-Burckhardt.)

single-life package with integrated gear and lube oil system for RIK and RIO types.

9. High reliability using generic designs

Journal and axial bearings

- *Two-lobe journal bearings* are used on the larger frame sizes of the ARI series running at a moderate speed (Fig. C-206).
- *Tilting-pad journal bearings* are incorporated in the RIK and RIO series operating in a higher-speed range. They contribute to the high rotor stability at high rotational speeds (Fig. C-207).

The horizontally split journal bearings are white-metal-lined and forced-feed-lubricated. Adjusting plates with a slight curvature in axial direction allow the bearings to be set accurately on erection. Shims placed between the plates and the bearing shell make corrective realignment easy. Thermoelement connections for white metal temperature measurement are fitted.

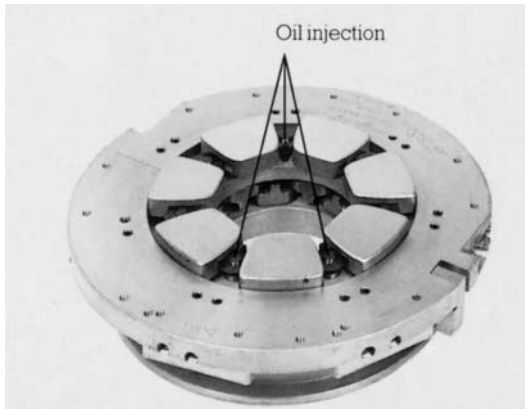


FIG. C-208 Kingsbury-type axial thrust bearing with self-equalized pads with directed lubrication. (Source: Sulzer-Burckhardt.)

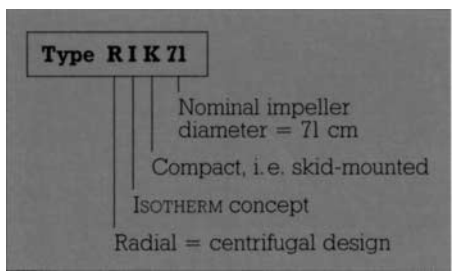


FIG. C-209 RIK series designation example: five centrifugal stages. (Source: Sulzer-Burckhardt.)

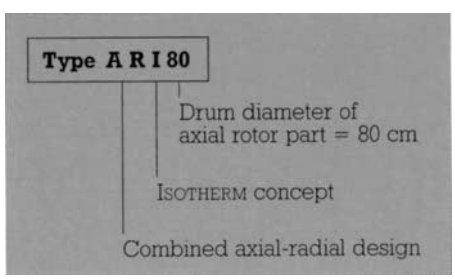


FIG. C-210 ARI series designation example: five axial and three centrifugal stages. (Source: Sulzer-Burckhardt.)

- *The axial thrust bearing* is normally located on the low-speed shaft of the gear. In multicasing arrangements with no gears it is normally located on the intermediate shaft. The thrust bearing is fitted with a load equalizing system. The pads are individually lubricated (Fig. C-208).

Performance data RIK and ARI. See Figs. C-209 and C-210 for type designation examples.

Figures C-211 through C-213 allow selection of the:

Compressor size	Nominal diameter	D (cm)
	Power input	P (kW)

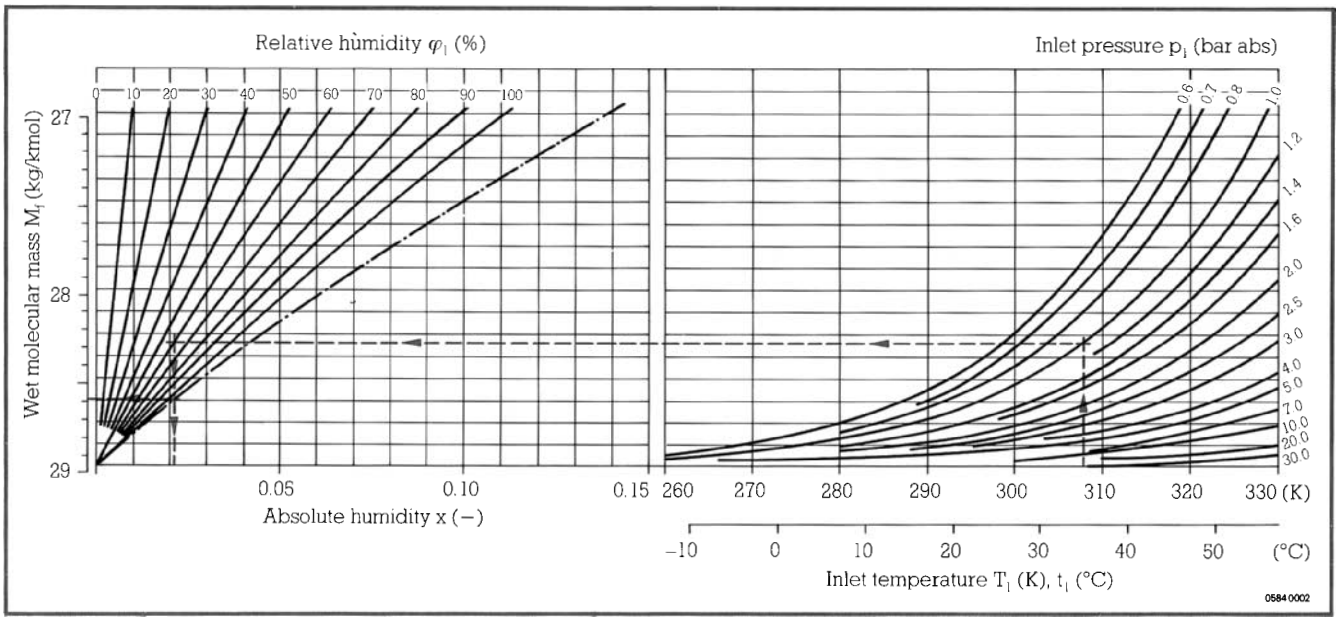


FIG. C-211 Determination of the absolute humidity *x* and the molecular mass M_f of the wet air. (Source: Sulzer-Burckhardt.)

Operating conditions	Mass flow	$\dot{m}$ (kg/s)
	Suction pressure	p_1 (bar abs)
	Suction temperature	T_1 (K)/t_1 (°C)
	Relative humidity of the air or gas	φ_1 (%)
	Discharge pressure	p_2 (bar abs)
	Molecular mass	M (kg/mol)

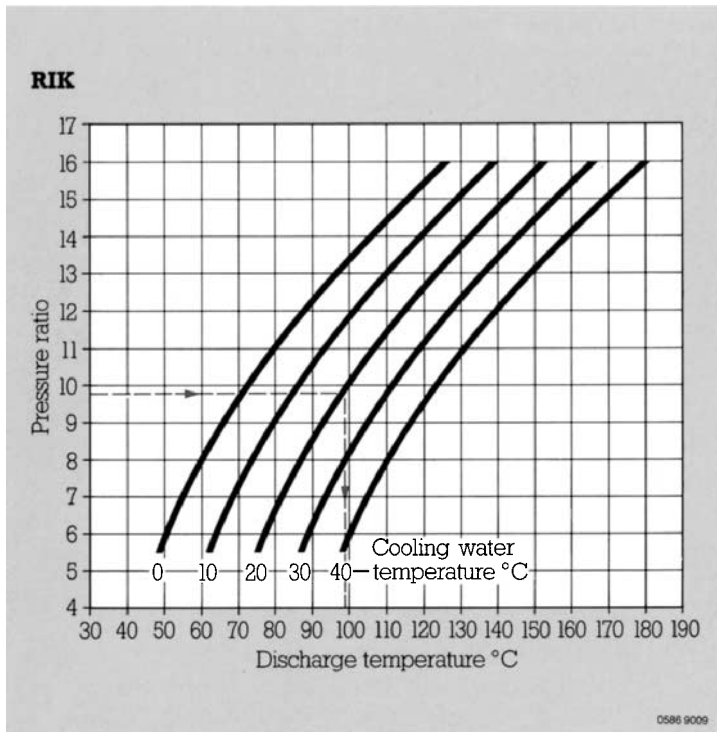
The following factors and symbols are also used for the calculation:

	Suction volume (actual)	$\dot{V}_1$ (m³/h)
	Absolute humidity	x (–)
	Isothermal efficiency	η_{iso} (%)
Indices	Suction branch	1
	Discharge branch	2
	Dry	t
	Wet	f

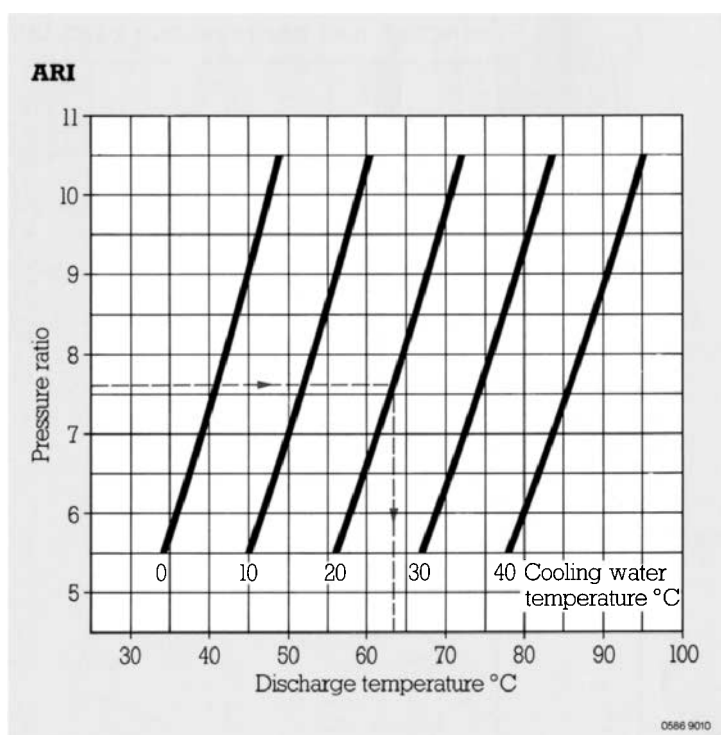
See also Table C-15 for how to specify an isotherm compressor.

In Figs. C-214 through C-216:

- NP = reference point (100 percent) = design point
- α = angular position of the inlet guide vanes (RIK models) or the adjustable stator blades (ARI models)
- Valid for air at constant inlet data.
- Depending on the specific process requirements, such as higher overload capacity, a certain pressure rise to surge, maximum efficiency at design point or rather at a certain part load, the process design point NP may be placed differently in the characteristic curve.



(A)



(B)

FIG. C-212 Determination of the discharge temperature (A) for RIK bodies, (B) for ARI bodies. (Source: Sulzer-Burckhardt.)

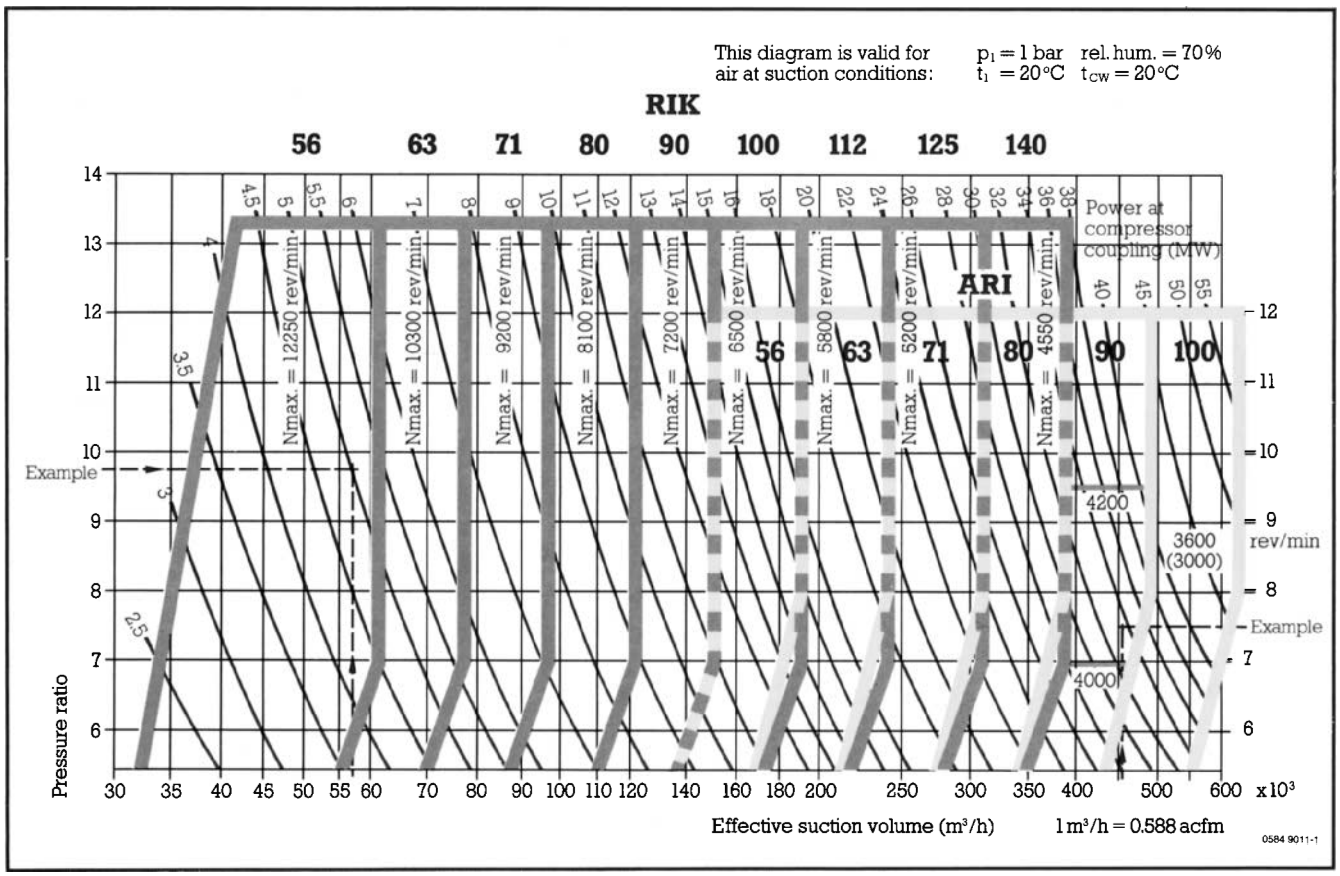


FIG. C-213 Compressor selection diagram. (Source: Sulzer-Burckhardt.)

Design features (see also Table C-16)

Skid-mounted single-lift concept. The various frame sizes are skid-mounted units with built-in intercoolers and integrated lube oil system. The erection on site includes placing the skid on a simple foundation slab, alignments with the driver, and connecting gas and cooling-water piping (Fig. C-217B).

Alternatives to the standard motor-driven concept are possible. For example:

- Compressor directly coupled to the steam turbine driver or expander
- A booster coupled to the compressor, with or without intermediate gear
- Separate freestanding lube oil system
- Suction nozzle facing downward

Casing. See Fig. C-217 for the labeling of internals.

The horizontally split casing contains the five centrifugal stages and three pairs of vertical intercoolers (Fig. C-218). The axial inlet ensures ideal flow conditions through the inlet guide vanes into the first stage. The bearings can be inspected without having to disconnect the gas or oil piping or to disturb the casing top half. The flow passages to and from the coolers are the result of exhaustive model tests and ensures that the gas flow in each stage is equally distributed between the parallel cooler elements (Fig. C-219).

TABLE C-15 Selection and Sample Performance Calculation of an Isotherm Compressor

	Example 1 Type RIK	Example 2 Type ARI
Given		
Mass flow (dry)	$\dot{m}_t$ = 17.29 kg/s	$\dot{m}_t$ = 136.48 kg/s
Suction pressure	p_1 = 1 bar	p_1 = 1 bar
Suction temperature	T_1 = 308 K; t_1 = 35°C	T_1 = 308 K; t_1 = 35°C
Relative humidity	φ_1 = 60%	φ_1 = 60%
Discharge pressure	p_2 = 9.8 bar	p_2 = 7.6 bar
Cooling-water temperature	t_w = 20°C	t_w = 20°C
Dry molecular mass	M_t = 28.96 kg/kmol	M_t = 28.96 kg/kmol
Calculation instructions		
1 Determination of the absolute humidity (x) and the wet molecular mass using Fig. C-211	x = 0.021 M_f = 28.58 kg/kmol	x = 0.021 M_f = 28.58 kg/kmol
Wet gas constant $R_f = \frac{8315}{M_f}$	R_f = 290.94 J/kgK	R_f = 290.94 J/kgK
2 Calculation of the wet mass flow $\dot{m}_f = \dot{m}_t\,(1 + x)$	$\dot{m}_f = 17.29 \cdot 1.021$ = 17.65 kg/s	$\dot{m}_f = 136.48 \cdot 1.021$ = 139.35 kg/s
3 Determination of the actual suction Volume $\dot{V}_1 = \frac{m_f \cdot R_f \cdot T_1 \cdot 3600}{P_1 \cdot 10^5}(\mathrm{m^3/h})$	$\dot{V}_1$ = 56,950 m³/h	$\dot{V}_1$ = 449,520 m³/h
4 Determination of the discharge temperature t_2 with Fig. C-212	t_2 = 98°C	t_2 = 63°C
5 Selection of the compressor frame size and power input P with Fig. C-213	PIK 56 P = 5.0 MW	ARI 90 P = 20.3 MW

Conversion factors

1,000 Nm³/h (1.013 bar, 273 K, dry) = 0.3592 kg/s
1,000 scfm (14.7 psia, 60 F, dry) = 0.5774 kg/s
1,000 scfm (14.7 psia, 70 F, dry) = 0.5665 kg/s
1 m³/h = 0.5886 cfm/1 bar = 14.5 psi

For the hydrostatic tests the casing is divided into several chambers and submitted to a water pressure of 1.5 times the maximum possible operating pressure of the corresponding compartment.

Seals. The shaft and interstage seals are of the labyrinth type. The stainless steel strips are fixed in grooves in the rotating parts (shaft, impeller hub, and cover disc) and have a very small radial clearance against stationary plastic ring segments in the corresponding partition walls. Internal and external leaks are thus kept to a minimum (Fig. C-199).

The axial thrust of the impellers is almost entirely compensated by a balance piston at the discharge end of the compressor. The piston is provided with labyrinth strips rotating against a white-metal–lined steel ring.

With the in-line arrangement of the impellers the resulting total axial thrust of the rotor is, contrary to a back-to-back arrangement, always acting in the same direction and of the same order of magnitude under all operating conditions (normal, reduced load, surge, and rotating stall). The balance piston is dimensioned in such a way that the compensated residual thrust is reduced to a minimum, but still always acting in the same direction. With this method the axial thrust bearing need not be oversized, and the bearing losses are reduced accordingly without any risk of overloading it under abnormal operating conditions.

On the suction side, a special sealing system prevents any oil or oil mist of the bearing space seeping into the surrounding suction ducts thus contaminating, for

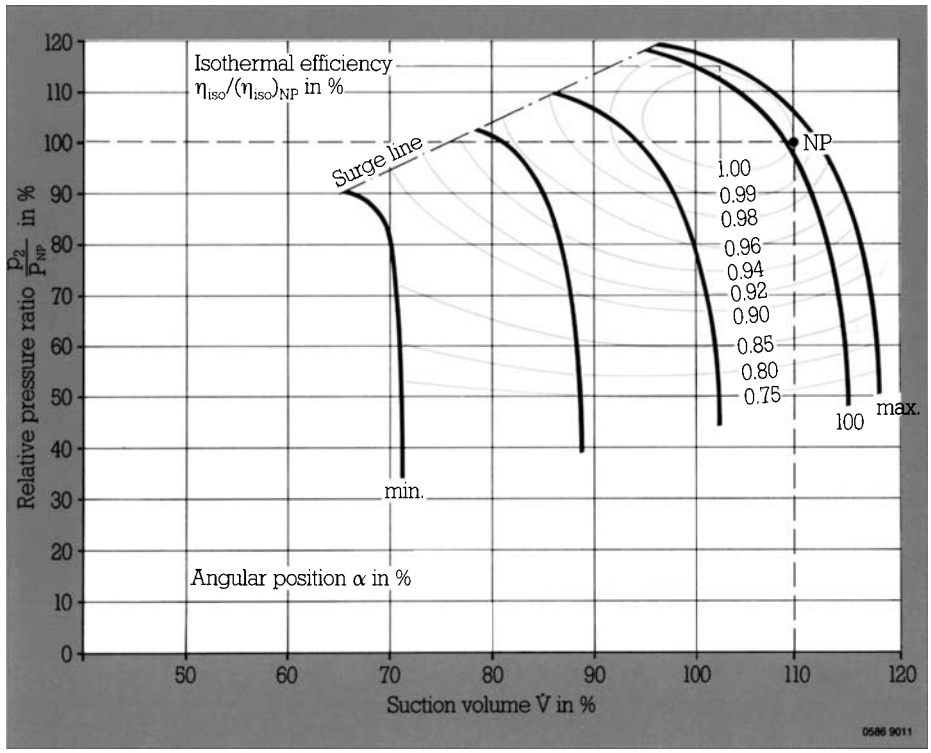


FIG. C-214 RIK compressor with inlet guide vanes and constant speed driver. (Source: Sulzer-Burckhardt.)

example, the air of an oxygen or nitrogen plant (Fig. C-220). The sealing air introduced in the middle of the shaft seal is discharged to atmosphere on the bearing side in order to avoid building up pressure in the confined bearing space connected with the oil tank.

Rotor and impellers (Fig. C-221). The impeller and shaft materials undergo a number of metallurgical tests. Further tests are carried out during manufacture. The finished impellers are then balanced at low speed and subjected to an over-speed test. The assembled rotor is dynamically balanced over the whole speed range up to full speed. Impellers with a medium to high flow coefficient are of the fully welded construction with the blades shaped in three dimensions (see Fig. C-195). Small and narrow impellers are of the combined welded/brazed construction. The sense of rotation of the rotor is clockwise, seen from the suction side.

Inlet guide vanes. To obtain the characteristics as shown in Fig. C-214 with an infinite number of operating points between maximum performance and surge line, inlet guide vanes are fitted ahead of the first stage. They are actuated by a pneumatic servomotor that may be connected to an automatic pressure or flow controller (Fig. C-222). The guide vanes are pivoting in self-lubricating bushes and are connected to an adjusting ring by a maintenance-free linkage system (Fig. C-223). Another link connects the ring with the pneumatic actuator. The absence of a lubricant avoids contamination of the process gas. When starting the compressor, the guide vanes are in an interlocked closed position to reduce the starting torque to a minimum.

Coolers. Three pairs of intercooler tube bundles are mounted in a vertical position on each side of the casing and bolted to the water box. They can expand freely. Round finned tubes ensure excellent heat transfer on the air side. The gap between

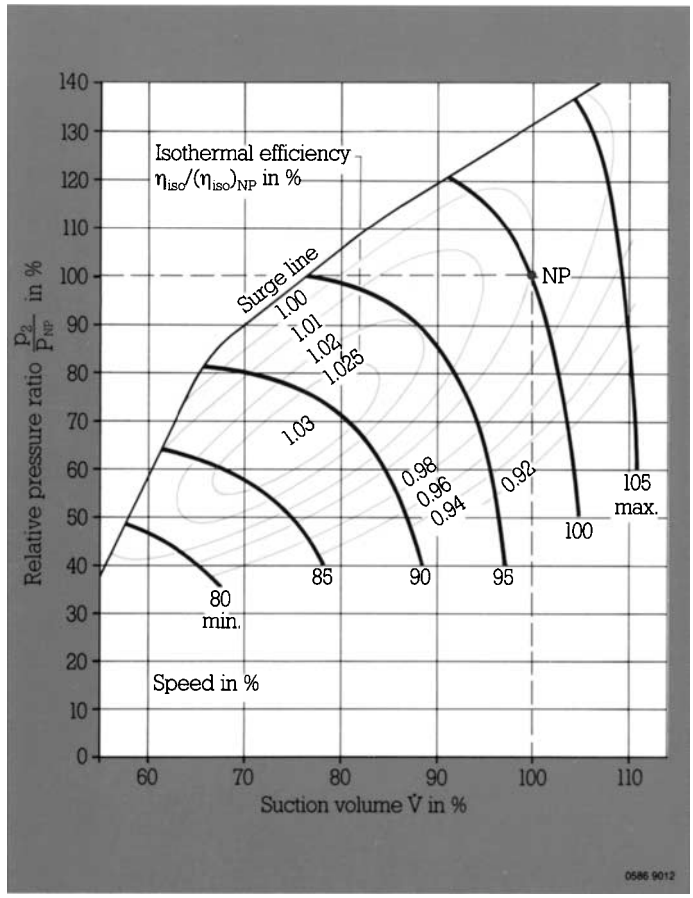


FIG. C-215 RIK compressor without inlet guide vanes, but running at variable speed. (Source: Sulzer-Burckhardt.)

tube bundle and casing at the exit of the cooler is sealed with a rubber membrane to avoid bypassing of uncooled air. See also Figs. C-224 and C-225.

RIK designation technical data and dimensions. See Table C-17 and Figs. C-226 and C-227.

Design features of the RIO isotherm designation. For an RIO series model for compressing oxygen service (see Figs. C-228 and C-229):

- Compact five-stage centrifugal in-line design
- Three pairs of vertical cooler bundles integrated in casing
- Nominal discharge pressure up to 20 bar
- Directly coupled booster compressor available for high discharge pressure

Advantages of the RIO designation. Compressor series has thorough cooling with built-in intercoolers. The compression approaches the ideal of efficient isothermal compression. Result: lowest possible energy consumption and a very compact machine. There are no external coolers, no crossover piping, and no expansion joints. Its simple, lightweight package requires less space, has low overall profile, is easy to erect, and results in minimum installation cost.

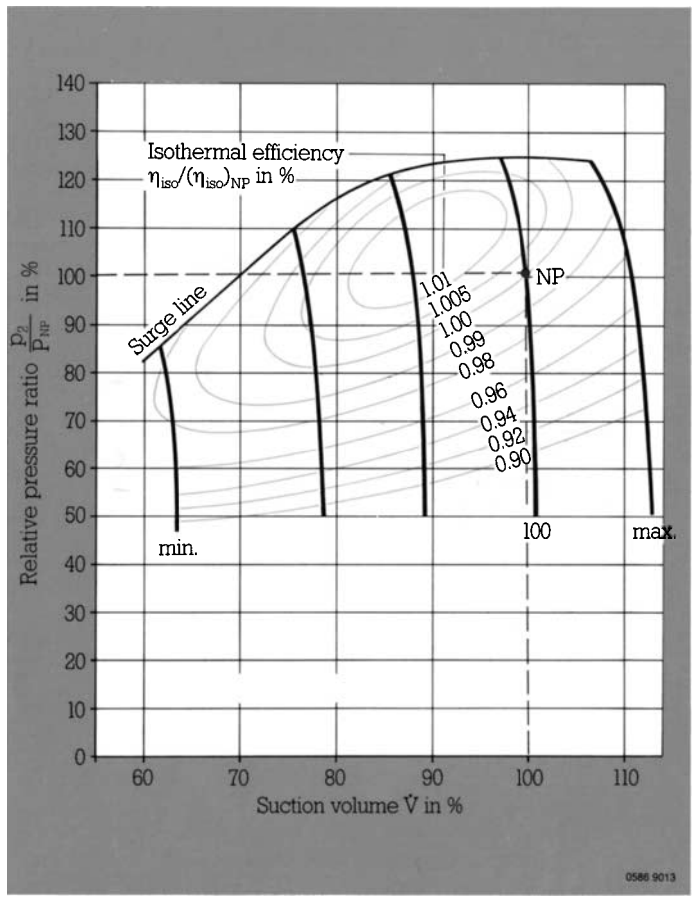


FIG. C-216 ARI compressor with adjustable stator blades and constant speed driver. (Source: Sulzer-Burckhardt.)

Technical data and dimensions. For the RIO designation see Figs. C-230 and C-231. For the ARF designation, design features include:

Axial-radial concept. Where a centrifugal isotherm compressor reaches its economical limits with regard to size, weight, and specific cost, the axial compressor design offers an attractive and technically convincing solution for large volume flows.

For equal aerodynamic loading (Mach number) of the machine and for the same tip diameter of the rotor, an axial stage will handle a volume flow about twice as large as that of a wide centrifugal impeller. If, therefore, the centrifugal section of an isotherm compressor is preceded by an axial booster with a pressure ratio of about 2, the centrifugal section is correspondingly reduced in size and its optimum speed will coincide with that of the axial part. Therefore, the two sections can be combined in one machine with one single rotor running in two bearings only, a design principle similar to that of the industrial single-shaft gas turbine.

Stage and cooler optimization for the predominant pressure ratios for air between 6 and 8 led to a compact axial-centrifugal compressor with six axial and three centrifugal stages and three pairs of intercoolers. This configuration results in an excellent overall isothermal efficiency due to the higher efficiency of the axial part and the high-stage efficiency of the subsequent three wide impellers (see Fig. C-194 and Figs. C-232 and C-233).

TABLE C-16 Materials of Construction

Part	Material	DIN Standard	Comparison ASTM Standard
Casing	Nodular cast iron*	GGG-40*	A 395*
Compressor inlet	Nodular cast iron	GGG-40	A 536
Bearing housing	Nodular cast iron	GGG-40	A 536
Partition walls	Cast iron	GG-20	A 48, Class 30
Diffusers: Discs	Nodular cast iron	GGG-40/1693	A 536
Blades	Carbon steel plates	HI/17155	A 515, Grade 55
Shaft	Low-alloy steel	28 NiCrMoV 8 5	A 470
Impellers	Forged steel	†	†
Dummy piston	Alloy steel	34 CrNiMo 6	AISI 4340
Journal bearings	Steel with white metal	CK 15 + WM	AISI 1015 + WM
Inlet guide vanes	Stainless steel	X 20 Cr 13	
Cooler tubes	Copper	SF-CuF 20 CuNi 10 Fe	
Alternatively:‡	Aluminum brass	CuZn 20 A1 F34/1785	B 111/687
	Copper nickel alloy	CuNi 10 F 29	B 111 C 70600
	Copper nickel alloy	CuNi 30 F 36	B 111 C 71500
Fins‡	Copper	For all tube alternatives	
Tube plates top	Carbon steel plate	HI/17155	A 515, Grade 55
Alternatively:	Muntz metal	CuZn 38 SNAL	B 171 C 36500
Tube plates bottom	Muntz metal	CuZn 38 SNAL	B 171 C 36500
Water separators	Stainless steel	X 5 CrNi 18 9	A 167, Grade 304
Water boxes	Cast iron	GG-20	A 48, Class 30

* Frame sizes RIK 90 and above welded design (carbon steel plate).
† On request, depending on application.
‡ Other alternatives, such as Duplex designs, on request adapted to prevailing cooling-water properties and air contamination.

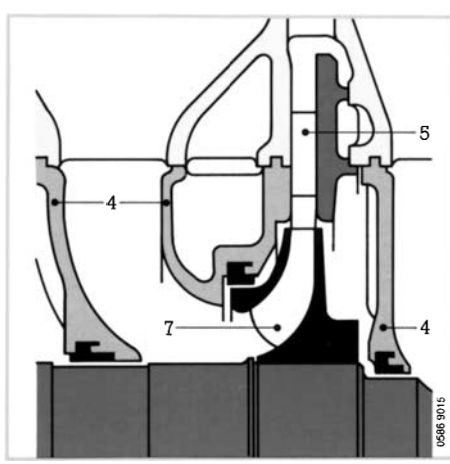


FIG. C-217A Casing internals. Impeller (7), cast iron partition walls (4), vaned diffusors (5). (Source: Sulzer-Burckhardt.)

Casing. The horizontally split casing consists of six major components (Fig. C-233). The cast axial inlet (1) and center part (2) cylinder are flanged to the welded centrifugal casing (3). The inlet casing alone or together with the center part can be lifted for the inspection of their internals while leaving the centrifugal casing in its place. The cast blade carrier (7) and diffuser wall (11) are also bolted to the

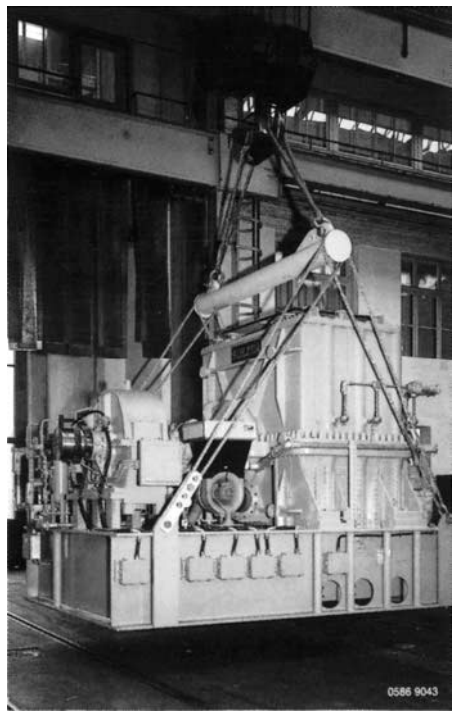


FIG. C-217B RIK skid completely shop erected, as transported to site as a single-lift package. (Source: Sulzer-Burckhardt.)

centrifugal casing. The discharge volute (4) flanged to the centrifugal casing need not be dismantled when lifting the top half of the latter for the purpose of removing the intercooler tube bundles or inspecting the impellers. For all maintenance operations the external pipe connections remain undisturbed.

The bearings are easily accessible by simply lifting the bearing housing top (6) on the discharge side or the top half of inlet casing (1) and bearing housing (5) on the suction side. The oil and sealing-air connections are located in the bottom half of the respective casings, and any inspection or maintenance work leaves them unaffected. The discharge nozzle—forming part of the bottom half discharge volute (4)—is normally pointing downward, but can also be directed horizontally. Two pendulum-type feet and two additional feet attached laterally to the centrifugal cooler casing support the machine on the foundation. See also Fig. C-234.

The flow passages to and from the intercoolers follow the same design principle as for the RIK series. See Figs. C-235 and C-236.

Blade carrier and stator blades. The blade carrier (7) and the short-diffuser wall (11) bolted together are flanged to the cooler casing (3) and can expand freely toward the suction side (Fig. C-233).

The double-casing design with outer casing and blade carrier offers various advantages:

- Rigid casing construction; the clearances in the blade duct are not influenced directly by external pipe forces.
- Simple fitting of the blades and assembly of the casing parts; the top half of the casing can be raised without dismantling the blade-adjusting mechanism.
- Possibility of fitting different blade carriers, for adapting the blade channel and thus the compressor characteristics to greatly changed operating conditions.

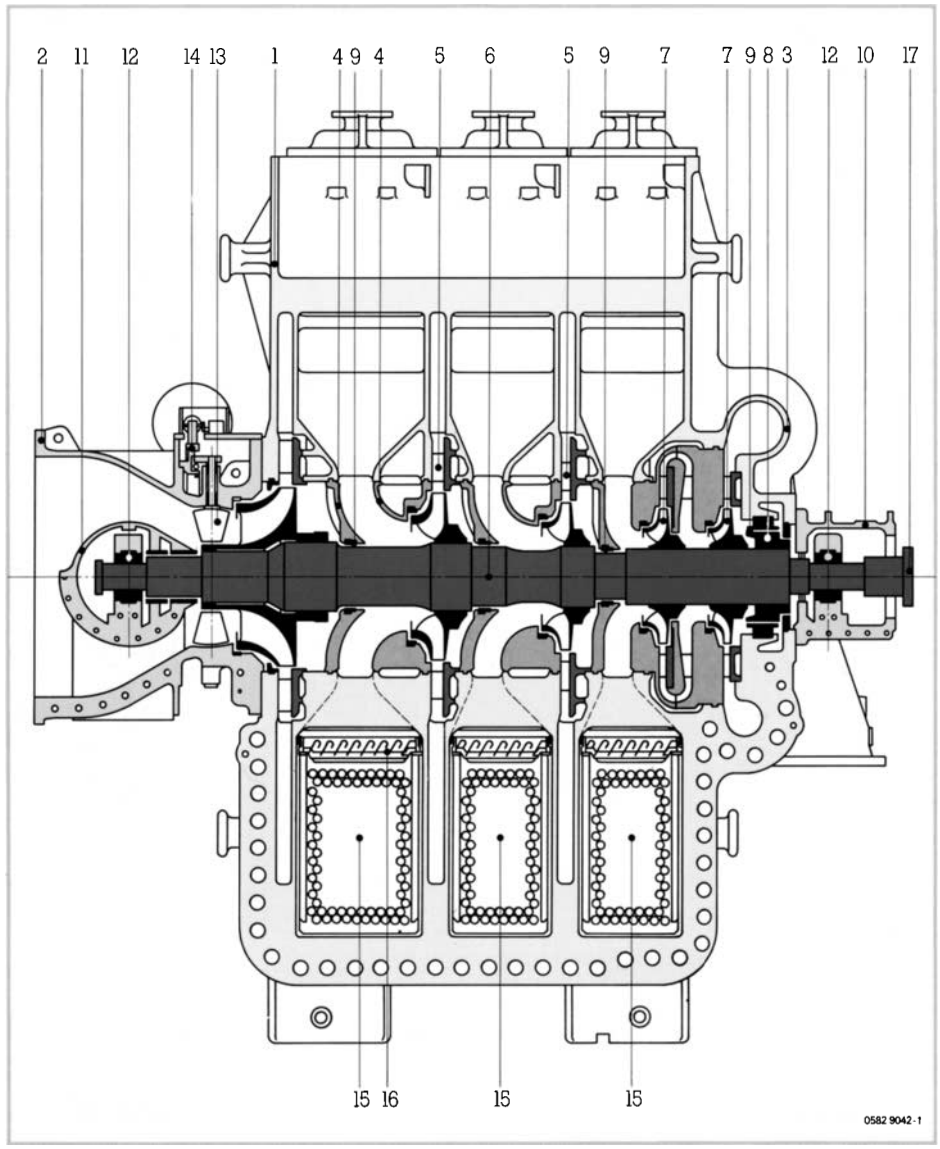


FIG. C-218 Section through an RIK series isotherm compressor (above, vertical section; below, horizontal section). Items 7, 4 and 5 are also illustrated in Fig. C-236. 1, casing; 2, inlet housing; 3, discharge volute; 4, partition walls; 5, diffusors; 6, shaft; 7, impellers; 8, balance piston; 9, shaft seals; 10, discharge-end bearing housing; 11, intake-end bearing housing; 12, journal bearings; 13, inlet guide vanes; 14, vane adjusting mechanism; 15, cooler tube bundle; 16, water separator; 17, coupling flange. (Source: Sulzer-Burckhardt.)

- Optimal protection of the adjusting mechanism in the space between the casing and blade carrier; the space is kept under suction pressure in order to safeguard the adjusting mechanism against condensation and corrosion attack.

Each of the adjustable stator blades (9) is made of one piece with a cylindrical shaft. The latter is seated in a bearing bush in the blade carrier (Figs. C-237 and C-238). The high damping characteristics of this seating arrangement practically excludes the occurrence of dangerous vibration amplitudes associated with the stator blades.

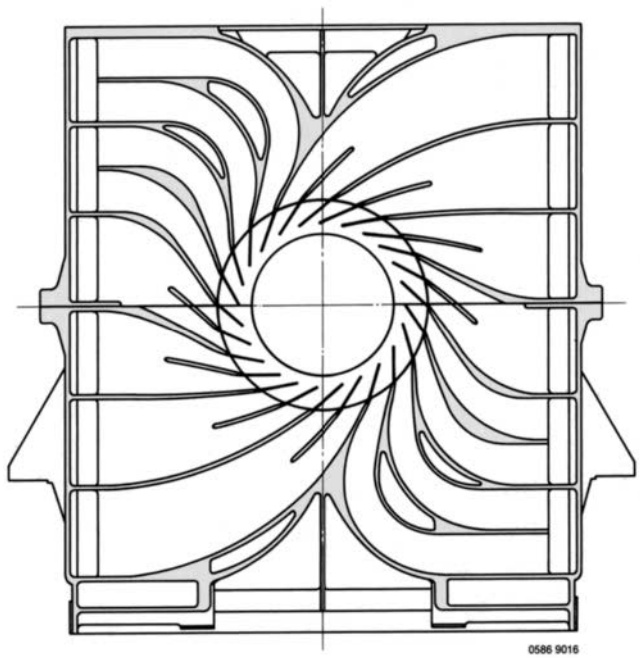


FIG. C-219 Cross-section through diffuser and return channel. (Source: Sulzer-Burckhardt.)

TABLE C-17 Technical Data and Dimensions

	RIK 56	RIK 63	RIK 71	RIK 80	RIK 90	RIK 100	RIK 112	RIK 125	RIK 140
A	4,600	4,480	5,900	6,500	6,700	7,230	8,450	9,300	10,400
B	3,800	4,000	4,200	4,400	4,900	5,270	5,750	6,200	6,500
C	3,400	4,100	3,700	4,000	5,700*	6,600*	7,300*	8,100*	9,000*
D	1,840	2,250	2,040	2,140	3,200	3,350	3,800	4,500	5,000
E	500	560	560	630	630	630	630	710	710
a	800	900	1,000	1,100	1,200	1,400	1,600	1,800	2,000
b	300	350	400	450	500	600	700	800	900
H_1	4,800	5,600	5,400	5,700	7,200	7,750	8,700	9,900	11,000
H_2	6,700	8,200	7,400	7,900	10,100	10,850	12,100	13,700	15,200
H_3	5,300	6,500	6,300	6,700	9,000	9,650	10,800	12,100	13,400
G_1	8.2	11.1	14	17.3	24	29	41	57	80
G_2	1.1	1.4	1.6	2.0	2.3	2.5	3.2	4.5	6.3
G_3	25	33	40	50	69	85	120	165	230
G_4	36	48	53	66	100	120	155	205	270
mr^2	18	27	45	85	150	280	500	880	1,570
Q 4 passes	325	325	430	540	593	680	760	900	1,025
2 passes	690	690	900	1,150	1,186	1,360	1,520	1,800	2,050
F	3,350	3,700	4,500	5,600	5,600	8,000	8,000	10,000	10,000

Technical data, dimensions, and weights
(dimensions in mm, weights G in metric tons)
E = Gear center distance (average)
H_1 = Crane height to lift casing over rotor
H_2 = Crane height to lift cooler tube bundles over casing
H_3 = Crane height to lift casing top half over cooler tube bundles
G_1 = Casing top half
G_2 = Heaviest single cooler tube bundle
G_3 = Bare compressor
G_4 = Complete skid, max.
mr^2 = Compressor rotor mass moment of inertia in $kg\,m^2$, referred to compressor speed, max.
Q = Cooling-water rate of compressor intercoolers in m^3/h
F = Maximum oil filling of oil tank in liters
* incl. servomotor for guide vane drive.

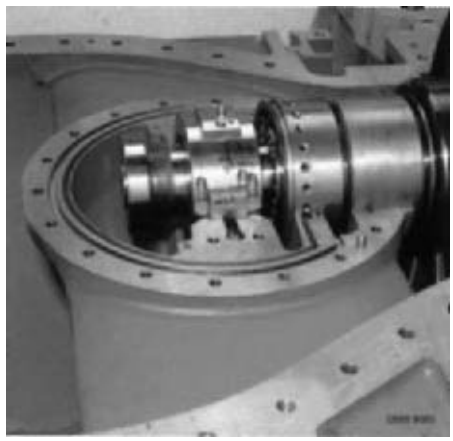


FIG. C-220 Bearing housing suction side with special sealing system. (Source: Sulzer-Burckhardt.)



FIG. C-221 Rotor. (Source: Sulzer-Burckhardt.)

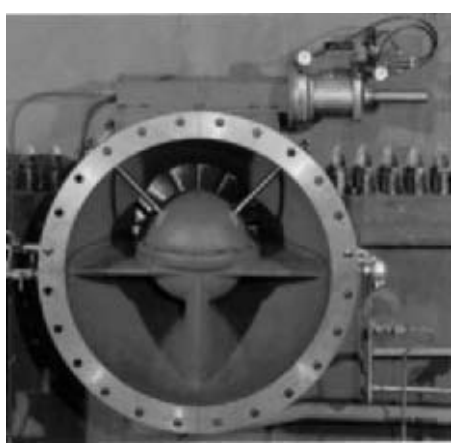


FIG. C-222 Inlet guide vanes with pneumatic actuator. (Source: Sulzer-Burckhardt.)



FIG. C-223 Inlet housing with inlet guide vane linkage system. (Source: Sulzer-Burckhardt.)

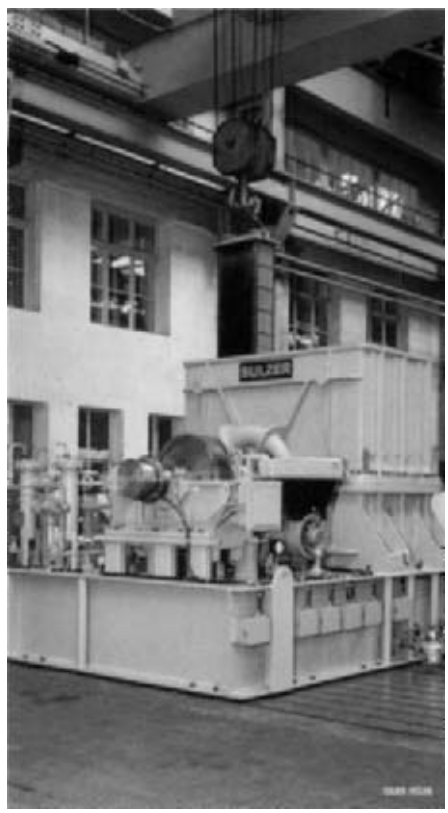


FIG. C-224 For inspection of the intercoolers, each tube bundle can be withdrawn individually. (Source: Sulzer-Burckhardt.)

Stator blade adjusting mechanism. The adjusting mechanism is located in the annular space between casing and blade carrier. It is maintenance-free and does not require any lubrication.

The adjusting mechanism is operated by means of two hydraulic servomotors (10) that are affixed laterally to the bottom-half casing. One of the servomotors is equipped with a positioning transmitter and the second operates hydraulically in parallel (Fig. C-239).

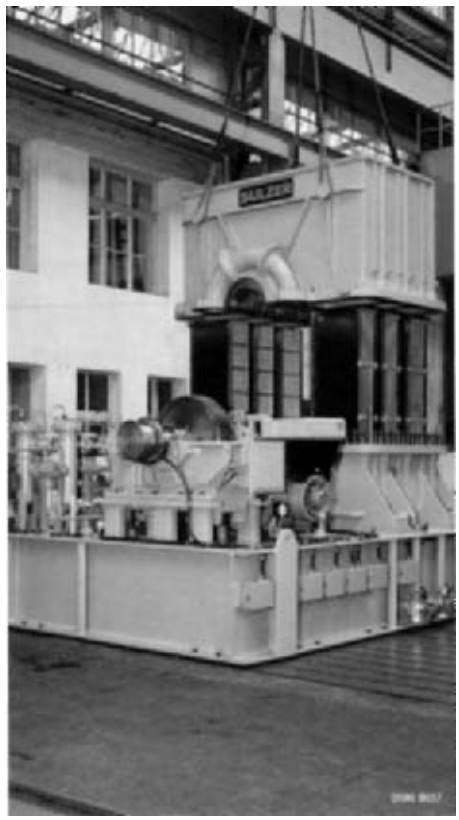


FIG. C-225 For inspection of rotor and casing internals, the intercoolers need not necessarily be withdrawn. (Source: Sulzer-Burckhardt.)

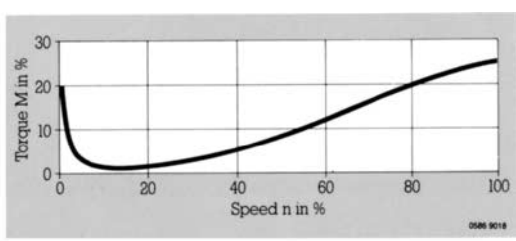


FIG. C-226 Compressor starting torque with closed inlet guide vanes. (Source: Sulzer-Burckhardt.)

The linear movement of the servomotor piston rods is transmitted directly to the adjusting cylinder (8) by way of two ball and socket joints. The adjusting cylinder of welded design can move in the axial direction and is dry-seated. There is no restriction of heat expansion in any direction. U-shaped guide rings are provided on the inner side in which the adjusting levers are engaged. These rings facilitate the assembly of the adjusting cylinder with the stator blade linkages.

The adjusting levers provided on the end of each stator blade shaft are connected to the guide rings of the adjusting cylinder by means of pivoting sliders. The axial movement of the cylinder is converted into a rotating movement of the stator blades (Fig. C-240).

The self-lubricating bearing bushes of the blade shafts are seated in the radial holes of the blade carrier. O-ring packings prevent the ingress of contaminants into the stator blade seating.

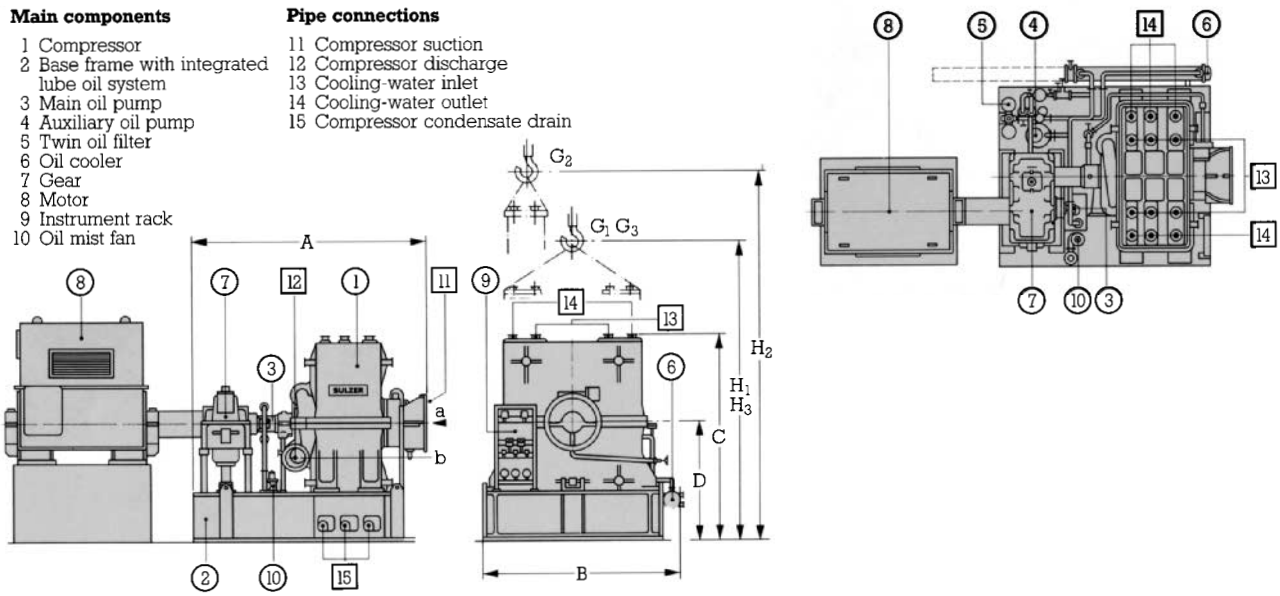


FIG. C-227 Main components and pipe connections (RIK designation). (Source: Sulzer-Burckhardt.)

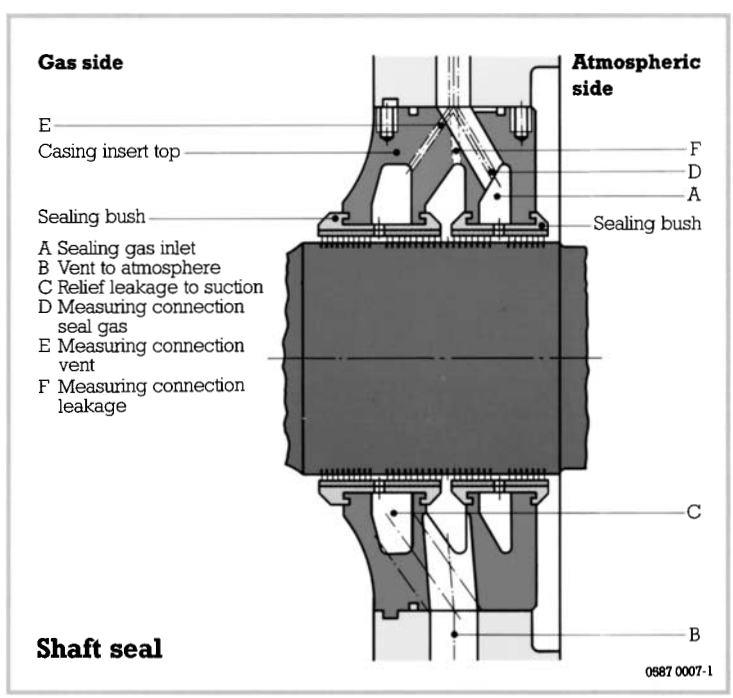


FIG. C-228 Shaft seal layout (RIO designation). (Source: Sulzer-Burckhardt.)

Suction side bearing seal. Double compressed-air sealing on the suction side combined with a double-walled bearing housing (5) that vents to the atmosphere, preventing any sacking in of oil mist in the event of subatmospheric pressure at the machine inlet (Fig. C-241).

Power oil supply. A separate high-pressure control oil unit (Fig. C-242) actuates the hydraulic servomotor of the adjustable axial stator blades. This control oil

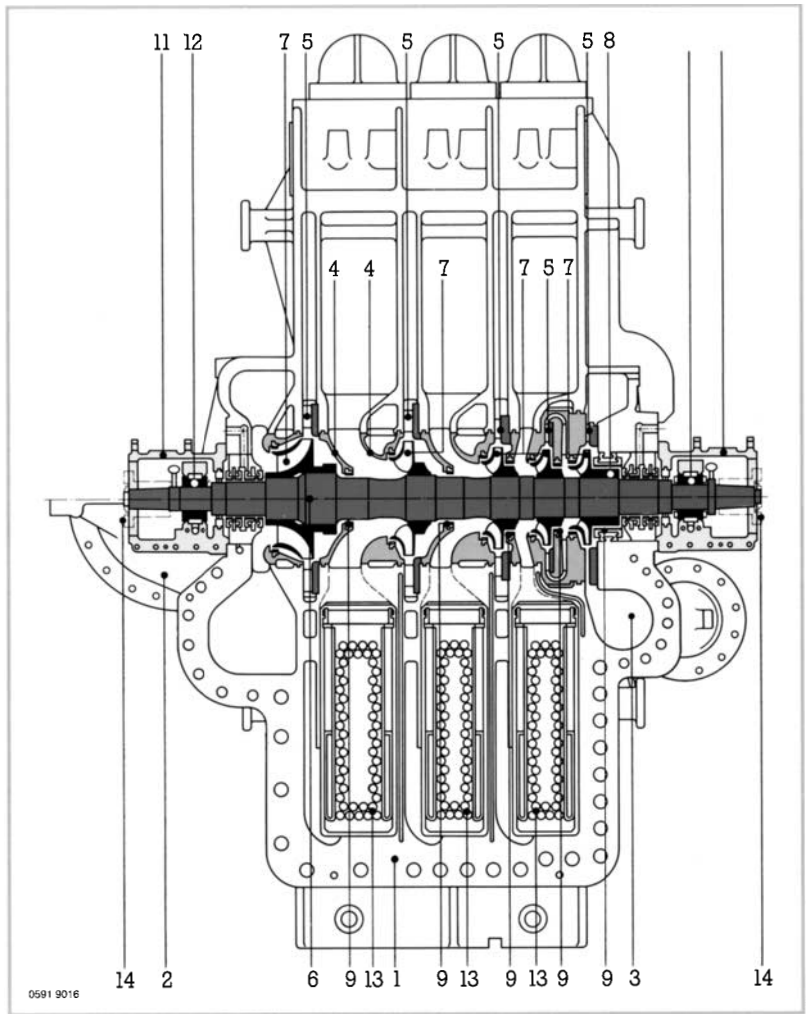
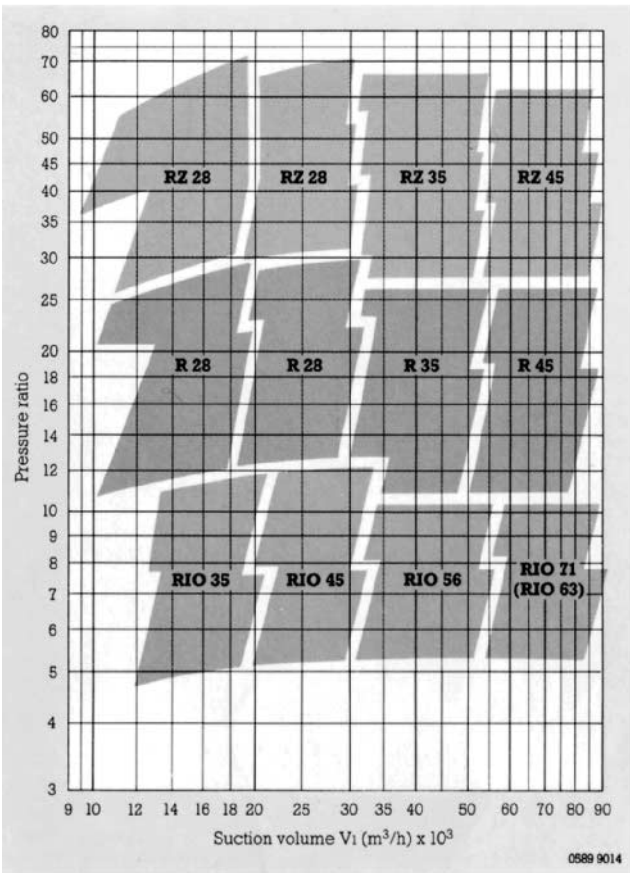


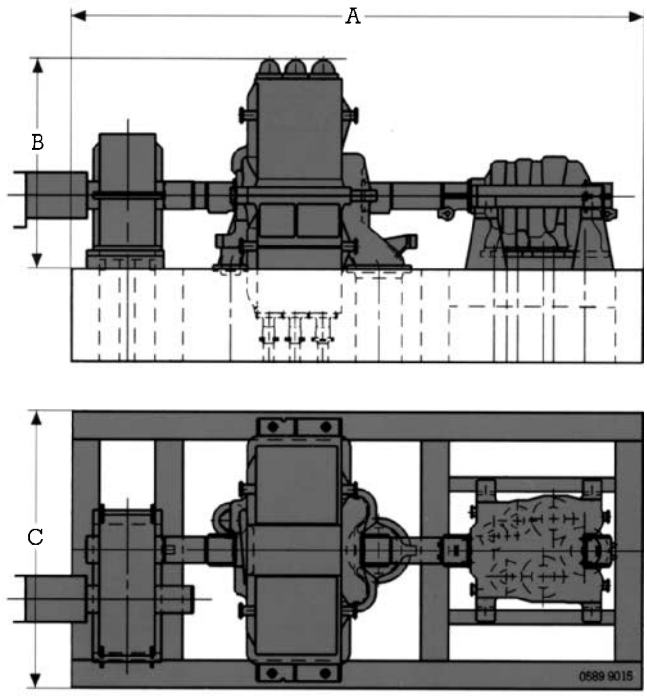
FIG. C-229 Main components (RIO designation). 1, casing; 2, inlet housing; 3, discharge volute; 4, partition walls; 5, diffusers; 6, shaft; 7, impellers; 8, balance piston; 9, seals; 10, discharge-end bearing housing; 11, intake-end bearing housing; 12, journal bearings; 13, cooler tube bundle; 14, coupling flange. (Source: Sulzer-Burckhardt.)

unit comprises an oil tank, two motor-driven pumps, a changeover-type twin oil filter, two bubble accumulators, a regulating valve for constant pressure, and the necessary instrumentation. All components are mounted on a bedplate and piped up accordingly. In case of failure of the control oil pumps the two accumulators will supply enough oil for a quick and safe reaction of the control elements.

Rotor. The basic design features are shown in Fig. C-199 and Table C-14. Both axial and centrifugal parts form a single forged monobloc shaft (see Fig. C-243). The axial blades have rhomboidal fir-tree roots and are firmly braced in an exactly defined position in peripheral grooves of the shaft (Fig. C-238). The three centrifugal impellers of a high flow coefficient, and therefore high efficiency (Fig. C-195), are of the fully welded construction with three-dimensional blades (Fig. C-196). They are balanced at low speed and subjected to an overspeed test. The assembled and bladed rotor is then dynamically balanced over the whole speed range up to full



230



Dimensions for RIO + RZ compressor group

	A	B	C
RIO 35 + RZ 28	6200	2300	3000
RIO 45 + RZ 28	7000	2500	3200
RIO 56 + RZ 35	7200	2600	3200
RIO 63 + RZ 35	7300	2700	3300

231

Technical data (FIG. C-230) and dimensions (FIG. C-231) (RIO designation) of five frame sizes covering flow range from 12,000 to 90,000 m³/h. (Source: Sulzer-Burckhardt.)

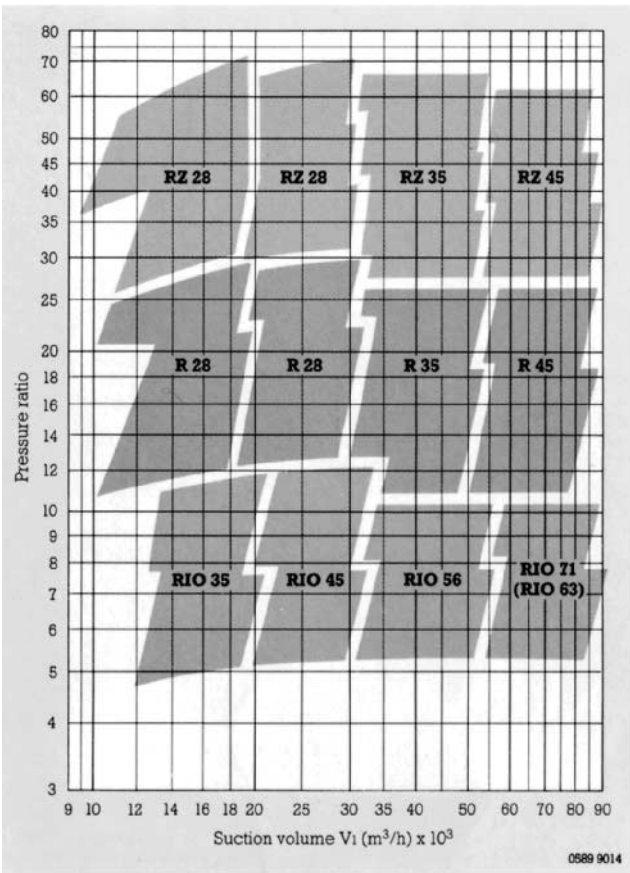
speed. The sense of rotation of the rotor is clockwise, seen from the suction side. See also Figs. C-243 and C-244.

Coolers. The three pairs of intercooler tube bundles are mounted in a vertical position on each side of the centrifugal casing and bolted to the lower water box. They can freely expand upward. The upper water box is fixed to the upper tube plate and guided in the top water chamber cover (Fig. C-245). Round finned tubes ensure excellent heat transfer on the air side. The gap between tube bundle and casing at the exit of the cooler is sealed with a rubber membrane to avoid bypassing of uncooled air. The cooling-water connection and condensate drains are located at the bottom.

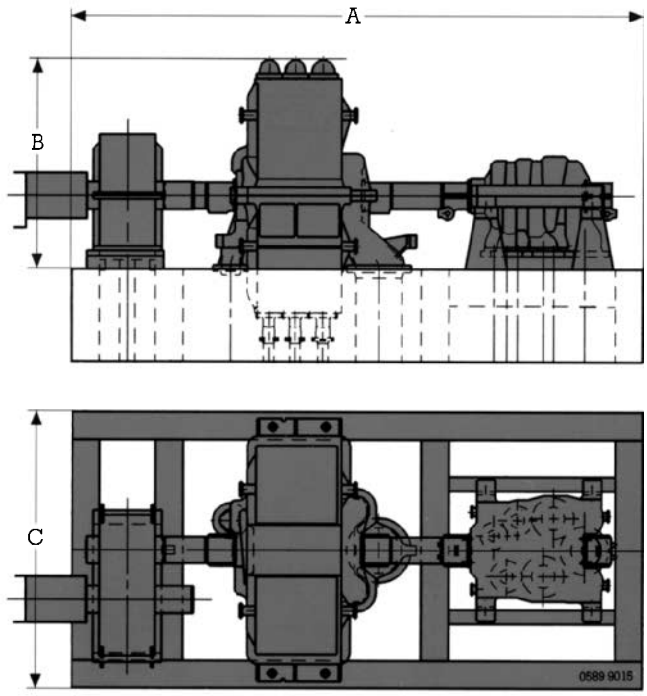
Integrally geared compressors*

Multistage integrally geared compressors. The design principle of integrally geared centrifugal compressors was established, implemented, and patented by Demag

* Source: Demag Delaval, USA.



230



Dimensions for RIO + RZ compressor group

	A	B	C
RIO 35 + RZ 28	6200	2300	3000
RIO 45 + RZ 28	7000	2500	3200
RIO 56 + RZ 35	7200	2600	3200
RIO 63 + RZ 35	7300	2700	3300

231

Technical data (FIG. C-230) and dimensions (FIG. C-231) (RIO designation) of five frame sizes covering flow range from 12,000 to 90,000 m³/h. (Source: Sulzer-Burckhardt.)

speed. The sense of rotation of the rotor is clockwise, seen from the suction side. See also Figs. C-243 and C-244.

Coolers. The three pairs of intercooler tube bundles are mounted in a vertical position on each side of the centrifugal casing and bolted to the lower water box. They can freely expand upward. The upper water box is fixed to the upper tube plate and guided in the top water chamber cover (Fig. C-245). Round finned tubes ensure excellent heat transfer on the air side. The gap between tube bundle and casing at the exit of the cooler is sealed with a rubber membrane to avoid bypassing of uncooled air. The cooling-water connection and condensate drains are located at the bottom.

Integrally geared compressors*

Multistage integrally geared compressors. The design principle of integrally geared centrifugal compressors was established, implemented, and patented by Demag

* Source: Demag Delaval, USA.

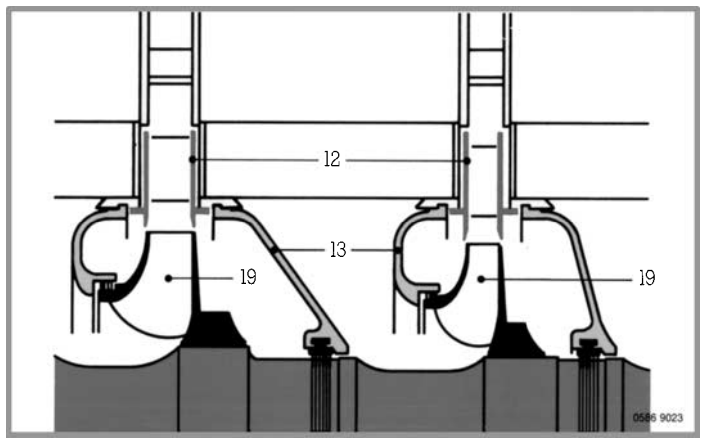


FIG. C-232 Centrifugal casing internals. (Source: Sulzer-Burckhardt.)

Delaval more than 50 years ago. Since then, this concept has been continuously further developed, enabling it to be successfully employed in demanding applications in process engineering. Figures C-246 through C-250 illustrate some typical applications. Figure C-251 illustrates the internal components of a multistage integrally geared compressor.

This information source's model VK/HVK integrally geared centrifugal compressors are characterized by a combination of several compression stages with one gear assembly, forming a compact unit for the multistage compression of a wide range of gaseous media.

Intergrally geared compressors offer the following advantages:

- Low power consumption due to concept, modern impellers, tailored aerodynamics, and optimized auxiliaries
- Wide control range and good part-load efficiency due to adjustable guide vane units in the first or on all compression stages
- Multiservice capability and many application options due to a system of standard components
- Package design available for almost all VK/HVK compressor sizes. A package includes compressor, process gas coolers, lube oil console, process piping, and all tubing and wiring, thus significantly reducing on-site installation time

Technical performance data (see Fig. C-252)

- Capacities from 1000 to 480,000 m^3/h (600 to 280,000 cfm)
- Discharge pressure up to 100 bar (1450 psi)
- Gearbox ratings up to 45,000 kW (60,000 hp)

Fields of application include

- Air separation
- Synthetic fibers
- Ammonia
- Ethylene oxide
- Hydrocarbons and more

Single-stage integrally geared or direct driven centrifugal compressors. Series SEZ/SEE process compressors represent the single-stage concept derived from multistage integrally geared VK/HVK machines.

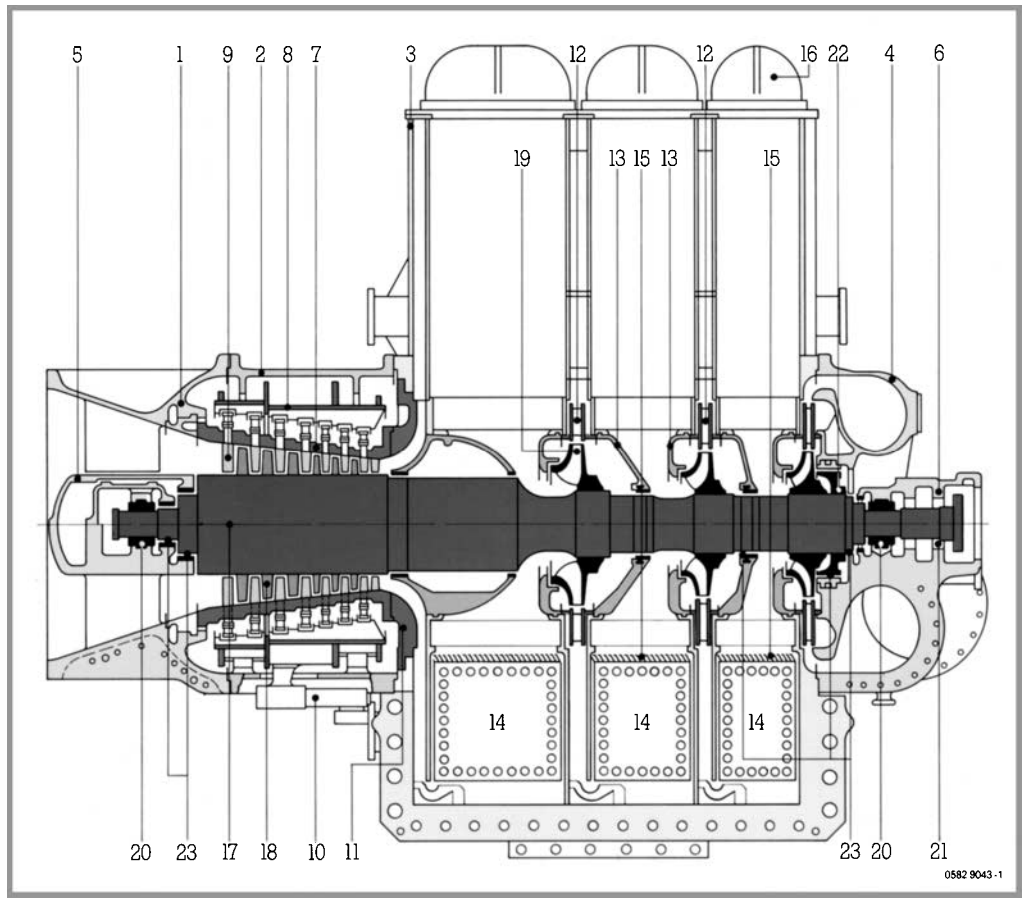


FIG. C-233 Section through an ARI series isotherm compressor (above, vertical section; below, horizontal section). 1, inlet casing of axial section; 2, center part of axial section; 3, casing of radial section; 4, discharge volute; 5, suction-side double-walled bearing housing; 6, bearing housing, discharge side; 7, blade carrier; 8, blade adjusting cylinder; 9, adjustable stator blades; 10, servomotor; 11, short-diffuser wall; 12, bladed diffusers; 13, partition walls; 14, cooler bundles; 15, water separator; 16, water chamber covers; 17, shaft; 18, rotor blades; 19, impellers; 20, journal bearings; 21, position of thrust bearing (if fitted); 22, balance piston; 23, shaft seal. (Source: Sulzer-Burckhardt.)

Components such as impeller volute casings, guide vane units, bearings, and seals are taken from standard components of that series.

The lube oil console is incorporated into the compressor base frame. This way compressor, driver, and auxiliaries form one single lift unit that is easy to install and easy to maintain.

While the SEZ series incorporates a stepup gear, the SEE has been designed for direct drive concepts.

Technical performance data

- Volume flow rates up to 48,000 m^3/h (280,000 cfm)
- Compression ratios up to 3 depending on the gas analysis
- Discharge pressure up to 80 bar (1200 psi)
- Discharge temperature up to 3000°C (572°F)

Fields of application include

- Petrochemical
- Refineries

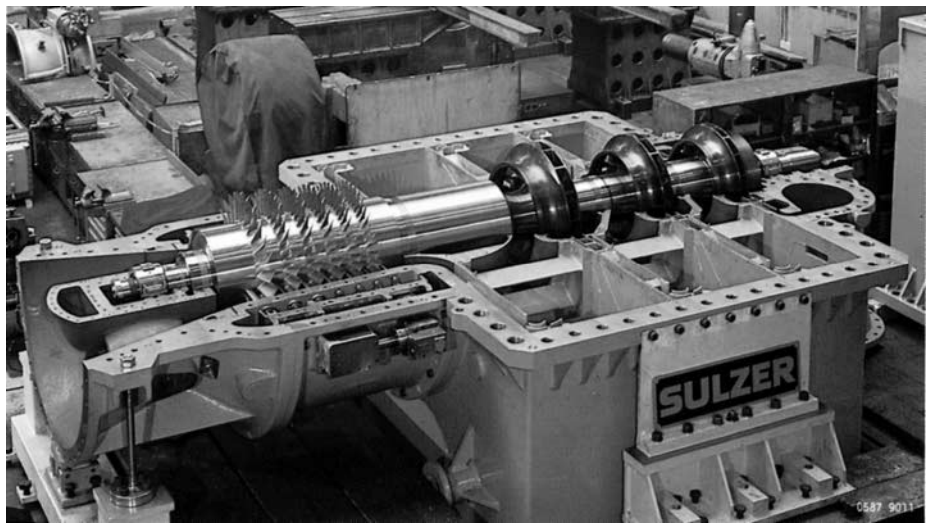


FIG. C-234 Functional and compact design with access to all vital components (ARI series). (Source: Sulzer-Burckhardt.)

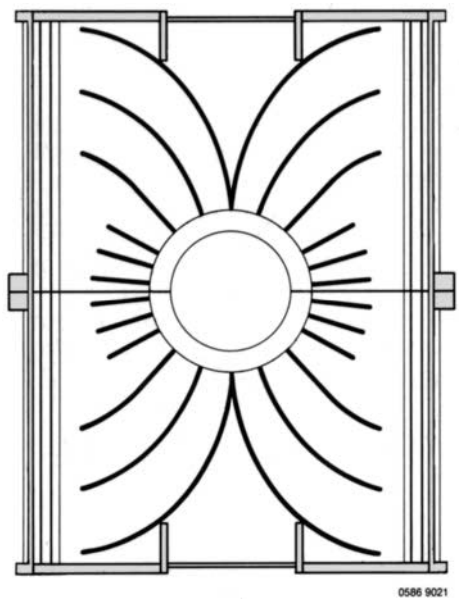


FIG. C-235 Flow passage between axial outlet and entry into the adjacent pair of intercoolers. (Source: Sulzer-Burckhardt.)

- Ammonia, fertilizers
- Metal production
- Air separation and more

See Figs. C-253 through C-259. Figure C-258 illustrates the internal components of a single-stage integrally geared compressor.

Integrally geared process gas radial turbines

An application for direct power recovery (see Figs. C-260 through C-262). The PRT series of integrally geared radial turbines has been developed on the basis of the integrally

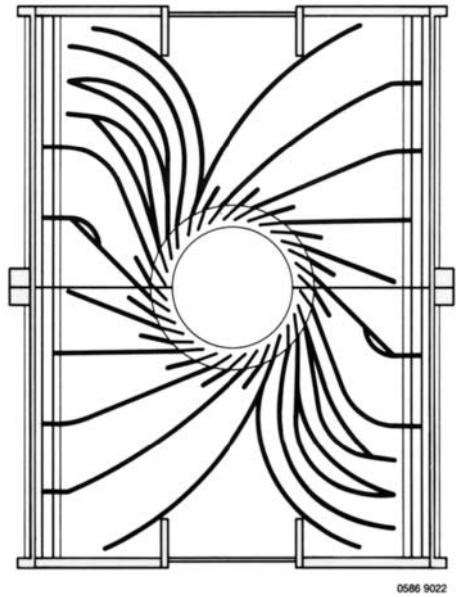


FIG. C-236 Flow passage from impeller outlet to the adjacent pair of intercoolers. (Source: Sulzer-Burckhardt.)

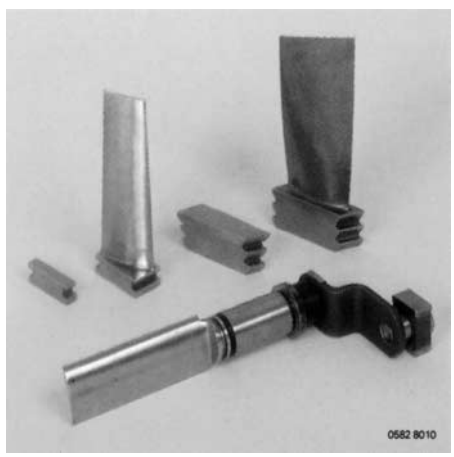


FIG. C-237 Adjustable stator blade, rotor blade, and fixed stator blade with intermediate piece. (Source: Sulzer-Burckhardt.)

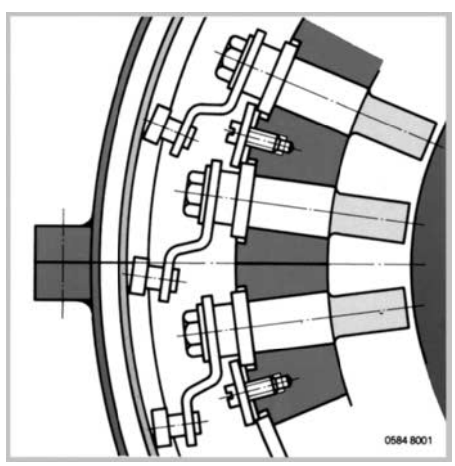


FIG. C-238 Fixation of adjustable stator blades. (Source: Sulzer-Burckhardt.)

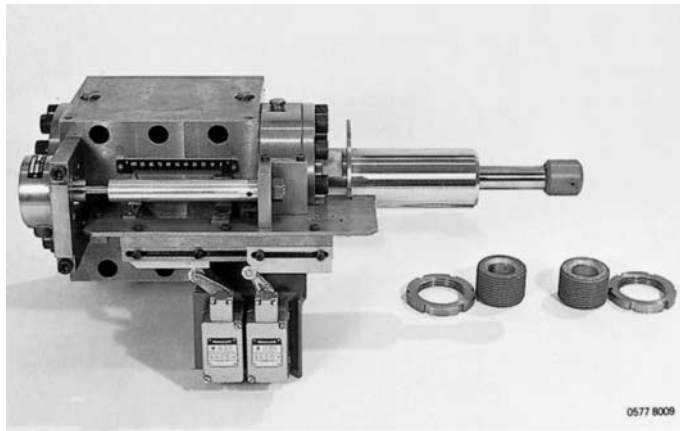


FIG. C-239 Hydraulic servomotor. (Source: Sulzer-Burckhardt.)

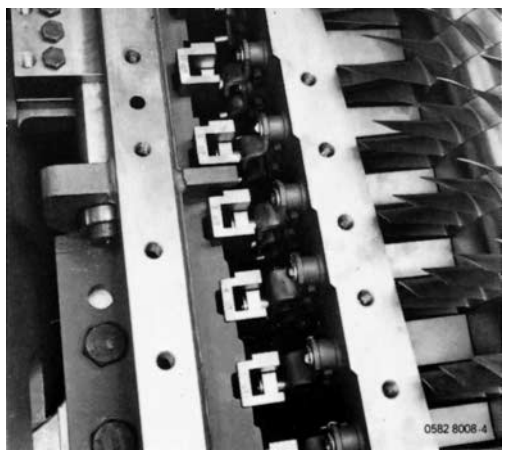


FIG. C-240 Stator blade adjusting mechanism. (Source: Sulzer-Burckhardt.)

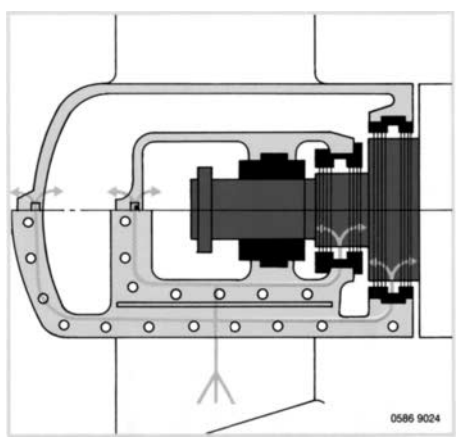


FIG. C-241 Double compressed-air sealing system on the suction side bearing housing. (Source: Sulzer-Burckhardt.)

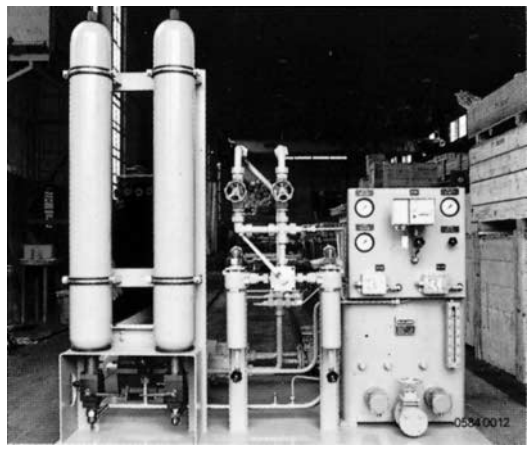


FIG. C-242 Power oil unit. (Source: Sulzer-Burckhardt.)

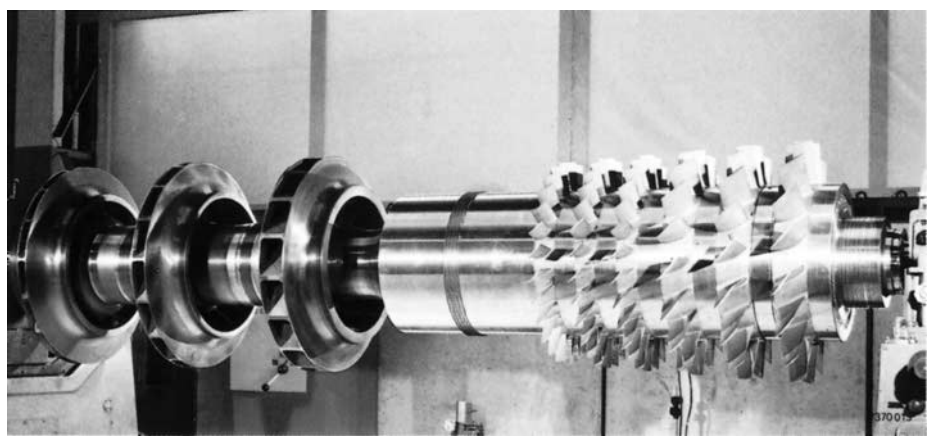


FIG. C-243 The single monobloc rotor running in only two journal bearings maintains high rotor stability and low vibration level. (Source: Sulzer-Burckhardt.)

geared centrifugal compressor concept using many of its components. Medium- to high-temperature process gas can be directly put to use driving a compressor or electric generator without the need for extra heat exchange or steam generation equipment.

Depending on the process conditions the PRT stages can either be mounted to the gearbox of a multiservice VK/HVK machine or designed as a separate unit.

For part-load operation the PRT can be fitted out with adjustable inlet nozzles.

Technical performance data (see Fig. C-263)

- Gas inlet temperatures up to 550°C (1020°F)
- Discharge volumes up to 350,000 m^3/h (200,000 cfm)
- Power recovery up to 35,000 kW (48,000 hp)

Fields of application include

- Power recovery
- Nitric acid and more

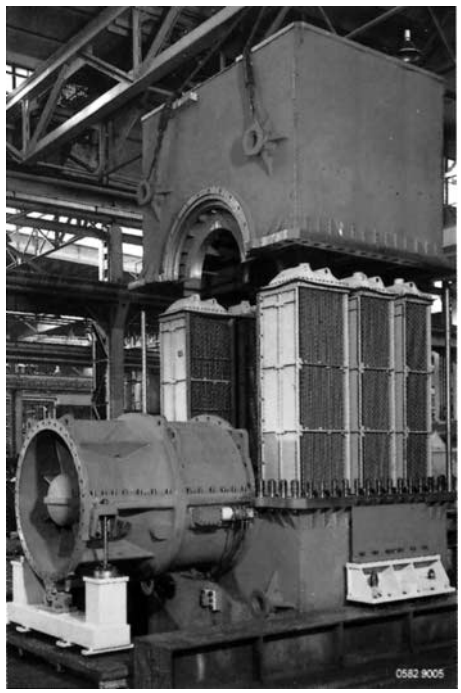


FIG. C-244 Inspection of the internal parts: Lift the centrifugal casing top half without disturbing coolers and adjacent casing parts. (Source: Sulzer-Burckhardt.)

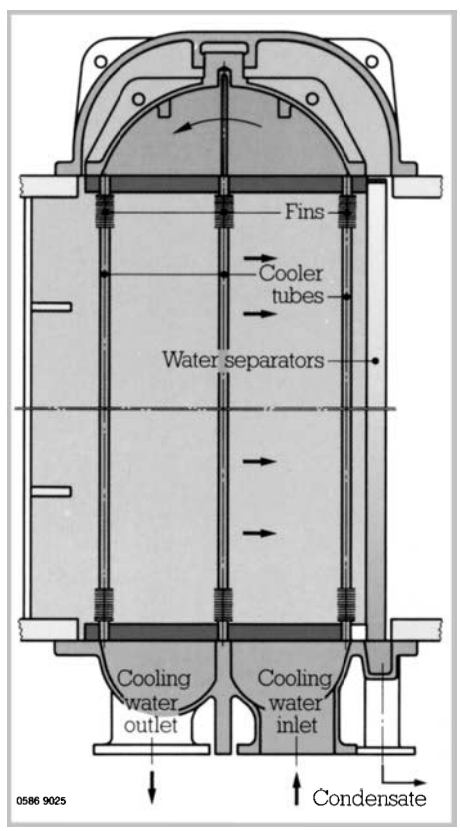


FIG. C-245 Two-pass version of the intercooler. Finned tubes and effective water separators contribute to the efficiency of the compressor. (Source: Sulzer-Burckhardt.)

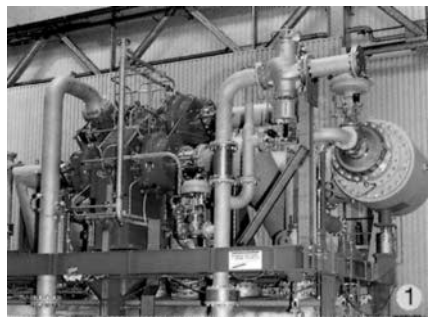


FIG. C-246 Six-stage oxygen integrally geared turbocompressor for a steel plant. (Source: Demag Delaval.)

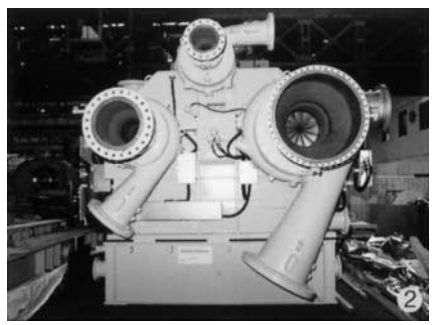


FIG. C-247 Dispatch preparation on a six-stage fuel gas compressor with dry gas seals for an LNG plant. (Source: Demag Delaval.)

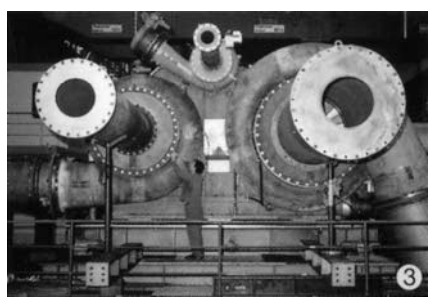


FIG. C-248 Test bed preparation of a five-stage integrally geared turbocompressor for an air separation unit. (Source: Demag Delaval.)

Compact integrally geared compressors. The CVK supplements the VK/HVK compressor range. Core units in four sizes and a wide range of pre-engineered options form the ideal basis to tailor a very cost-effective VK-gear–type compressor as a compact single-lift unit. (See Figs. C-264 and C-265.)

The application range. The CVK application range is subdivided into four sections, each of which is covered by one machine size. Depending on the required pressure ratio they can be designed with up to four stages.

Technical performance data (see Fig. C-266)

- Capacities from 10,000 to 32,000 m^2/h (6000 to 19,000 cfm)
- Suction pressure up to 1.1 bar (16 psi)
- Pressure ratio up to 20

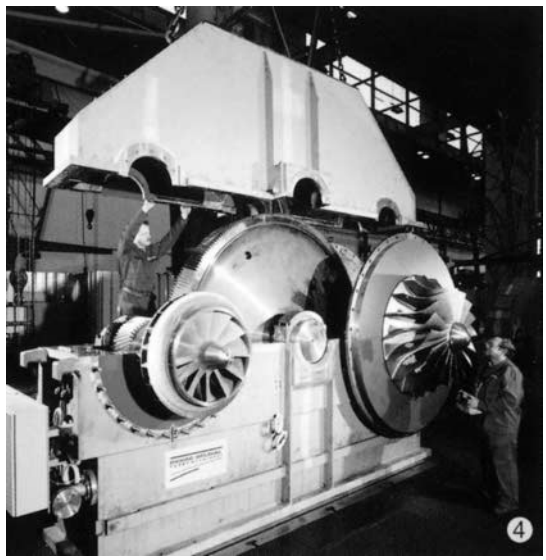


FIG. C-249 Assembly of an integrally geared turbocompressor for an integrated gasification combined cycle plant. (Source: Demag Delaval.)

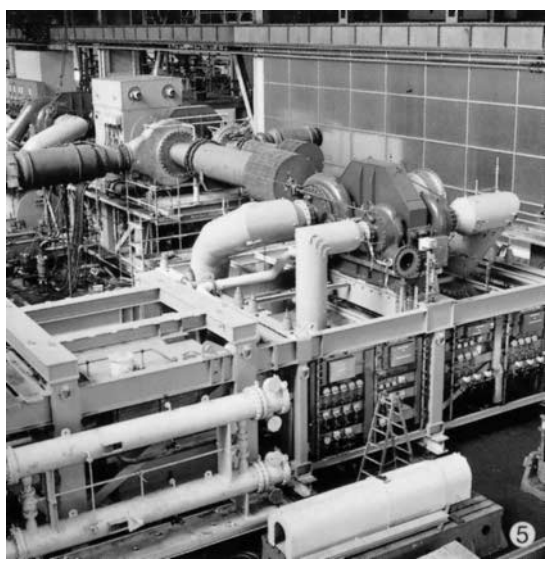


FIG. C-250 Packaging of two four-stage integrally geared turbocompressors with one expander for a PTA plant. (Source: Demag Delaval.)

Fields of application include

- Air separation
- Compressed air supply
- Hydrogen peroxide and more

Rotating Positive Displacement Compressors*

Rotary screw compressors and rotary piston blowers belong to the machinery group making up rotating positive-displacement compressors. Of these two machines,

* Source: Aerzener Maschinenfabrik, GmbH, Germany.

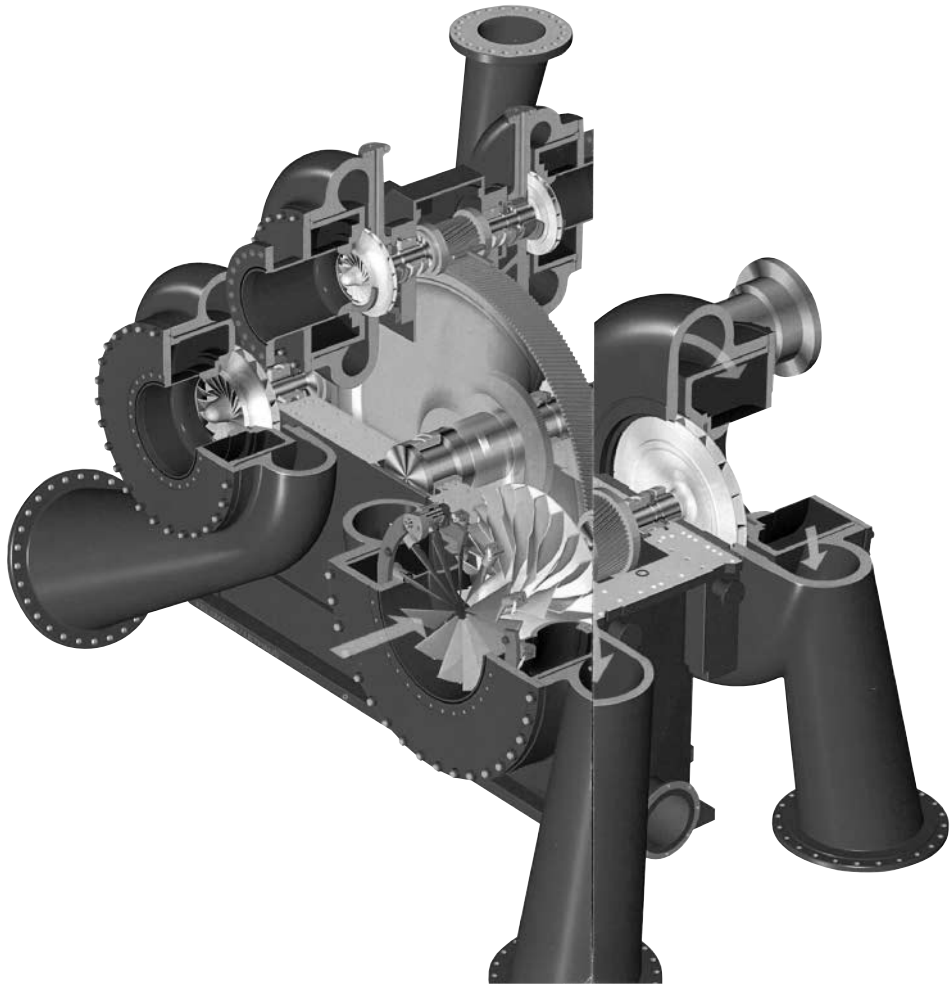


FIG. C-251 Internals of an integrally geared compressor. (Source: Demag Delaval.)

rotary screw compressors are primarily used in higher pressure air and process gas services. Rotary piston blowers are more typically used in lower pressure, high-volume applications. Both machines can be used as dry or wet fluid movers.

Rotating positive-displacement machines offer the same advantage as reciprocating positive-displacement equipment with regard to flow versus pressure relationships, i.e., nearly constant inlet flow volume under varying discharge pressure conditions. Also, positive-displacement machines do not have a surge limitation, which is to say, there is no minimum throughput requirement for these compressors.

The rotor tip speeds on rotary screw and rotary piston blowers are low; this allows for liquid injection and handling of contaminated gases. By design, the rotors are self-cleaning during operation, which is a significant advantage in dirty-gas services.

Rotary screw compressors are available in oil-free or oil-flooded construction. Fields of application for oil-free machines include all processes that cannot tolerate contamination of the compressed gas or where the lubricating oil would be contaminated by the gas. Oil-flooded machines can achieve slightly higher efficiencies and utilize the oil for cooling as well.

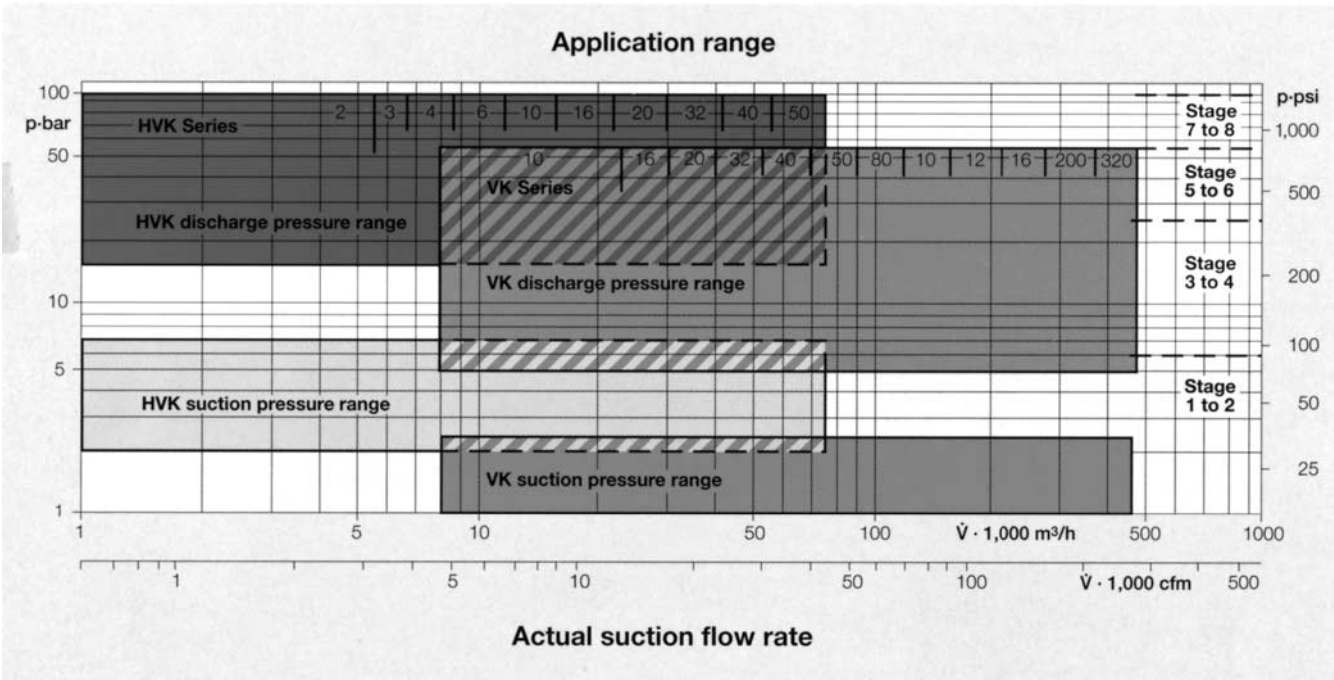


FIG. C-252 Operating envelopes for multistage integrally geared compressors. (Source: Demag Delaval.)

Properly designed rotary screw compressors are constructed with no metallic contact whatsoever inside the compression chambers, either between the rotors themselves or between these and the walls of the housing.

Although originally intended for air compression, rotary screw compressors are now compressing a large number of process gases in the petrochemical and related industries. These include air separation plants, industrial refrigeration plants, evaporation plants, mining, and metallurgical plants.

Most gases can be compressed: ammonia, argon, ethylene, acetylene, butadine, chlorine gas, hydrochloric gas, natural gas, flare gas, blast furnace gas, swamp gas, helium, lime-kiln gas, coking-plant gas, carbon monoxide gas, all hydrocarbon combinations, town gas, air/methane gas, propane, propylene, flue gas, crude gas, sulfur dioxide, oxide of nitrogen, nitrogen, styrene gas, vinyl chloride gas, and hydrogen gas can be found on the reference tabulations of experienced manufacturers.

Application limits for rotary screw compressors. Application limits for rotary screw compressors are given by the pressure and temperature ranges and by the maximum allowable speed of the machines.

Oil-free rotary screw compressors can be mechanically loaded with pressure differences up to 12 bar, and oil-flooded compressors up to 20 bar. Higher pressure differences are possible in special cases.

The maximum allowable compression ratio for one screw compressor stage that will not cause the final compression temperature to rise above the permitted value of 250°C will to a very large extent depend on the specific heat ratio c_p/c_v of the gas to be compressed. For example, where the specific heat ratio c_p/c_v equals 1.4, the maximum compression ratio would be approximately 4.5, and where the specific heat ratio c_p/c_v equals 1.2, the maximum compression ratio would be approximately 10 for one oil-free screw compressor stage.

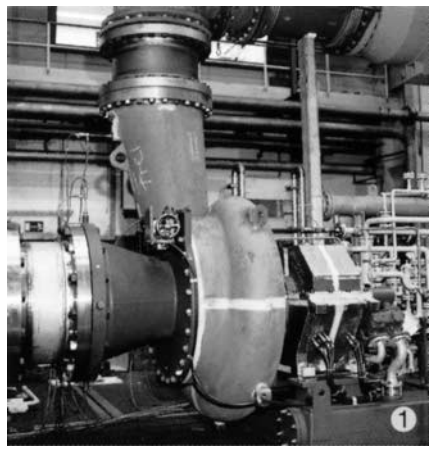


FIG. C-253 For a chemical plant during testing. (Source: Demag Delaval.)

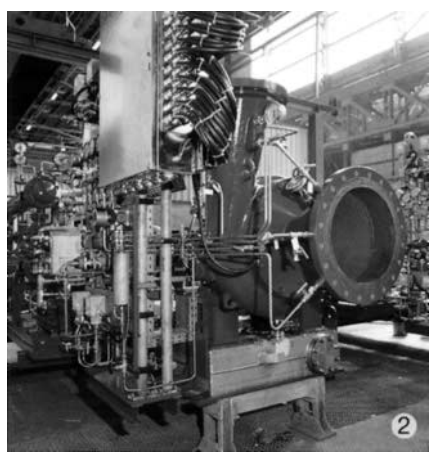


FIG. C-254 For nitrogen service in an ammonia plant. (Source: Demag Delaval.)

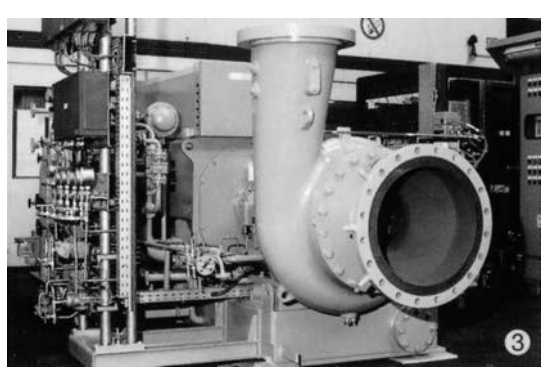


FIG. C-255 In preparation for shipment to a refinery. (Source: Demag Delaval.)

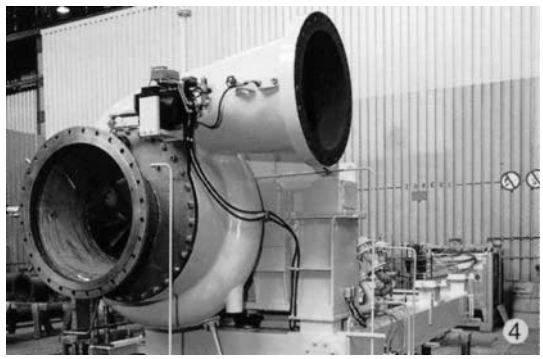


FIG. C-256 For a steel plant. (Source: Demag Delaval.)

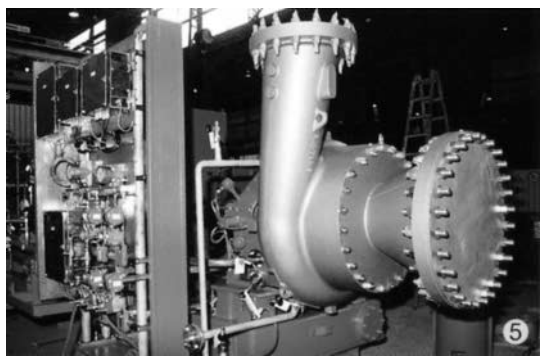


FIG. C-257 For oxygen service in an air separation unit. (Source: Demag Delaval.)

Multistage (multicasing) arrangements are not uncommon and can result in pressure ranges from approximately 0.1 bar absolute to 40 bar. Interstage cooling is used in many of these applications.

Depending on compressor size, speeds from 2000 to 20,000 rpm can be encountered. The limiting factor is typically the circumferential speed of the male rotor, which typically ranges from 40 to approximately 120 m/sec, and up to a maximum of 150 m/sec for very light gases.

Flow volumes up to 60,000 m^3/h can be accommodated in these compressors.

Reference and Additional Reading

1. Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Rotary piston blowers*

Rotary piston, or lobe-type, blowers derive from the Roots compressor principle and have been built since 1864. They are used in a large variety of process plant applications, including pneumatic conveying of bulk materials, pressurized aeration of water at treatment plants, creation of vacuum, and gas movement in the petrochemical, pharmaceutical, and metallurgical industries. They range in size from fractional horsepower to literally hundreds of kilowatts.

*Source: Aerzener Maschinenfabrik, GmbH, Germany.

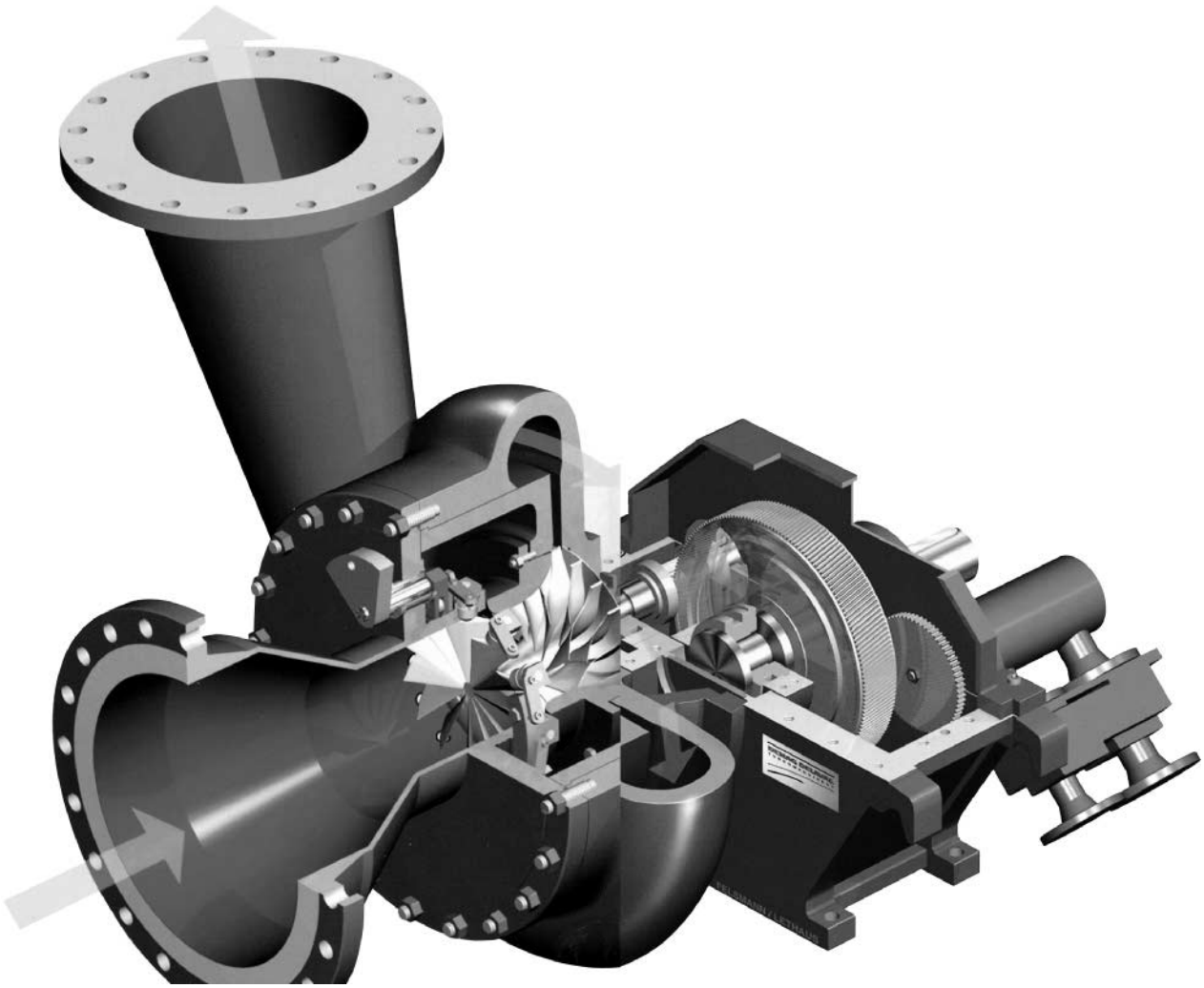


FIG. C-258 Internals of a single stage integrally geared compressor. (Source: Demag Delaval.)

Condensers*

The task of condensers is to condense the working fluid in a power plant cycle. The turbine exhaust steam condenses on the outside of the cooling tubes and the condensation heat is conducted to the cooling water inside the tubes. The condensate is collected and routed via the feedwater heaters and the feedwater line to the boiler where it again evaporates and is employed as operating steam to drive the turbine.

Via the turbine back pressure the performance of the condenser decisively influences the efficiency of the entire plant and with it the generator output (Fig. C-267).

In addition, the condenser represents a connecting link to the environment via the cooling water. This means that a condenser must fulfill not only the requirements of the power plant but also of environmental legislation.

---

*Source: Alstom. Adapted with permission.

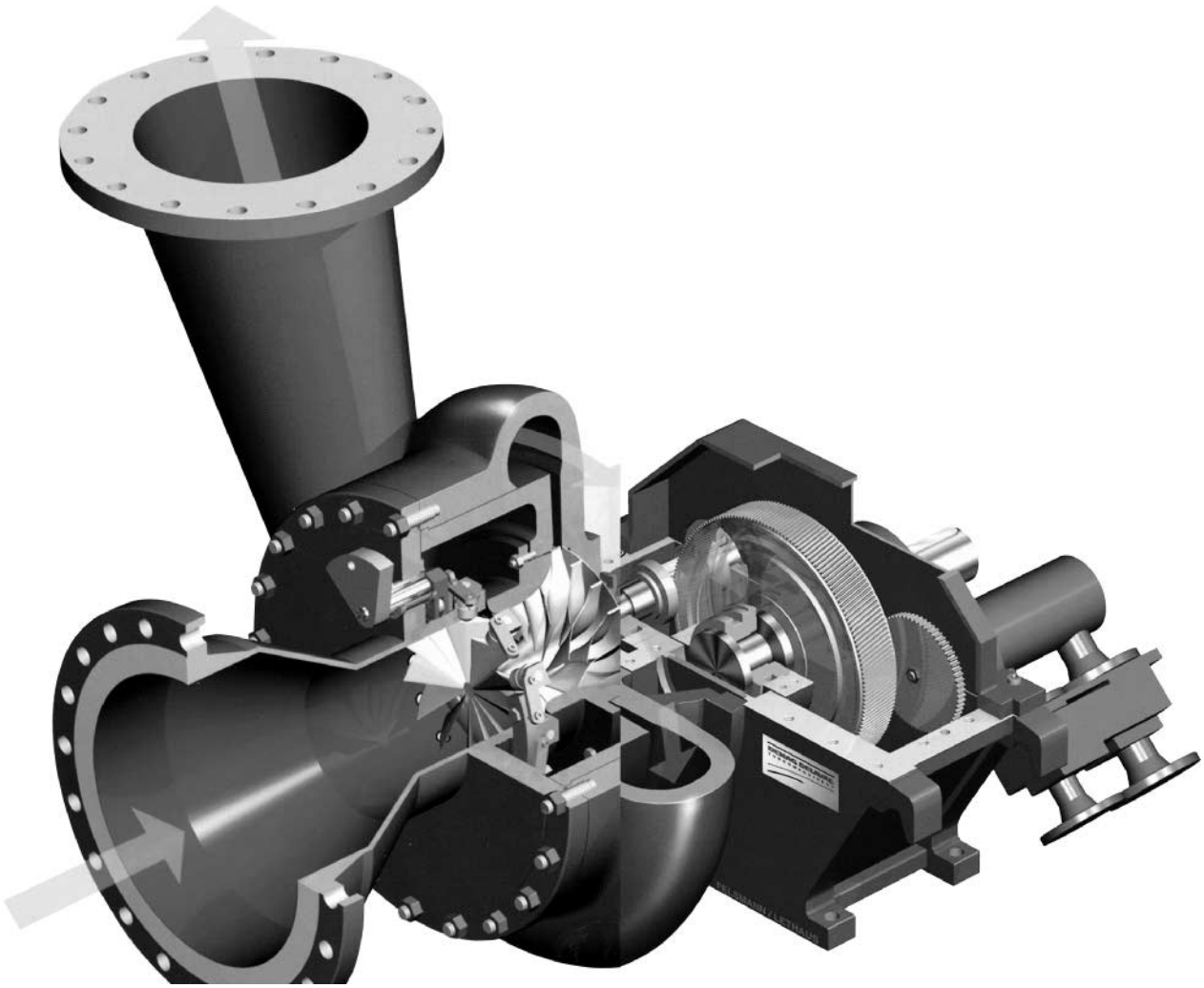


FIG. C-258 Internals of a single stage integrally geared compressor. (Source: Demag Delaval.)

Condensers*

The task of condensers is to condense the working fluid in a power plant cycle. The turbine exhaust steam condenses on the outside of the cooling tubes and the condensation heat is conducted to the cooling water inside the tubes. The condensate is collected and routed via the feedwater heaters and the feedwater line to the boiler where it again evaporates and is employed as operating steam to drive the turbine.

Via the turbine back pressure the performance of the condenser decisively influences the efficiency of the entire plant and with it the generator output (Fig. C-267).

In addition, the condenser represents a connecting link to the environment via the cooling water. This means that a condenser must fulfill not only the requirements of the power plant but also of environmental legislation.

---

*Source: Alstom. Adapted with permission.

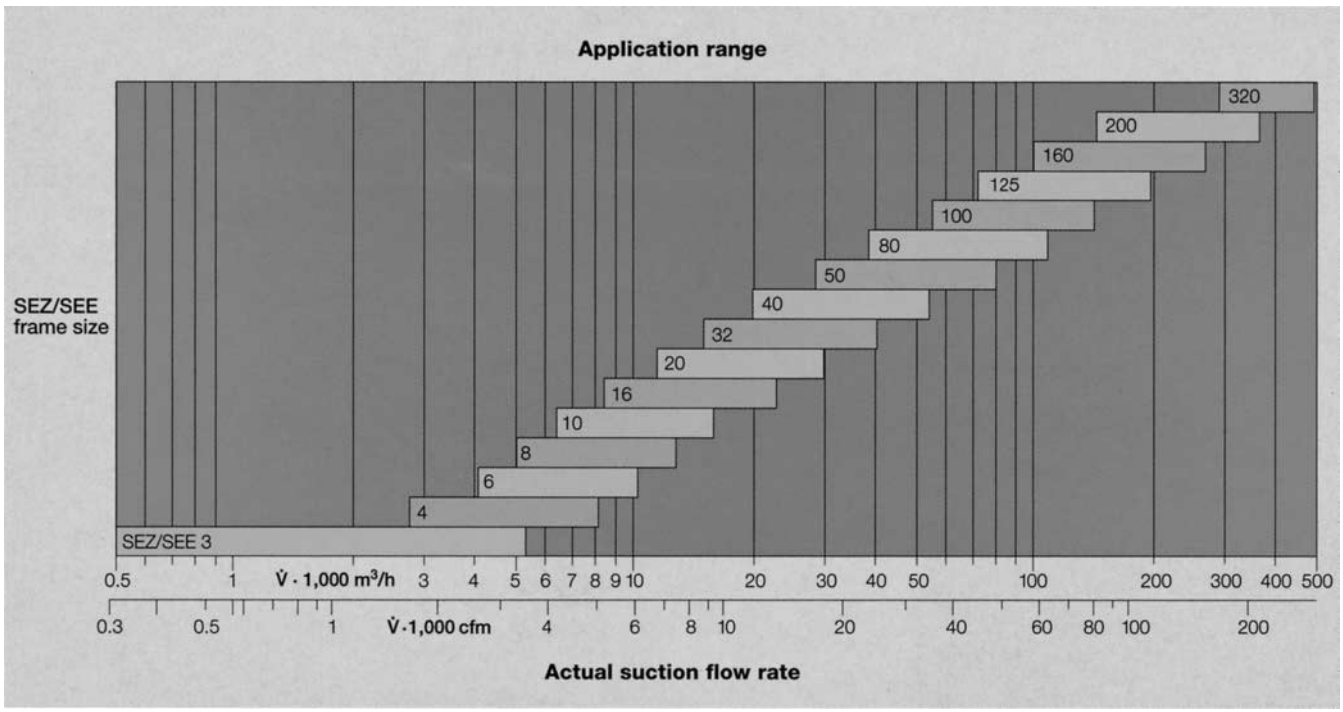


FIG. C-259 Operating envelopes for single-stage integrally geared compressors. (Source: Demag Delaval.)

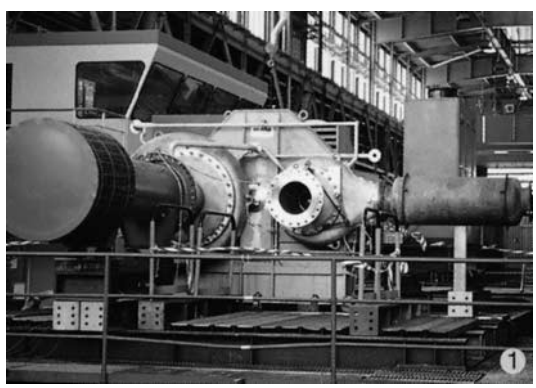


FIG. C-260 Test run preparation of a two-stage PRT model. (Source: Demag Delaval.)

Manufacturers of steam turbines and condensers ensure that an optimum balance is achieved between these two important components.

For the purposes of illustration the Alstom condenser model, designation "CB" will be discussed. The condenser of the CB-type series is for power plant units of up to approximately 200 MW_{el}.

Design Layout of a CB Condenser (see Fig. C-268)

Condenser dome

The turbine exhaust steam flows through the condenser neck and the steam dome to the tube bundles arranged symmetrically in relation to the longitudinal axis. The

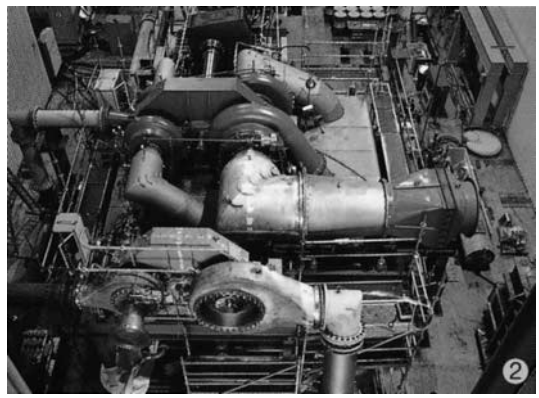


FIG. C-261 String test of a compressor-expander package. (Source: Demag Delaval.)

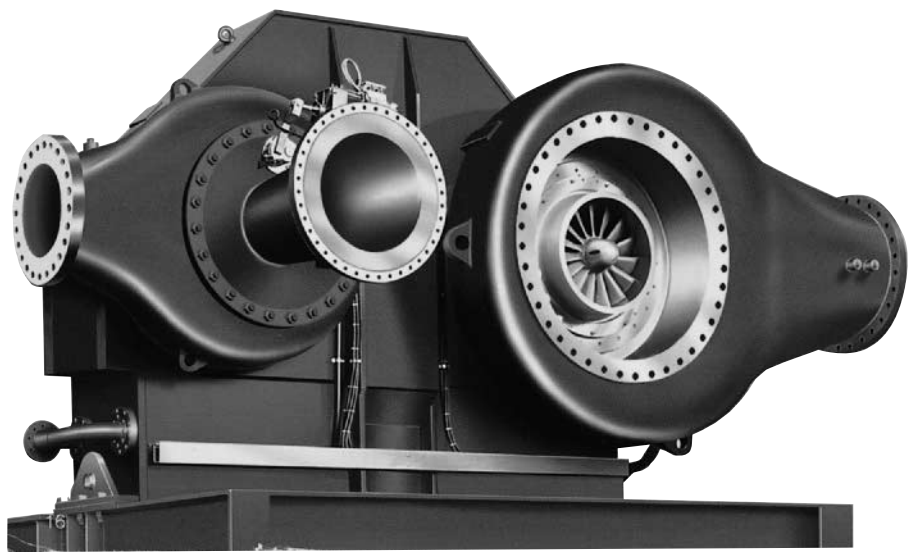


FIG. C-262 Internals of an integrally geared process gas radial turbine. (Source: Demag Delaval.)

steam dome is configured such that uniform admission of the steam to the bundle tubes is ensured.

Tube bundle

The condenser tubes are divided into two tube bundles into which steam can flow from all sides. The bundles are designed to obtain a low condenser pressure along with a good deaeration and no subcooling of the condensate.

Air cooler

As the condenser pressure is below atmospheric pressure, some leakage air continuously enters the turbine and the condenser. It must be removed. The air is extracted at the lowest pressure points, the air coolers. The CB condenser has two air coolers, one located in the center of each bundle. The air is extracted uniformly over the entire length of the condenser.

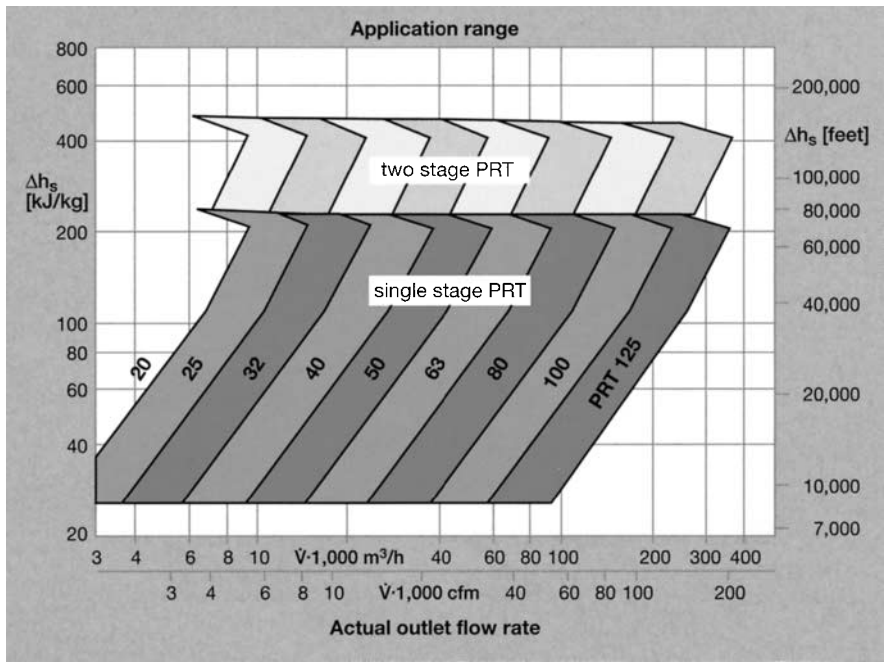


FIG. C-263 Operating envelopes for integrally geared process gas radial turbines. (Source: Demag Delaval.)

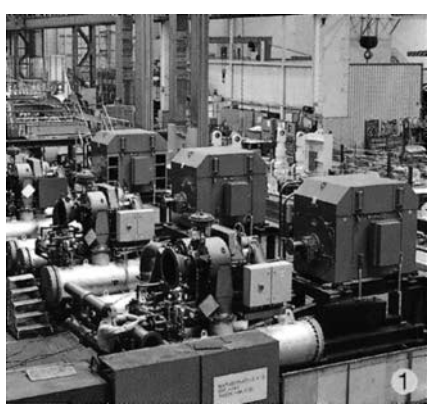


FIG. C-264 Three compact integrally geared turbocompressors type CVK for plant air supply in a chemical works. (Source: Demag Delaval.)

Condenser shell

The shell is welded to the steam dome, the hotwell, and the tubesheet and ensures efficient steam guidance. The cylindrical shell is self-supporting and can absorb the vacuum forces. For this reason the support plates are connected to the shell without bearing load.

Support plates

In order to prevent destructive tube vibrations, generated by the velocity of the steam, the tubes are mounted in support plates. The support plate spacings are carefully calculated to withstand extreme operating conditions.

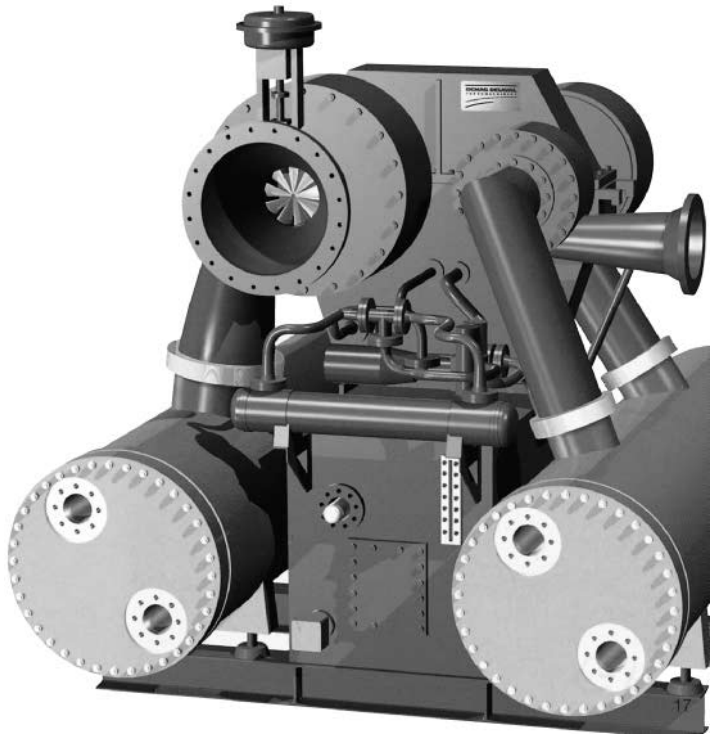


FIG. C-265 Internals of a compact integrally geared compressor. (Source: Demag Delaval.)

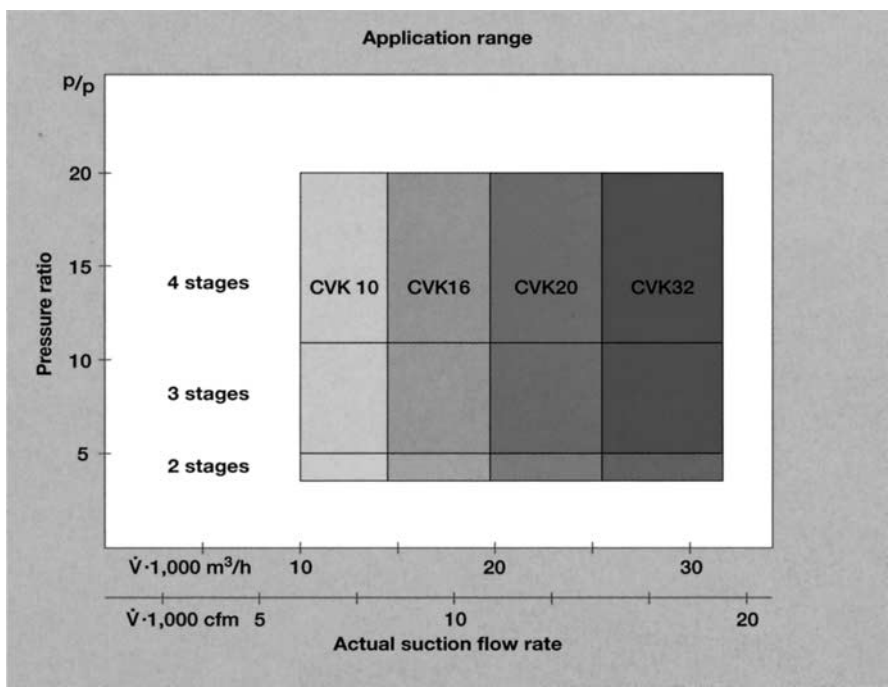


FIG. C-266 Operating envelopes for compact integrally geared compressors. (Source: Demag Delaval.)

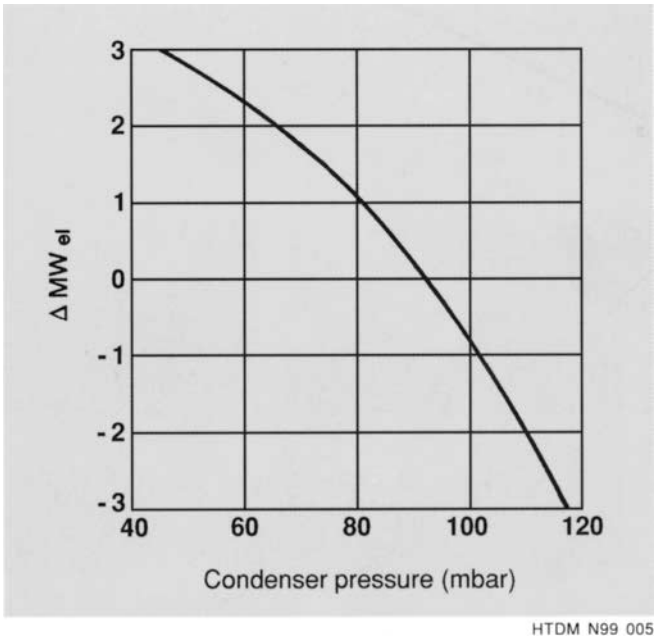


FIG. C-267 Influence of turbine back pressure on the electrical output of a typical 150-MW plant. (Source: Alstom.)

Hotwell

The condensate flows into the hotwell and is returned to the circuit by the main condensate pumps. The pump flow rate is controlled by the condensate level in the hotwell.

Waterboxes

The waterboxes ensure uniform distribution of the cooling water to the condenser tubes. They are designed for low hydraulic pressure losses. If necessary, they are fitted with special internals to ensure optimal distribution of the sponge cleaning balls to all tubes. A single or two-pass water system is provided. In every case it is possible to shut down one condenser half with the plant still in operation to enable maintenance work to be carried out. Manholes ensure ready access.

The waterboxes are welded steel structures with a welded connection to the tubesheets. If the end user requires it, one of the two waterbox covers can be bolted to withdraw or replace tubes as necessary.

To prevent corrosion the waterboxes are coated on the inside with epoxy coating or are rubber lined.

Flashbox

During startup and load transients the high-energy internal turbine drains are routed via a flashbox into the condenser. In the flashbox flashing and phase separation into steam and condensate takes place. Injection water lowers the temperatures of high-temperature inlets before they enter the condenser. This protects the condenser and the steam turbine from the hot drains.

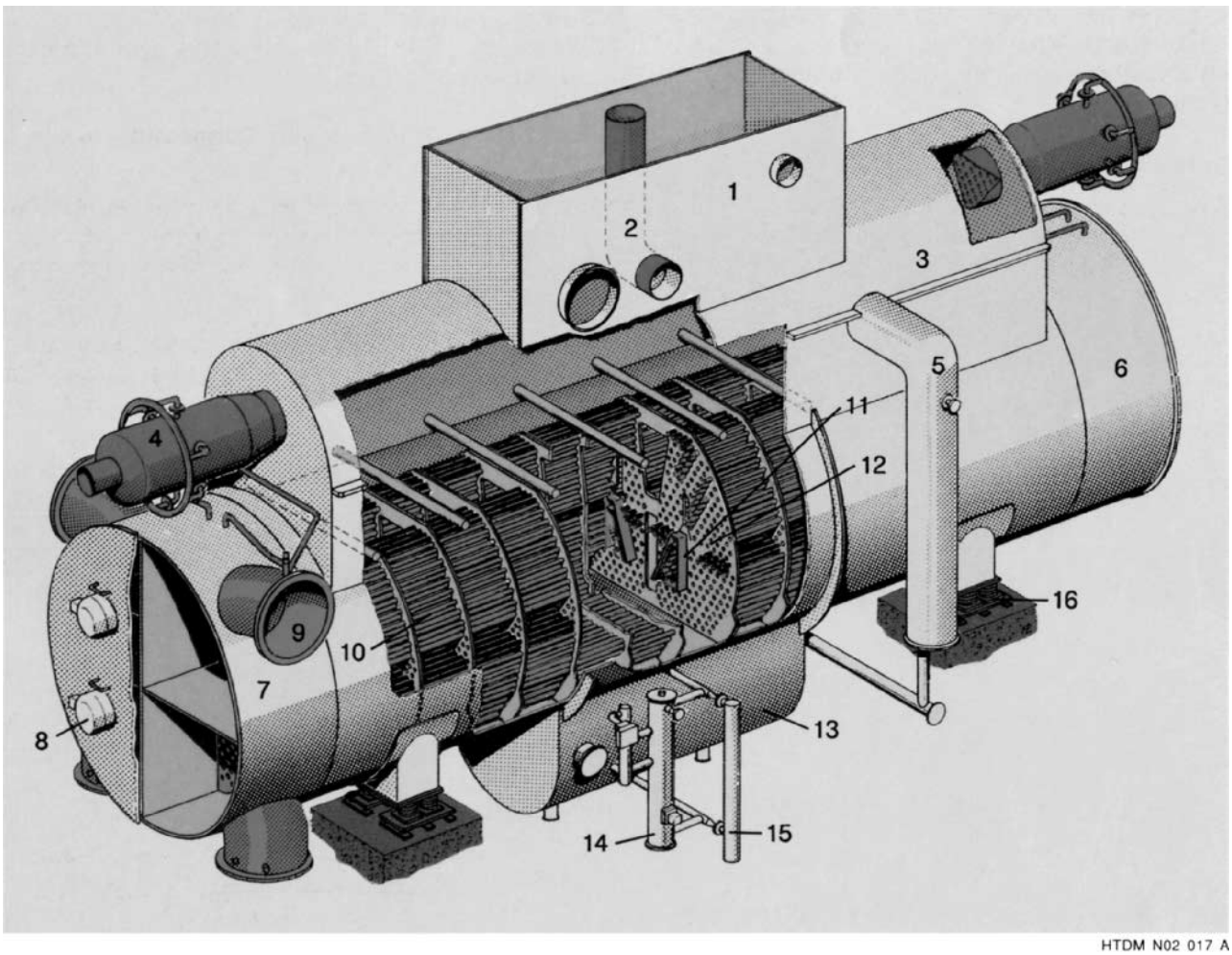


FIG. C-268 Design layout of two-pass CB condenser. 1, condenser neck; 2, extraction line; 3, steam dome; 4, steam dumping device; 5, flashbox; 6, reversing waterbox (for two-pass type); 7, inlet/outlet waterbox (for two-pass type); 8, manhole; 9, cooling waterline (outlet); 10, tube bundle; 11, air cooler; 12, support plate; 13, hotwell; 14, standpipe; 15, level indicator; 16, spring assembly. (Source: Alstom.)

Foundation mounting and connection to the steam turbine

Normally the condenser is welded to the turbine and spring supported on the foundation.

As an alternative the condenser can also be firmly anchored to the ground. In this case an expansion joint is provided between the condenser and the turbine.

Features of This Design

This condenser is a robust piece of equipment that offers the following features:

High thermal performance

Due to an optimized tube arrangement, high heat transfer coefficients (k-values) are achieved at full load and partial loads, thus improving the heat rate of the power plant. The good thermal characteristics mean that considerably less condenser surface is required than for conventional condenser concepts.

Efficient space utilization

The apparatus is compact and saves space in the turbine building. The tube arrangement (*see* Bundle Concept below) yields a large heat exchanger surface with comparatively smaller overall dimensions of the condenser.

Self-supporting structure

As condensers operate below atmospheric pressure, relatively high vacuum forces must be absorbed. Because of its cylindrical shell the structure is self-supporting. Expensive internal supports can be dispensed with.

Economic manufacture and maintenance

The dimensions of the type series are standardized. All tube pattern coordinates are prepared for manufacture on numerically controlled machines.

Simple transport and assembly

The condenser can be delivered to the plant ready-assembled or in separate modules. It is supplied complete with waterboxes, tubed, and tested. Only a small number of assembly welds remain to be done on site, ensuring minimum assembly time. See Fig. C-269.

Low oxygen content in condensate

With deaeration of the condensate, extremely low O_2 concentrations are achieved in the condensate. This provides corrosion protection.

Makeup water

The makeup water is sprayed over the bundles and deaerated.

No condensate subcooling

In many condenser types the condensate is considerably cooler than the saturation temperature to the condenser pressure. This subcooling has a negative effect on the plant efficiency because the following feedwater heater requires more steam than would otherwise be the case without subcooling. The CB concept uses “negative subcooling.”

Bundle Concept (see Fig. C-270)

The tube pattern of the CB condenser is configured such that it shows regenerative characteristics similar to those of the model CM condenser of this information source for large power plants.

Regeneration means that the condensate that drips out at the bottom of the tube bundles comes into close contact with the turbine exhaust steam. In this way the condensate is deaerated and heated up before it falls onto the condenser bottom and the hotwell.

With conventional condensers having a round cross-section an attempt is made to provide the greatest possible heat exchanger surface in the given space by employing many tubes. At best a middle lane is left free in order to supply the bottom tubes with steam as well. These tube configurations do not allow any pressure regeneration, neither do they permit any thermohydraulic optimum location of the air cooler for all load conditions. The result of this is the formation

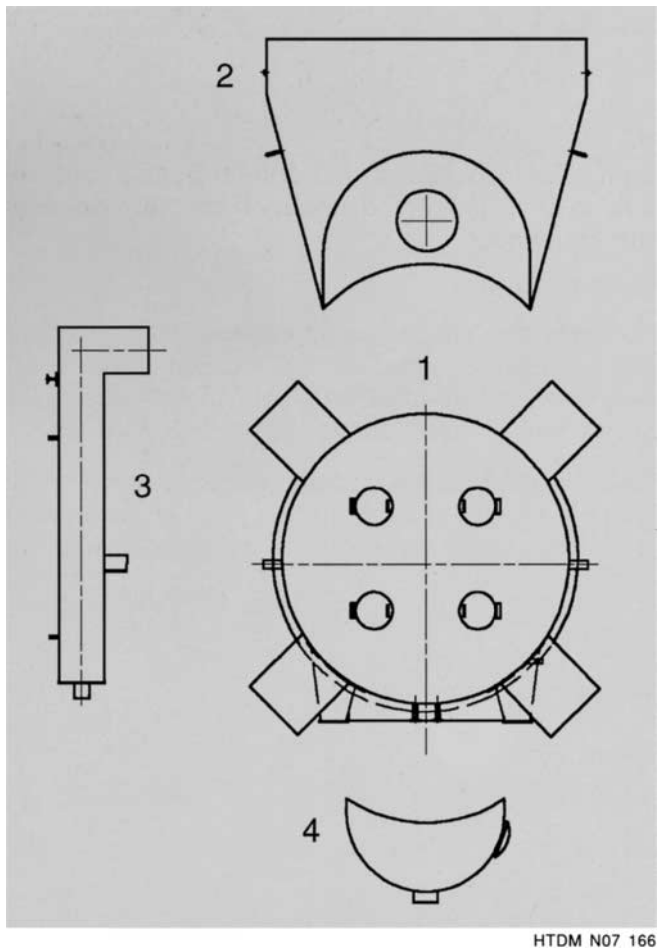


FIG. C-269 Transport packaging arrangements for the CB condenser. The actual condenser (1), the neck (2), the flashbox (3), and the hotwell (4) can be supplied separately to the erection site. This considerably reduces the welding work on site. In every case the condenser is already tubed and pressure tested. (Source: Alstom.)

of air blanketing and consequential impairment of the condenser pressure and other performance parameters.

In contrast to this, steam is admitted to the CB bundle not only from the center lane but from all sides. As a result the local steam velocities and the steamside pressure losses are lower.

As the steam can flow into the CB bundle from all sides, the steam to the bundle inside need only flow through a few tube rows. This keeps the pressure losses on the way through the bundle to the air cooler very low. In consequence the steam temperature decreases only slightly and the driving temperature gradient between steam and cooling water remains high. This ensures a high condensation rate.

The calculation of the multidimensional flow conditions in the CB condenser is performed with the aid of computer programs. In this way the bundle contours can be determined and optimized so that there is practically uniform distribution of the steam from the middle lane and from the side lane.

The flows from the side lane and the middle lane meet below the bundle. Here the velocity is zero, i.e., the dynamic pressure content is again converted to static pressure, causing a temperature rise. This enables optimal regeneration.

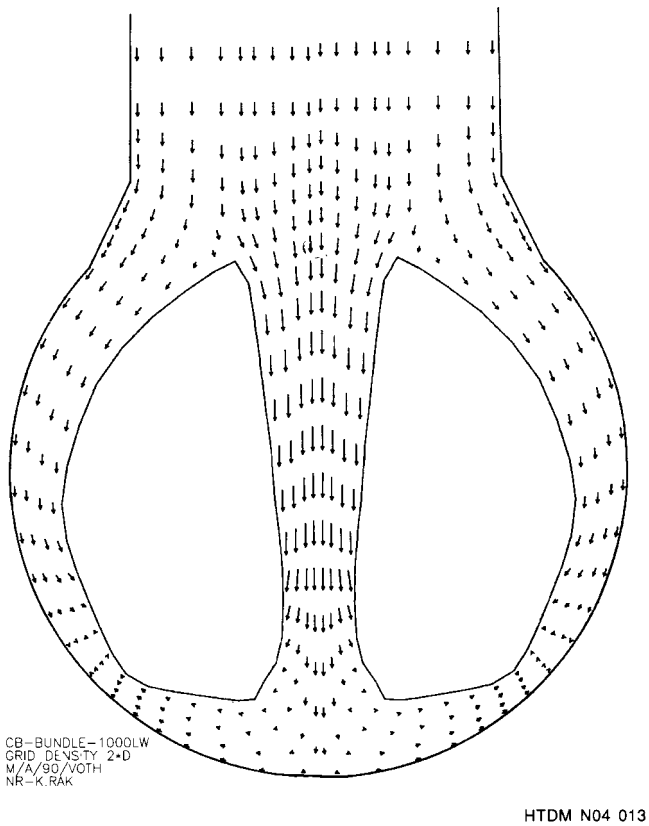


FIG. C-270 Distribution of steam velocity around the tube bundle. (Source: Alstom.)

In design, close attention is also focused on the position of the air coolers. They are arranged such that the location of the air intake coincides with the zone of minimum pressure in the bundle.

In order to improve utilization of the available cooling water the condenser can also be provided as a two-pass type. In this case the cooling water flows through the bottom bundle half first and, in warmer state, is routed to the top bundle half via the reversing waterbox (Fig. C-268). This improves the admission of steam to the bottom bundle half, resulting in a uniform distribution of the load over the condenser.

Thermodynamic Layout (see Fig. C-271)

The layout of a condenser and the calculation of the various types of operation is done by computer. The bundle data of the entire type series are computer-stored. The computer program employs the newest correlations for heat transfer and pressure drop. The exact properties of water (including sea water) and steam as a function of pressure and temperature are integrated in the program, enabling highly reliable calculations to be made. Each load is individually calculated.

Steamside pressure loss

Special attention is paid to the steamside pressure loss. When the steam flows through the bundle, the pressure drops. As the steam in this case is saturated the temperature decreases also. Tubes in the middle of the bundle are therefore exposed

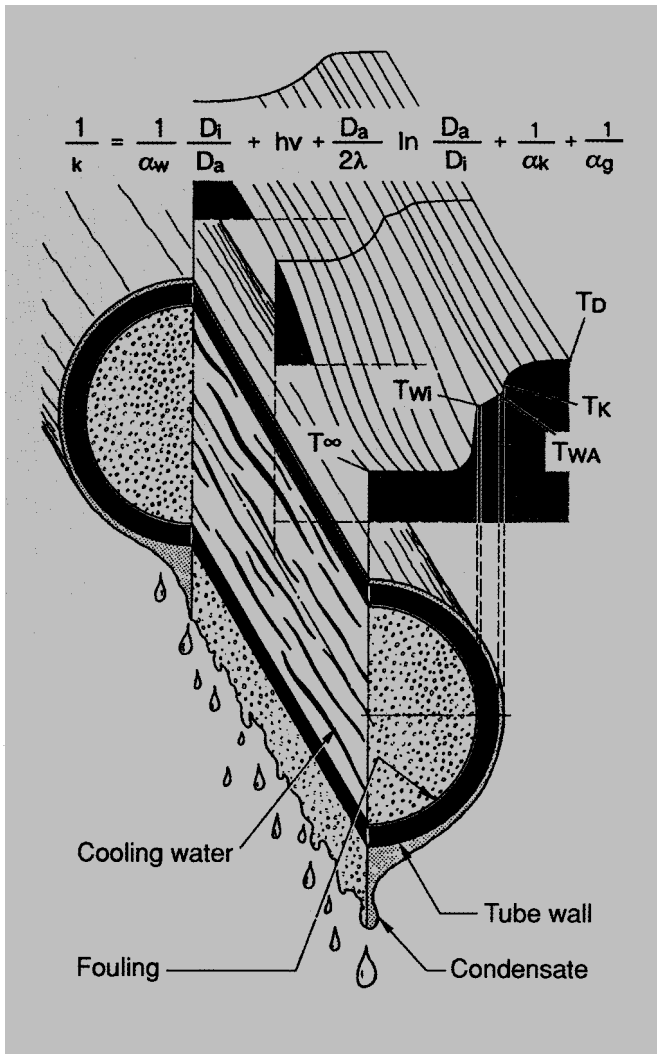


FIG. C-271 Schematical representation of heat transfer between cooling water and steam on a condenser tube. (Source: Alstom.)

to a lower condenser temperature than the peripheral tubes and condense correspondingly less. The steamside pressure loss depends on the steam velocity, the ligament width, and the condensate load. The comprehensive multidimensional computer model enables all parameters to be taken into account.

Tube fouling

The surfaces in condensers are not bare metal but are always covered with a certain coat of dirt and corrosion residue. This layer influences the heat transfer and must be taken into consideration. The fouling is a constant h_v (m^2K/kW) for a given tube material, which is not dependent on the condenser load or the operating mode. Therefore this definition of fouling differs from the widely used so-called "cleanliness factor" c_v (%), because in the latter case according to its definition the "fouling" changes with the load.

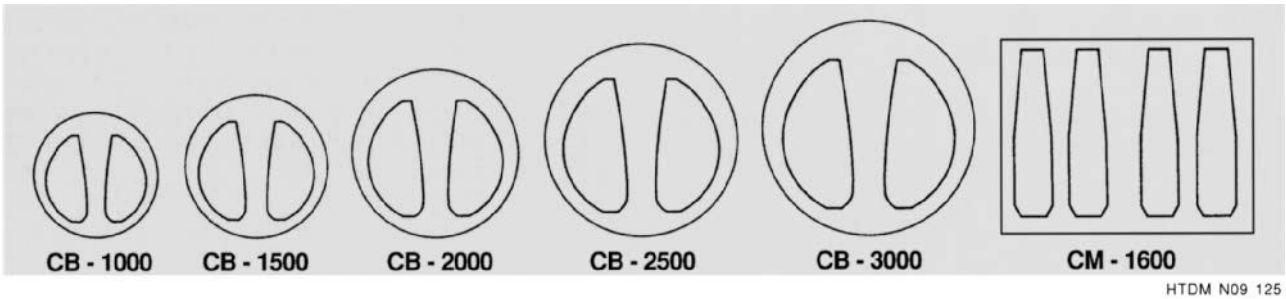


FIG. C-272 Schematic representation of the CB condenser-type series up to the CM series. (Source: Alstom.)

For condenser calculation, the OEM uses standard fouling coefficient h_v for every tube material based on experience.

Layout philosophy

Due to the routing of the steam flow, CB condensers achieve better heat transfer values (k-values) than some other designs. In order to obtain the required condenser pressure up to 20 percent less surface area is required in comparison with some older designs.

Cold end optimization

This manufacturer establishes the optimum combination condenser/turbine in each case using the following parameters:

- Cooling water temperature
- Cooling water flow rate
- Power consumption of the cooling water pump
- Space conditions

Type Series (see Figs. C-272 and C-273)

In the interests of standardization, a type series has been developed for the CB condenser that covers the entire power range of industrial turbines and small steam and cogeneration plants. This series provides an appropriate link to the large-scale condensers made by this OEM (type series CM).

Together with the option of varying the tube length it is possible to provide the optimum condenser for every requirement and plant size.

Selection of Material

In most cases the condenser is manufactured from material according to DIN standards. If required by the customer, materials can also be used in compliance with other standards, such as ASTM.

Tube material

The tube material on the steam side must satisfy the requirements of the water/steam circuit. On the water side it must meet the cooling water requirements.

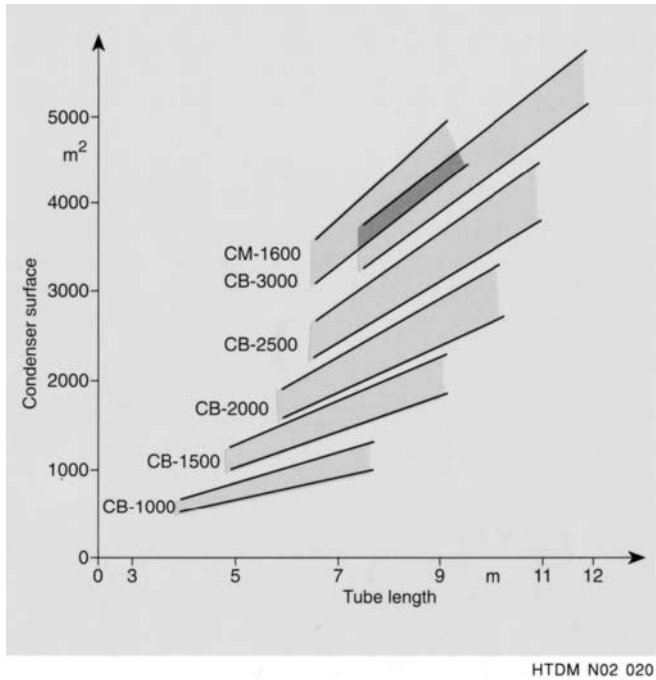


FIG. C-273 Type series of CB condenser. (Source: Alstom.)

Due to the manifold requirements the selection of the tube material is of great importance.

The basis of every selection is a sample of the cooling water, which is analyzed by specialists. In cooperation with the end user the appropriate material is then selected. The most important criteria are:

- Corrosion resistance to cooling water
- Sand content of the cooling water
- Cooling water velocity
- Thermal conductivity of the material
- Chemistry of the steam circuit
- Resistance to droplet erosion

The following materials are mainly employed:

Brass. If the quality of the cooling water is good (river water, freshwater lakes) admiralty brass or aluminum brass is a well-proven material, an important feature being high thermal conductivity.

Copper nickel alloys. If the quality of the water is poor, such as met with in ports and large rivers, CuNi alloys are preferred because they are more resistant than brass. In cooling tower operation also these alloys are of advantage as the cooling tower water is usually highly concentrated and therefore aggressive.

Stainless steels. For special requirements, such as for brackish water or sea water, high-grade steels are suitable. These have the advantage that much higher water

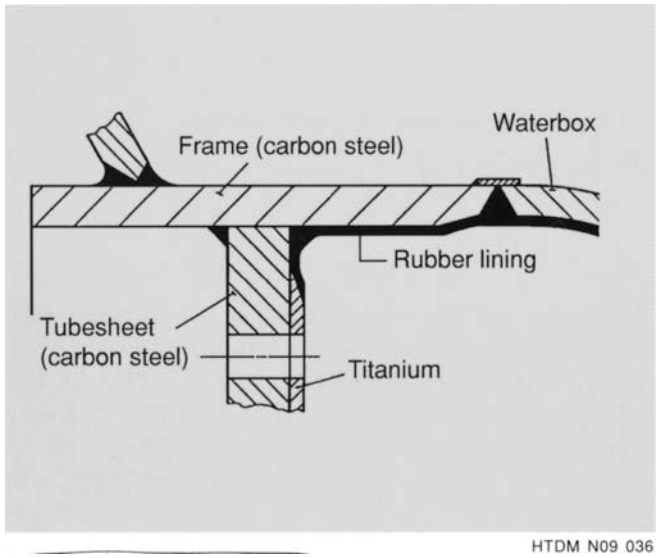


FIG. C-274 Cladded tubesheet, welded to the condenser housing. (Source: Alstom.)

velocities are admissible; in the case of nonferrous metals the velocity is limited by waterside erosion.

Titanium. This material fulfills practically all requirements. It is extremely resistant to corrosion and allows high water velocities as well as offering very good resistance to steamside droplet erosion. The price and the relatively low thermal conductivity can be compensated for, to some extent, by providing thinner tube wall thicknesses.

Tubesheet

In general, tubesheets are made of carbon steel with a stainless steel or titanium cladding on the cooling water side. The tubes are roller expanded into the tubesheet. Upon request, the tubes can also be welded into the tubesheet. See Figs. C-274 and C-275.

The tubesheets are welded to the condenser shell, thus ensuring reliable tightness.

The waterboxes are also welded to the tubesheet. If required, a flange connection can also be provided.

Venting

Venting the steam shell

The steam shell of a condenser is under vacuum. Careful manufacture and the use of high-grade sealing materials help to reduce the amount of air inleakage to a minimum but it can never be completely eliminated. The steam shell of a condenser must therefore be permanently vented.

For evacuation purposes, this OEM uses water-jet ejectors or steam-jet ejectors and, in special cases, also water ring pumps. The layout of the suction units employs concepts in compliance with the German VGB (technical association of large power

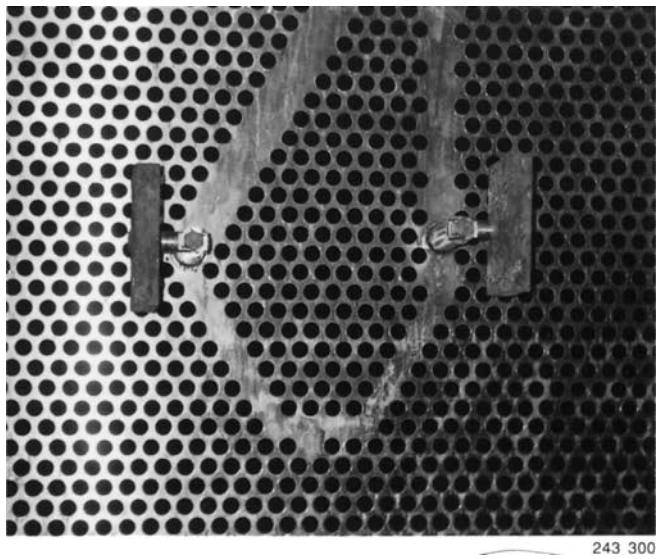


FIG. C-275 Example of sacrificial anodes for protection of tubes and waterbox. Here the typical shape of the CB air cooler can be seen. (Source: Alstom.)

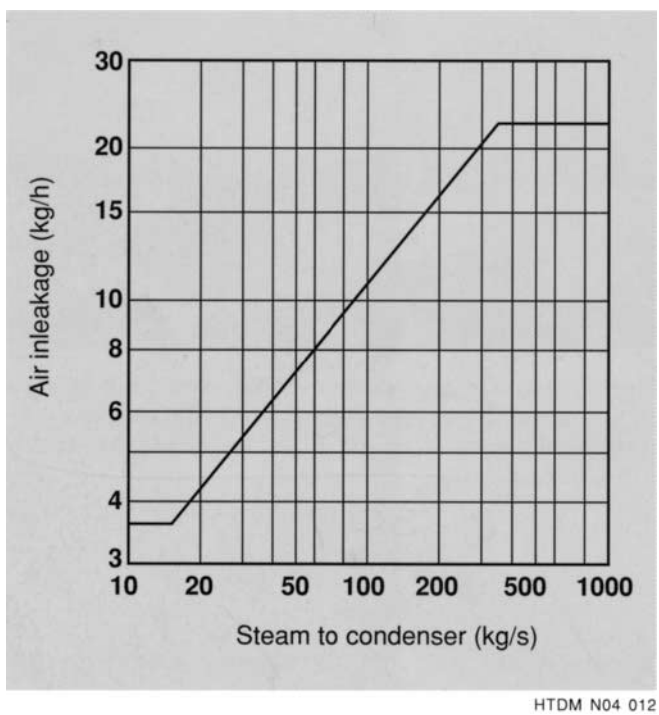


FIG. C-276 Determining the air inleakage as a function of the steam flow to the condenser. (Source: Alstom.)

utilities) recommendations. This is a reliable venting system for all types of loads, requiring minimum equipment and operational outlay. With improved venting characteristics, considerable savings in investment can be achieved with this OEM's design (see Fig. C-276).

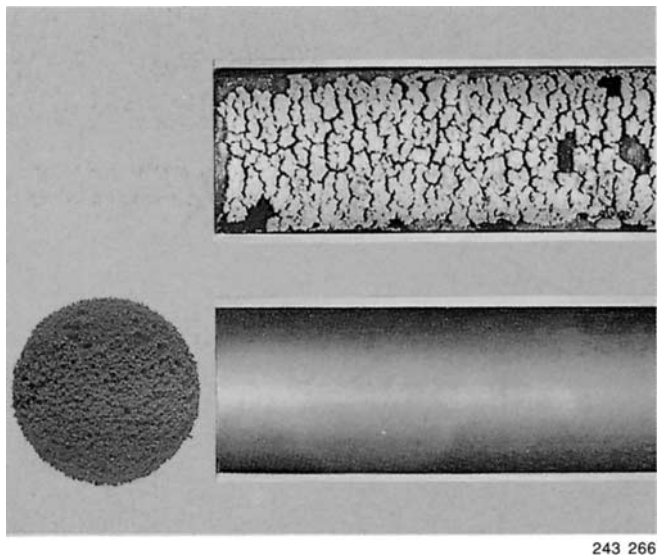


FIG. C-277 Example of a fouled tube (without cleaning system) and a clean tube (with sponge ball cleaning system). (Source: Alstom.)

Startup venting

Generally the service ejectors are also used as startup ejectors for generating the necessary vacuum in the water/steam system before starting up the plant. With an end-user request, special hogging vacuum pumps can also be employed to reduce the evacuation time. For this purpose usually water ring pumps are provided, as these have a constant high flow rate over a wide pressure range.

Waterbox venting

For economic cooling with fresh water or sea water, the outlet waterbox must have a slight vacuum due to the geodetic requirements. This results in degassing of part of the cooling water's dissolved air. This degassed air must be constantly removed and for this purpose single-stage water ring pumps are usually employed.

Accessories

Basically there are two major accessories: the sponge ball cleaning system and the steam dump device (SDD).

Sponge ball cleaning system (see Fig. C-277)

To a greater or lesser degree all cooling water contains dirt particles that, without countermeasures being taken, adhere to the insides of the condenser tubes thus impairing the efficiency of the heat transfer.

With a continuously operated cleaning system fouling can be reduced to a minimum, the so-called standard fouling. This standard fouling also protects the tube material from erosion or corrosion.

A cleaning system is also recommended for corrosion-resistant materials, such as titanium or high-grade steel. In contrast to alloys containing copper, these materials tend to biofouling, i.e., to forming layers of bacteria. This, in contrast to copper, is due to them being nontoxic to bacteria.

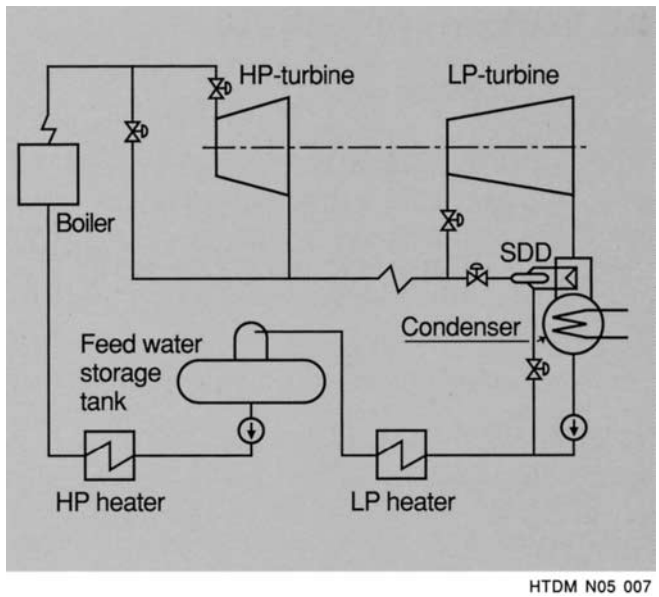


FIG. C-278 HP/LP bypass system with steam dumping into the condenser. (Source: Alstom.)

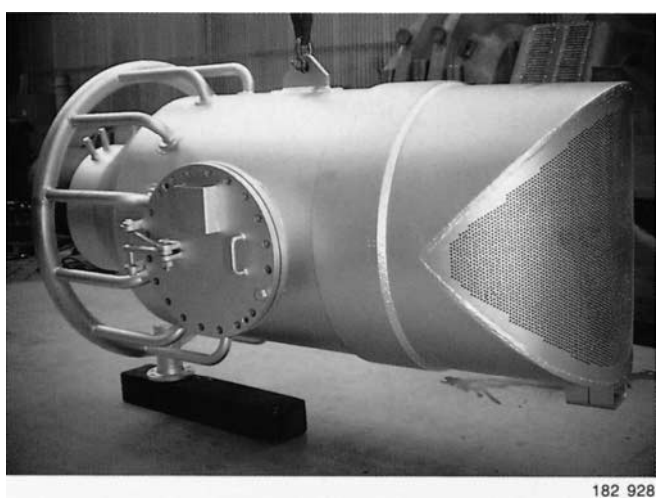


FIG. C-279 Steam dump device (SDD). The high-energy steam is cooled with condensate and led into the condenser via a perforated cone above the tubes. (Source: Alstom.)

Steam dump device (SDD)

End users often need to bypass the turbine during the startup operation or in the event of load rejections and to route the boiler steam directly into the condenser.

A component part of this bypass system is the steam dump device (SDD) into the condenser. See Figs. C-278 and C-279. With SDDs the high-energy steam is attemperated with spray water (taken downstream of condensate pumps) and introduced into the condenser via a perforated cone above the tubes.

This SDD system transports the steam smoothly into the condenser. It has a low noise level.



FIG. C-280 CB condenser during manufacture, ready for tubing. (Source: Alstom.)

Design, Manufacture

Manufacturing drawings and tube patterns are raised on CAD systems, enabling direct transfer to numerically controlled tool machines. See also Figs. C-280 through C-282.

Figure C-283 shows the front end (cooling water inlet and outlet) of a two-pass CB condenser in the turbine building of a power plant.

The water inlet is at the bottom and the water outlet at the top. The CB condenser itself is compact and allows simple piping assembly.

The space saving contributes to reducing the costs of the turbine building.

Choosing a Condenser

The condenser of choice should be an optimum combination of:

- High thermal performance
- Compact design
- Optimum space utilization
- Self-supporting, robust structure without additional internal supports required



FIG. C-281 Tubesheet and support plates must be in exact alignment for tubing to be carried out correctly. (Source: Alstom.)



FIG. C-282 Titanium tubes welded into the tubesheet. The weld quality achieved requires years of experience and careful attention to detail. (Source: Alstom.)

- Economical manufacture
- Simple transport and assembly
- Extremely low oxygen content in the condensate without any additional measures
- Simple makeup water supply
- No condensate subcooling, resulting in higher efficiency
- High availability

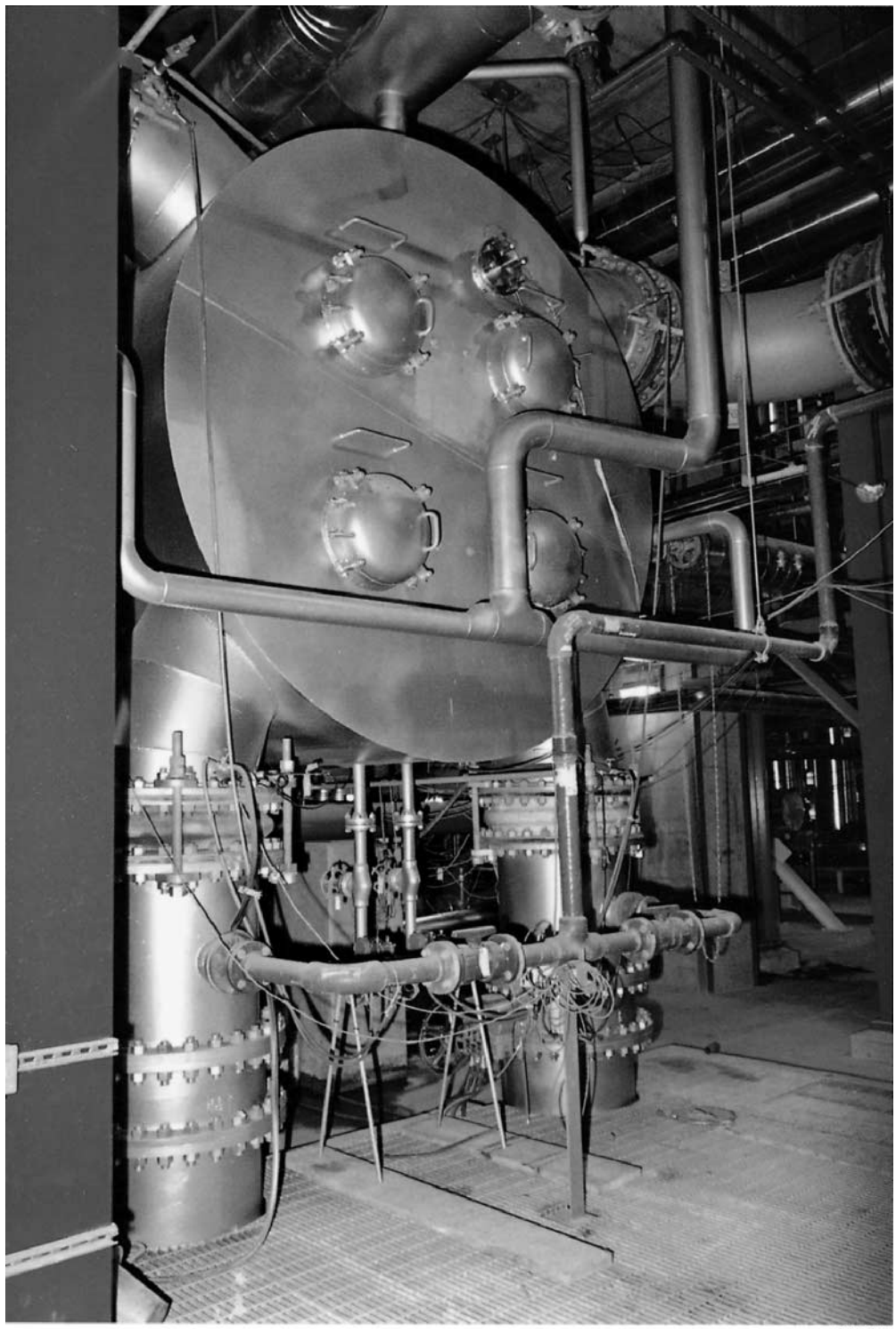


FIG. C-283 CB condenser in the turbine building of a power plant. (Source: Alstom.)

Condition Monitoring (CM); Condition-Monitoring System(s) (CMS); Engine Condition Monitoring (ECM); Engine Condition–Monitoring System(s) (ECMS)* (*see also* Measurement)

These four terms are synonyms for the same concept. Most commonly, the system that can be maintained by a CMS, in a process engineer's world, is a plant system consisting of:

- A driver (gas-turbine, steam-turbine, or electric motor) and a driven component (typically compressor or pump) plus gearbox and coupling and all accessories
- A generating turbine set (many process plants and refineries are becoming small power producers or SPPs due to deregulation of the electrical industry worldwide)

The CM theory that applies to mechanical (or electrical) drive or generating package is the same. The scope of that concept varies, however, among engineers. To some, CM means just the vibration analysis (VA) system that accompanies the machinery system, so it is best to define meaning and intent immediately. To others, CM means anything, including VA, that can be used to determine the health of a machinery system and its components.

In modern plants or plants that are being modernized, CMS [together with life-cycle assessment (LCA)] is a potential basis for some very expensive retrofit engineering or reengineering, especially if major failures or production-loss events have occurred, without any warning provided by the existing system. Process engineers frequently find themselves being pressured into buying expensive CMS, without being actually convinced of the relevance of the entire system. Process engineers generally affirm that rotating machinery is the cause of most of their problems. Therefore, CM, retrofitted or otherwise, overdesigned or not, the cause of increased "nuisance" trips or otherwise, is high on their list of "learn more about." If they do design the right system scope, however, they will see:

- A reduction in costs per fired hour
- Reduced incidents of lost time
- Improved environmental performance with a potential for reduced emissions
- An opportunity to debate and reduce the plant's insurance premiums

Note that the working definition of CM in this book includes LCA and any associated performance analysis (PA). (LCA and PA will be dealt with under LCA in this book.)

Scope and Selection of Condition Monitoring Systems

The troubleshooting process—how easy it is to troubleshoot a machine, how often it needs to be done, and so forth—depends largely on the type of maintenance philosophy applied during the machine's operational life. This basic philosophy affects how often one has to think about the machine.

Not surprisingly, the same strategies and philosophy that are behind a turbomachinery item's selection play a key role in defining maintenance requirements. Unfortunately, a number of operators never link maintenance

*Source: Soares, C. M., vibration course notes, 2000.

philosophy and troubleshooting to an appropriate extent. They either leave themselves wide open for disastrous repair bills or spend more than they need to on maintenance.

This is poor *risk management*. Applied to an item of rotating machinery in critical service it translates into bad business practice. This section describes three basic strategies appropriate for all rotating machinery. A choice of these strategies should not be made by the manufacturer, but by the end user. It needs to fit the operator's specific application and comfort level. The manufacturer's resources and technical expertise should be utilized to support the decision process by providing relevant information.

The operator typically proceeds as follows. Basic goals are itemized. The highest priority is to maximize production. Optimizing production per unit of energy is part of that aim. Maximum availability and reliability (i.e., no unplanned downtime) are also critical. Operators struggle with financial budgets and therefore pressure to minimize cost. They want to minimize the maintenance, service, and repair activity.

Too little maintenance results in unexpected failures and consequential major losses of production and/or customers. This impractical approach is termed *reactive strategy*, and should be avoided on all important machinery. Optimum maintenance strategy balances reasonable costs with maximum possible availability and reliability. The two main maintenance strategies employed by companies today are labeled *predictive strategy* and *preventive strategy*.

Predictive strategy

Predictive maintenance strategies operate without a regular plan for service work or exchange of parts. A maintenance plan is only set up if there's proof of deterioration. Consequently, a company with a predictive strategy favors minimizing cost over maximizing use. Annual cost of this strategy may typically only average out to 1 to 2 percent of the prime equipment price.

With a predictive maintenance strategy, long-term plans may involve only two regular procedures:

1. Monitoring of operating data as follows:
   - Gas path (mass flow, heed, efficiency)
   - Water coolant (differential temperature)
   - Oil analysis (water content, deterioration of antiaging additives)
2. Vibration analysis measurements as follows:
   - Fast Fourier transfer (FFT) analysis (shaft, pinions), using eddy current probes at normal load and turndown
   - FFT analysis (all bearing housings), at normal load

Preventive strategy

In contrast with predictive strategy, preventive strategy aims toward maximum safety against unexpected failures. The concept here is to predict the average lifespan of a part and then replace it before the end of that lifespan. Annual cost is therefore higher (anywhere from perhaps 8–10 percent of the prime equipment price to about 35 percent of the cost of a replacement machine, depending on type) because of the higher numbers of spare parts that need to be purchased and warehoused.

Besides the effects of choice of maintenance strategy on the troubleshooting time and effort required, the application service the unit is in also has an effect. With

increasingly tough environmental legislation that in turn demands maximum energy usage and recovery, power recovery processes are increasing in number. The deregulation of the power industry, which in turn results in the increase of small power producers (such as process plants) also serves to increase this number.

In some installations today, vibration probes and other items used for vibration analysis may be installed separately (frequently at different times) than other items used to monitor the health of a machine. However, increasingly vibration monitoring is part of an overall CMS that then monitors the overall health of all parameters that might indicate the health of a machine. The CMS is usually provided by the OEM as part of the machinery purchase price.

On occasion, the end user finds that the original CMS is limited in that it cannot do as much analysis as is desired. Certain items on the CMS (such as specific vibration probes) may be *retrofitted*. Or the OEM or an external vendor supply may retrofit a different CMS.

One key element in a CMS is VA. VA can solve up to 85 percent of the problems found on rotating machinery. As VA instrumentation gets more sophisticated, the number of problems it can uncover increases. At the same time, however, VA readings themselves may not finally confirm the existence of a problem. Other corroborating readings from other instrumentation in the CMS are required to confirm a problem condition.

The selection of VA instrumentation therefore should match the accuracy of the rest of the CMS system. For example, it is pointless having very accurate VA sensors and crude bearing temperature sensors. This is because the problem associated with, for instance, high vibration in a certain location and low bearing cavity temperature may be different from the one that occurs with the same vibration reading and a slight increase in temperature.

Some examples of how temperature readings fit into the overall problem diagnosis grid follow:

Problem 1. Crossover tube failure:

Fuel pressure: Up or down

Unevenness of flame in combustor (sound indication): No change

Exhaust temperature spread: Up considerably

Exhaust temperature (average): No change

Problem 2. Cracked combustion liner:

As for Problem 1, except there is audible unevenness (noise) in combustor

Also, vibration readings may be observed to increase

Problem 3. Combustor fouling:

As for Problem 2, except exhaust temperature drops and vibration levels may not indicate any change

Problem 4. Incipient bearing failure:

Differential temperature (bearing): Up

Bearing pressure: Down

Vibration: Up

Problem 5. Damaged turbine blades:

Vibration increase: Large

Exhaust temperature increase

Turbine Diagnosis Table

Problem	Therm Efficiency	P3/P4	T3/T4	Vibration Reading	Different Bearing Temperature	Wheel Space Temperature	Bearing Pressure
Fouling	–	NC	–	+	NC	+	NC
Damaged blade	–	NC	–	+	NC	NC	NC
Nozzle	–	–	–	+	NC	+	NC
Bearing failure	NC	NC	NC	+	+	NC	–
Cooling air failure	NC	NC	NC	NC	+	+	NC

NC, no change.
+, increase in reading.
–, decrease in reading.

Temperature readings may be useful for diagnosing problems not related to continual operation. For determining, for instance, optimized time between hot section inspections, life cycle assessment is used. Equivalent running hours based on starts, fuel consumption, and peak temperatures are among other factors used.

Most VA equipment manufacturers now provide the entire CMS and may offer end users the option of selecting different probes to be used in the package. The end user needs to be aware of the VA manufacturer's strengths and weaknesses in order to effectively diagnose problems with the CMS or the VA equipment within the CMS.

The effectiveness of condition monitoring systems depends on what they consist of and where, when, and how they are applied.

CMS are, deservedly, one of the most controversial and discussed items in turbomachinery technology today. The application the system is put together for, the machinery that it has been put on, and the accuracy and positioning of its components may make the same system a great cost saver in one instance and an elaborate waste of money in another.

Appropriately used, they are a highly effective means of extending component life and time between overhauls (TBOs). Inappropriately used, they can provide a false sense of security at a very high price and even increase the number of service calls required on a system. All of these factors are heavily in evidence in global areas of high development activity, which also have the finances to pay (appropriately or otherwise) for these systems.

The case for CMS. As long as CMS are not thought of as a substitute for common-sense or process knowledge, having the appropriate CMS pays dividends. Industry measurements have provided the following impressive figures (1998 $U.S.):

1. From data drawn from the North American power industry:
   - Run-to-failure strategy cost was $18/hp
   - Planned maintenance (overhaul at specified time intervals) cost was $13/hp
   - Cost using a CMS was $9/hp, indicating maintenance cost increase of about 100 percent if no CMS or planned maintenance was done.
2. From a U.S. nuclear plant's data:
   - Savings of $2 million in one year using CMS and $3.5 million the next.
3. From a survey of power, paper, metal, food, and textile producers in the U.S., Europe, and Australia:

- Savings of 50 to 80 percent in repair and maintenance costs
- Savings of over 30 percent in spares inventories
- Profit margin and revenue up 20 to 60 percent

The only catch is: how much money do you spend on measurement of what parameters, on what machines, and when.

The strongest case for CMS is in prototype applications, design and development runs, or where the cost of failure, financial or in terms of human safety, would be unacceptable. In some of these cases, it may be appropriate to integrate certain control functions within the CMS.

Users purchasing a CMS for a new application or for retrofit would be well advised to ensure that the designers of that system know the user's process, application history, global history of the models in question, and the CMS elements (software, hardware, and theory) thoroughly. CMS vendors should be invited to discuss the anticipated life of their proposed components, because particularly in the case of CMS integrated with controls, the following scenario has occurred. CMS vendors who do not manufacture the controls in their system have to buy them off the shelf. It is unlikely that they will be able to purchase the best or the most rugged controls available, as those are made by control manufacturers (who also make and want to sell CMS) who will not sell controls to a competitor. When the CMS vendors buy their off-the-shelf components, they may not be able to offer maintenance of those model numbers for the life of the plant. This means another model number for the control in question at some point and all the associated potential problems.

Were end users to approach a primarily controls manufacturer for a quote on a CM controls system, they might find that they are offered a system with fewer features than one quoted by a primarily diagnostics manufacturer. This may mean that the controls manufacturer doesn't like the additional features. Or it may mean the customer did not need the features in the first place. Any potential user of an expensive CMS should seek the advice of an independent—one that doesn't sell the systems, or associated hardware of software—authority on the system scope. Sometimes the "included in purchase price" training that comes with CM systems achieves only promotion of further sales of such packages. The customer avoids this by taking the "free" training and then using appropriate independent training to enhance critical assessment of the application and equipment.

Basic CMS components. A basic CMS consists of VA, temperature monitoring, and surge control, pressure, and flow measurements. It may also include performance analysis (PA), LCA, nonintrusive wear monitoring, and a variety of other techniques.

Vibration analysis. In the early operational years of the aviation RB 211 (on which land-based RB211s are based), vibration monitoring (VM) instrumentation helped avoid fan shaft locating bearing failures. These bearings at that point exhibited a rapid rate of progression to failure, but the VM provided a few seconds of warning to allow power reduction and avoid catastrophic failure. VM also helped avoid failure due to:

- Partially missing turbine blades
- Excessive blade tip rubs, spacer ring frettage
- Misplaced or missing locking plates
- Oil migration into compressor drums
- Shaft coupling misalignment
- Compressor stack blot loosening
- Disc frettage

Installed at the outset, VM more than paid for itself in a short while, although once early design problems were settled, it revealed fewer problems.

With another fleet operator, of JT9Ds this time, the VM picked up five cases of high engine vibration in 65,000 h (each with a major problem), including a case of a 150-degree round crack in the high-pressure turbine shaft. In three of those five cases, VM was the only alarm indicator.

The above instances all required a tracking filter to properly isolate the flaws in question. So would similar land-based applications, as aeroderivatives in the land-based gas-turbine mix are becoming more prevalent.

How much is too much: which parameters on what machine. Observations of CMS proposals and purchases in large power plants included one case where a highly sophisticated system was being recommended as part of the remodeling for a thermal plant. The thermal plant had a large number of steam turbines. Their design was conservative in terms of peak operating temperatures and they had been operating relatively trouble free for about 20 years, as had the associated boiler feed pumps. The OEM-supplied CMS would have trouble tackling the parameter accuracies a newer turbomachinery equivalent item might require. However, it was adequate for the limited temperature ranges and vibration these very conservatively designed steam turbines and feed pumps had ever seen or will ever see.

One such recommended system would have been more appropriate for the test cells of the latest 90,000-lb thrust development aeroengines; it was expensive "overkill" for the steam turbines. Its specification described an expert system that included (many of these items were already measured on the existing system):

- For the turbine generator(s):
  - Displacement probes VM
  - Velocity probe VM
  - Eccentricity monitoring
  - Dual thrust position monitoring
  - Dual case expansion monitoring
  - Differential expansion monitoring
  - Dual valve position monitoring
  - Rotor speed indicator (all of the above with panel indicators)
  - Phase angle transducer
- For the boiler feed water pumps:
  - Displacement probes VM
  - Accelerometer monitoring for two positions on the hydraulic coupling
  - Speed monitor
  - Dual thrust position monitor
  - Phase angle transducer

Such a system might measure to greater accuracy than the existing system, but greater accuracy for these parameters in this application might not be necessary in the first place. The expert system then went into specific, in some instances too conservative, specifications with respect to operating temperature ranges of transducers, temperature sensitivity of probe and cable, differential pressure withstanding (of gear oil) potential of transducers, double braiding of shields on cables, transducer protection, electrical isolators, environmental specifications, cable pull strengths, probe mounting potential, transducer frequency response range, relay configuration and contact ratings, visual displays, calibration reprogramming or disabling of any monitor, online continuous monitoring, expert

system diagnostics, and a whole host of other factors that could effectively eliminate other vendors that had not suggested the limits specified within the specification.

What might have been more to the point was continuous monitoring for creep fatigue degradation, considering the vintage of the units. The need for this expense would need to be assessed in terms of where the turbomachinery components lay on their stress endurance curves for normal operation and abnormal cyclic operations. Earlier methods would have used measurements taken from thermocouples and pressure transducers to perform offline calculations, and used conservative design codes. The online method system supplied by some vendors has the potential for inputting data from inspections at shutdown, modeling future potential temperature excursions, storing data for trending, and visual, real-time displays.

Approaching the OEM of the machinery in question may be the cheapest route to follow, even if the machines are past the warranty cycle. Thanks to the advance of modern electronics, most of the programmable logic controllers (PLCs) that are supplied as integral to their machinery package could accommodate the additional readings that the client may want based on operating experience. Sometimes the provision might already be built in, but often an OEM stays away from giving a customer too much extraneous data. Some OEMs err on the side of providing less data, until operations experience warrants otherwise. If the OEM proves stubborn about adding the monitoring facilities, there are now monitoring packages available that will work online, real time, remote, take data from marshaling stations, or any combination thereof. Generally they provide the same useful data as a much more expensive system that uses pentium equivalents. They may not have the same storage capacity, but current storage capability may be adequate and less expensive. Also, the PLC systems can be designed to allow assessing the machinery situation from another remote terminal or computer.

Some users prefer the use of nonintrusive wear monitors (in applications where bearing or other component deterioration might be an issue) to oil debris monitoring (better suited to reciprocating machines because of the lead time) or relying totally on temperature readings for those indications. These monitors work using neutron bombardment techniques and are slowly gaining in exposure.

Selection of the overall monitoring package. Selection depends on turbomachinery conditions:

1. Complex or simple (for instance modern gas turbine versus simple pump)
2. Prototype or mature model of machine (age of particular machine(s), model number, history in the world, in specific plant)
3. Prototype application or not
4. Changing or declining process field
5. Estimated time for return on investment
6. Environmental regulations currently in place or newly instituted
7. Changing environmental regulations anticipated (for instance impending CO_2 tax)
8. Component life requirements (consider life cycle assessment)
9. Changing performance analysis requirements (consider life cycle assessment)
10. What are your retrofit limitations?

Based on the above, a decision is made. The most common choices today are:

- To buy an expensive comprehensive system. Cost typically in $150 to $250K per machinery train. The main advantage is that such a system may pick up flaws that vibration and performance analysis may not. The disadvantages include:

- Capital cost is high
- System may be prone to large amounts of nuisance data that require expensive analysis
- High level of operator knowledge may be required

- To buy the best vibration monitoring and PA that is needed and use common sense. Typically vibration monitoring picks up from 60 to 85 percent of the problems encountered on turbomachinery. Performance analysis can pick up 10 to 35 percent of the remaining problems. Generally the remaining 5 percent can be solved with common sense and sufficient expertise. Advantages include:
  - PA can also pick up areas of
    - Operational economic optimization
    - Potential to extend component lives
    - Environmental optimization
  - Personnel tend to develop more expertise than with an "expert" system

Temperature monitoring. Temperature monitoring is very important in determining the health of turbomachinery and is particularly effective when used in conjunction with other parameters, such as vibration. To discuss temperature monitoring, we shall observe this facet of condition monitoring with specific application to the gas turbine—probably the toughest application of temperature monitoring in turbomachinery.

In a gas turbine, exhaust gas temperature is monitored to avoid overheating of turbine components. This is measured with a series of thermocouples in the turbine exhaust. Most gas turbines average the readings of the exhaust thermocouples to produce two single values. (If there are eight thermocouples, half of them will be averaged to give one reading, the other four another reading.) This way if one thermocouple fails, the control can pick the more credible value and avoid a machine trip. This value is one of the inputs fed into the gas turbine's temperature topping control, which in turn controls the fuel flow.

Note also that some VA probe manufacturers have been able to detect problems such as combustion liner cracks with VA *before* temperature indicators picked up differing spread in adjacent thermocouple readings. The temperature monitoring system is set up to take such readings in addition to provide an average reading in more sophisticated gas turbines.

Iron/constantan and chromel/alumel provide the best economical compromise and are the most commonly used. Thermocouples are sheathed in magnesium oxide sheaths for corrosion protection.

Typical effective ranges for thermocouples are as follows:

Copper/constantan	–300 to 750°F
Iron/constantan	–300 to 1600°F
Chromel/alumel	–300 to 2300°F
Chromel/alumel	32 to 1800°F
Platinum and 10% rhodium/platinum	32 to 2800°F
Platinum and 13% rhodium/platinum	32 to 2900°F
Platinum and 30% rhodium/platinum and 6% rhodium	100 to 3270°F
Platinel 1813/platinel 1503	32 to 2372°F
Iridium/iridium 60% and rhodium 40%	2552 to 3326°F

A gas turbine generally has a protective system completely separate from the control system. Another set of temperature sensors record the same or similar

temperatures to the control system sensors. The redundancy provides an additional safety factor. Once the unit is running, at least two flame detectors (many gas turbines only have two) are generally required to indicate flameout for a trip to occur.

As previously mentioned, some systems, either native to the turbine or as part of an overall external comprehensive condition monitoring package, take adjacent thermocouple readings. This technique has succeeded in pinpointing hot spots in the combustor section. These hot spots may happen as a result of a cracked combustion liner, a broken or cracked cross tube, or other source of localized heating.

Monitoring of turbine inlet temperature (TIT), the temperature just before the combustion gases reach the turbine first stage inlet guide vanes, would be preferable and more accurate than exhaust gas temperature monitoring. However, this temperature is not monitored for a practical reason: the damage that would occur if a thermocouple were to break and enter the turbine as FOD (foreign object damage-causing material).

Bearing oil temperature is monitored as it leaves the bearing (discharge temperature). This is the most critical location in the entire oil system. If the oil overheats to the point that appropriate film characteristics are not maintained, the bearing may fail, resulting in overall engine failure. To accurately measure bearing temperature, the thermocouples need to be embedded in the bearing babbitt. These readings give early warning of impending failure in journal (sleeve or rolling element) bearings.

Oil temperature monitoring is particularly critical in rotating machinery. Oil chemical analysis (generically referred to as SOAP) sample-taking is too slow to be useful with high-speed rotating machinery in general. (It can, however, be very useful with reciprocating machinery, where degradation rates are slower.)

Typically, thermocouples can measure from about –200°C to about 2800°C. Thermocouples consist of a bimetallic strip of two dissimilar metals. A voltage proportional to the temperatures of the two junctions is developed. The temperature at one junction is known, so the other temperature can be determined with calibration. Since a voltage is generated, no external voltage needs to be created. However, for accuracy, a reference junction is required for each thermocouple type.

Resistive thermal detectors (RTDs) determine temperature by measuring the change in resistance of an element due to temperature. Platinum is generally used in RTDs because it is mechanically stable and chemically inert. The useful range of platinum in terms of measuring temperature is from about –270°C to 1000°C. An electric current must be supplied to the RTD, and the temperature is determined by the resistance in the element. Any type of conducting wires can be used to connect the element to the measuring device.

RTDs are accurate to within 0.01°C, thermocouples to within 1°C.

Temperature readings can be used as primary indicators in CM or as backup to vibration and other readings to confirm a condition.

Vibration Analysis

Introductory concepts and definitions

Vibration means oscillation in a mechanical system. The parameters that measure vibration are frequency (or frequencies) and magnitude (or amplitude). With vibration, it is either a physical object or structure, or a force that is oscillating. The history over time of vibration may therefore be considered to be simple harmonic or sinusoidal; in other words, it follows a *wave form*.

temperatures to the control system sensors. The redundancy provides an additional safety factor. Once the unit is running, at least two flame detectors (many gas turbines only have two) are generally required to indicate flameout for a trip to occur.

As previously mentioned, some systems, either native to the turbine or as part of an overall external comprehensive condition monitoring package, take adjacent thermocouple readings. This technique has succeeded in pinpointing hot spots in the combustor section. These hot spots may happen as a result of a cracked combustion liner, a broken or cracked cross tube, or other source of localized heating.

Monitoring of turbine inlet temperature (TIT), the temperature just before the combustion gases reach the turbine first stage inlet guide vanes, would be preferable and more accurate than exhaust gas temperature monitoring. However, this temperature is not monitored for a practical reason: the damage that would occur if a thermocouple were to break and enter the turbine as FOD (foreign object damage-causing material).

Bearing oil temperature is monitored as it leaves the bearing (discharge temperature). This is the most critical location in the entire oil system. If the oil overheats to the point that appropriate film characteristics are not maintained, the bearing may fail, resulting in overall engine failure. To accurately measure bearing temperature, the thermocouples need to be embedded in the bearing babbitt. These readings give early warning of impending failure in journal (sleeve or rolling element) bearings.

Oil temperature monitoring is particularly critical in rotating machinery. Oil chemical analysis (generically referred to as SOAP) sample-taking is too slow to be useful with high-speed rotating machinery in general. (It can, however, be very useful with reciprocating machinery, where degradation rates are slower.)

Typically, thermocouples can measure from about –200°C to about 2800°C. Thermocouples consist of a bimetallic strip of two dissimilar metals. A voltage proportional to the temperatures of the two junctions is developed. The temperature at one junction is known, so the other temperature can be determined with calibration. Since a voltage is generated, no external voltage needs to be created. However, for accuracy, a reference junction is required for each thermocouple type.

Resistive thermal detectors (RTDs) determine temperature by measuring the change in resistance of an element due to temperature. Platinum is generally used in RTDs because it is mechanically stable and chemically inert. The useful range of platinum in terms of measuring temperature is from about –270°C to 1000°C. An electric current must be supplied to the RTD, and the temperature is determined by the resistance in the element. Any type of conducting wires can be used to connect the element to the measuring device.

RTDs are accurate to within 0.01°C, thermocouples to within 1°C.

Temperature readings can be used as primary indicators in CM or as backup to vibration and other readings to confirm a condition.

Vibration Analysis

Introductory concepts and definitions

Vibration means oscillation in a mechanical system. The parameters that measure vibration are frequency (or frequencies) and magnitude (or amplitude). With vibration, it is either a physical object or structure, or a force that is oscillating. The history over time of vibration may therefore be considered to be simple harmonic or sinusoidal; in other words, it follows a *wave form*.

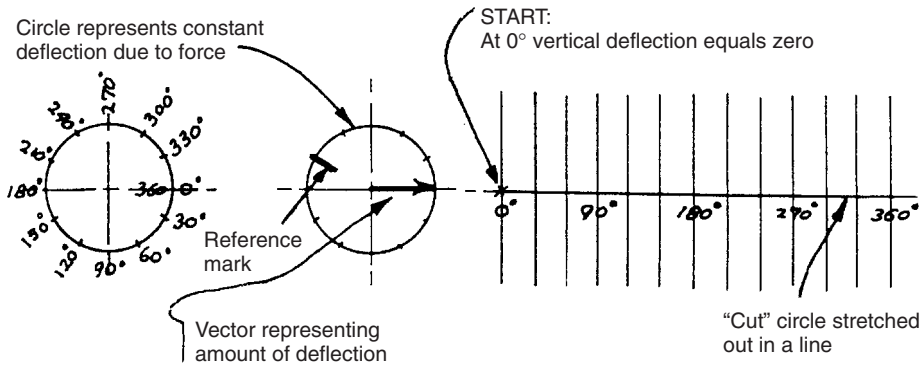


FIG. C-284 A circle represents a constant deflection due to a force vector.[18]

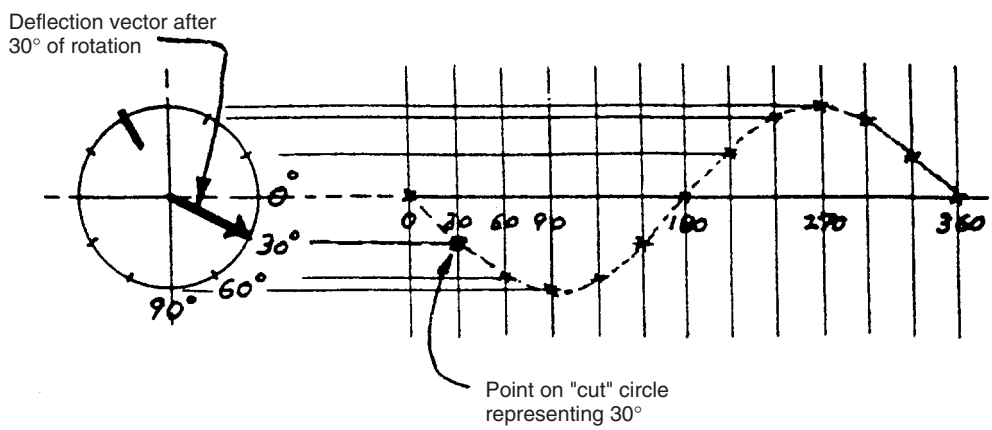


FIG. C-285 Plot of vertical deflection for every 30° of rotating.[18]

Figure C-284 indicates in simple terms how the sinusoidal or wave form is generated. Assume than 0° and 360° occur at 3 o'clock. (Angular position in degree increases in the clockwise direction.) Visualize the circle as a long string that one cuts at the 0/360° point and stretches into a long line with 0° at the left end.

Assume that the circle represents constant deflection due to, for instance, a centrifugal force resulting from unbalance. For the purposes of illustration, assume that we are considering vertical vibration at one point of the machine only, such as a particular bearing housing. Figure C-285 shows the plot continued for a full cycle of 360°.

Frequency is defined as cycles per unit time. To visualize the concept of frequency, consider the cycle in Fig. C-285 takes 1 s to occur, as indicated on the horizontal axis in Fig. C-286.

If this cycle took 1/30 s, the rotational speed that the sine wave represents would be 1800 rpm. The force causing the vibration and the vibration itself also occur at 1800 rpm.

Let us assume that the unbalance force position is as in Fig. C-287. Let us also assume that there is a reference mark at 7 o'clock. If there is a strobe light directed on the rotor, the reference mark or key will appear to be standing still at the 7 o'clock position.

With the horizontal axis representing zero vertical deflection in Fig. C-288, assume all values above the line are positive and values below are negative. The strobe flashes at the peak of positive deflection, once per cycle.

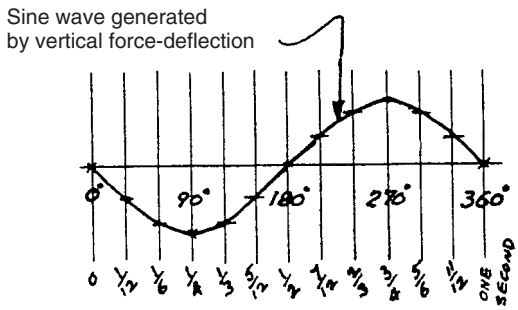


FIG. C-286 Time elapsed as force deflection vector rotates clockwise at one cycle per second (60 rpm).[18]

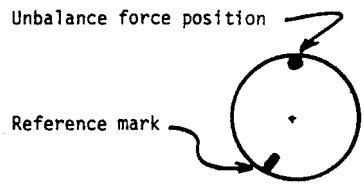


FIG. C-287 Rotor as seen when light is flashing once per revolution. Reference mark "stands still."[18]

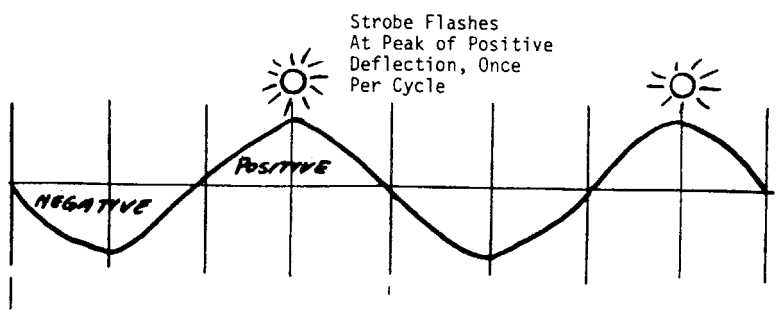


FIG. C-288 Vibration sine wave, assuming strobe light flashes at peak of positive deflection.[18]

Magnitude, or *amplitude*, is the maximum value of a sinusoidal quantity (from the peak of the sine wave to the next peak of the sine wave). This simple model is adequate for theoretical study.

In practice, however, vibration is rarely regular or symmetrical. It is often a combination of several sinusoidal quantities, each with their own frequency and amplitude, that gives one *total signal*. If each of these quantities has a frequency that is a multiple of the lowest frequency, the vibration repeats itself after a specific period, i.e., it is *periodic*. It is possible to use filtration methods to isolate any one or more of the total signal components, resulting in a *filtered signal*. If there is no such relationship between component frequencies, there is no periodicity and the vibration is then *complex vibration*.

Vibration may also be called *deterministic* or *random*. Deterministic vibration is totally predictable at any point in time based on its past history. Most practical cases of vibration are *random*; in other words, future vibration is predictable based on probability that is used to assess past history.

The model most frequently used for vibration is that of a mass attached to a spring. The vibration of this model may be *free* or *forced*. In *free vibration* motion results from the energy of an initial disturbance and then continues to decay to a stable state. For analysis purposes, in an *ideal system* there is no *damping* (or energy dissipation), so the free vibration continues indefinitely. In a *real system*, damping causes vibration to decay to a negligible value. Such free vibration is often called *transient vibration*. *Forced vibration* is different from free vibration in that energy is supplied to the system continuously, which compensates for energy lost due to damping. The *forcing frequency* of the applied energy appears in the vibration signature of the system. Whether vibration is transient or forced, system vibration depends on the relation of the force causing vibration to the rest of the system.

In certain machines, fluid flow movement may go from steady (in a well-designed machine operating near its design point) to erratic or *pulsating* (a steady back and forth movement). A typical example is a reciprocating compressor that develops an inadequate inlet flow condition. The gas in the compressor starts a pulsating movement.

In a centrifugal compressor where inlet flow conditions fall below those required for positive forward flow into and through the compressor, the flow goes back on itself, trying to get the pressure at inlet back up to design point. When this is reached, the flow can proceed on its normal path until pressure falls again due to the same causative factors. Flow reversal again results. This phenomenon is called *surge*.

When a similar condition occurs in a centrifugal pump, where the fluid is liquid, bubbles of gas form due to inadequate pressure. When the bubbles get to a condition of sufficient pressure, they collapse. This phenomenon is called *cavitation*. Both these phenomena will be dealt with further when troubleshooting is discussed.

Natural frequency is the frequency of free vibration of a system. Contrast the following two responses: a rubber band with a stone attached to the end that is pulled, then released and allowed to oscillate; and the vibration of a tuning fork when it is struck. It is evident that the more flexible a part, the lower its natural frequency. Adding weight to the spring system will lower the natural frequency. When the weight's position is placed so as to increase deflection, this also lowers natural frequency.

Resonance of a system in forced vibration, by definition, exists if any change in the excitation frequency causes a decrease in the system's response.

To better understand resonance, consider the case of a car that develops maximum vibration at a speed lower than that of when it is developing maximum power. It may have this maximum vibration at, for instance, idle speed. However, the vibration goes away when rpm are increased. This happens because at the lower rpm, the natural frequency of some part of the car's structure, frame, or wheel assemblies coincided with the rpm at that time. Resonance resulted. At rpm's above or below that value, the vibration lessens.

Resonance can magnify the vibration amplitude of systems with low damping by a large multiple; 10 to 30 times greater for relatively simple systems is not uncommon. Damping can reduce the amplitude magnification, but it does not eliminate the potential for component failure.

A shaft between bearings can be described as a spring, as can a concrete floor on which vibrating machinery is placed, the columns or piles that support the floor, and pipes and tubes in a process plant. A spring then is a part that can flex upon having a force applied and then tend to return to its former position. Each of these springs has its own natural frequency that, when matched by a vibration frequency, will resonate. Piping systems are then a collection of many springs, the natural frequency of which varies according to the span between piping supports and the

TABLE C-18 Conversion Factors for Translational Velocity and Acceleration[19]

Multiply Value in → or → By ↘ To obtain value in ↓	*g*-sec, *g*	ft/s ft/s^2	in/s in/s^2	cm/s cm/s^2	m/s m/s^2
g-s, *g*	1	0.0311	0.00259	0.00102	0.102
ft/s ft/s^2	32.16	1	0.0833	0.0328	3.28
in/s in/s^2	386	12.0	1	0.3937	39.37
cm/s cm/s^2	980	30.48	2.540	1	100
m/s m/s^2	9.80	0.3048	0.0254	0.010	1

TABLE C-19 Conversion Factors for Rotational Velocity and Acceleration[19]

Multiply Value in → or → By ↘ To obtain value in ↓	rad/s rad/s^2	degree/s degree/s^2	rev/s rev/s^2	rev/min rev/min/s
rad/s rad/s^2	1	0.01745	6.283	0.1047
degree/s degree/s^2	57.30	1	360	6.00
rev/s rev/s^2	0.1592	0.00278	1	0.0167
rev/min rev/min/s	9.549	0.1667	60	1

rigidity added by hangers. For instance, although two pieces of pipe may not be resonant, the connecting elbow between them may cause resonance in the system. Fluid in pipes adds to their weight and lowers the natural frequency. The fluid also adds damping that lowers vibration levels.

Not all vibrating pipe sections are in resonance. A quick check is provided by using a temporary brace—either a handheld brace for a small gauge that has its indicator needle vibrating furiously or a pipe length or anchor temporarily used to provide a brace that changes the natural frequency of a system. If there is a major change—the gauge needle stops shaking—then the problem was resonance. If it continues to shake at higher or lower levels, there is another problem, and the source frequency should be investigated.

Sometimes response displacement occurs at a different angle than excitation force. The angle between the response and the excitation is called the *phase angle*.

Tables C-18 through C-20 and Figs. C-289 and C-290 list several definitions common to the field of vibration analysis.

TABLE C-20 Conversion Factors for Simple Harmonic Motion[19]

Multiply numerical value in terms of → By ↘ To obtain value in terms of ↓	Amplitude	Average value	Root-mean-square (rms) value	Peak-to-peak value
Amplitude	1	1.571	1.414	0.500
Average value	0.637	1	0.900	0.318
Root-mean-square (rms) value	0.707	1.111	1	0.354
Peak-to-peak value	2.000	3.142	2.828	1

Types of vibration transducers or probes

There are three main kinds of probe that measure:

- Displacement
- Velocity (rate of change of displacement)
- Acceleration (rate of change of velocity)

If one considers the sine wave representation of vibration earlier in these notes, the difference between these three parameters can be represented mathematically with successive differentiation, as follows:

Displacement x is defined by:

$$x = A \sin \omega t$$

$$\text{Velocity} = dx/dt = \dot{x} = A\omega \cos \omega t$$

$$\text{Acceleration} = d^2x/dt^2 = \ddot{x} = -A\omega^2 \sin \omega t$$

In practice, these are specified thus:

$$\text{Displacement (peak to peak)} = 2A$$

$$\text{Velocity (maximum)} = A\omega$$

$$\text{Acceleration (maximum)} = A\omega^2$$

Note that displacement is independent of frequency, velocity is proportional to frequency, and acceleration is proportional to the square of frequency.

A *transducer* or *probe* is a device that translates some aspect of vibration into a voltage varying signal output that can then be depicted graphically on, for instance, a *spectrum analyzer* and analyzed for information on the machine's condition.

Figure C-291 illustrates parameters for:

- Conversion factors for translational velocity and acceleration
- Conversion factors for rotational velocity and acceleration
- Conversion factors for simple harmonic motion

Selection of transducers for specific applications. *Displacement probes* are sensitive to external conditions, such as heat. Vibration and acceleration probes are direction

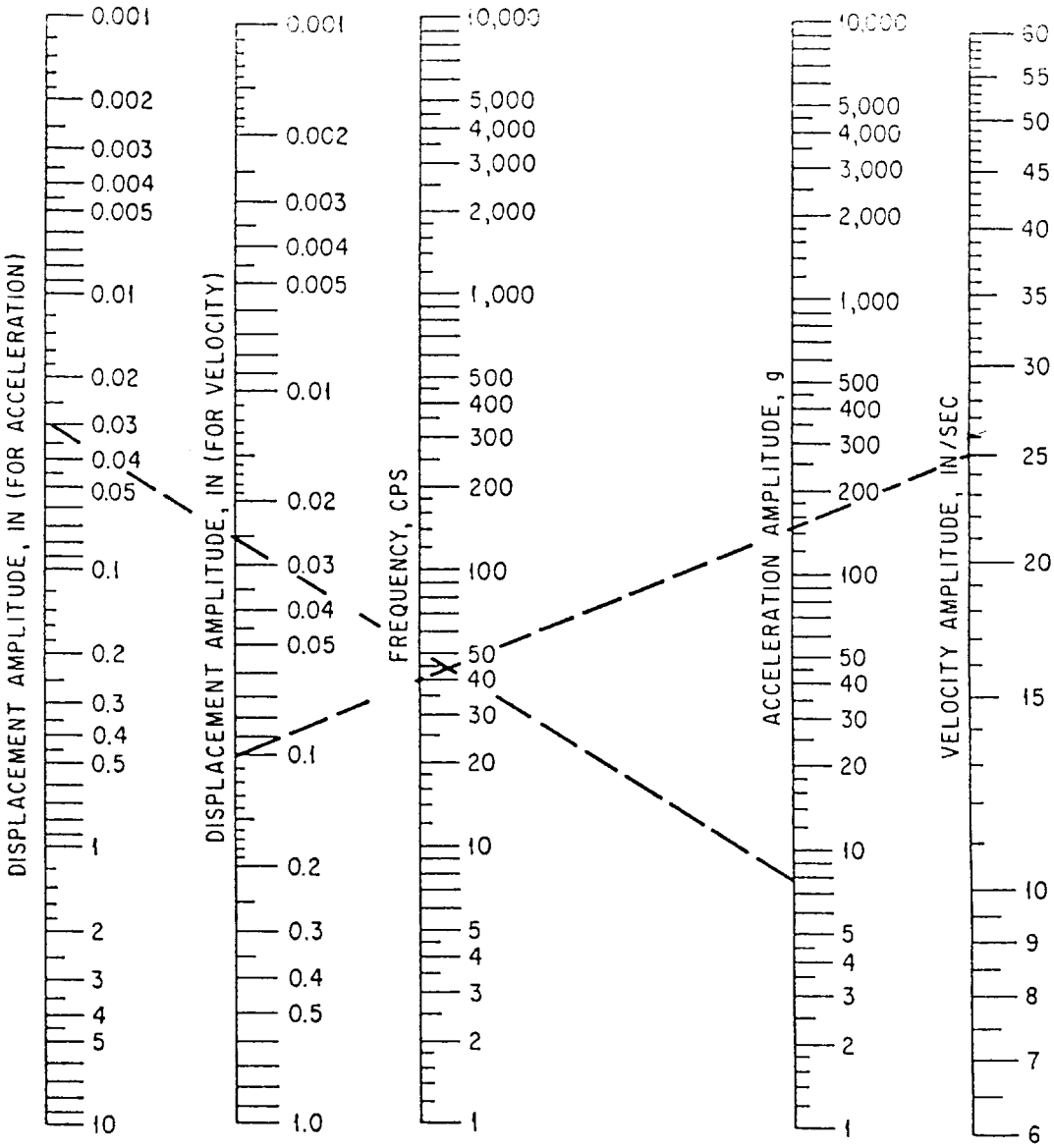


FIG. C-289 Sample nomogram for harmonic motion showing relation of frequency and acceleration amplitude to displacement amplitude (extreme left scale) and relation of frequency and velocity amplitude to displacement amplitude (middle scale on left). Use of the nomogram is indicated by the broken lines. For example, harmonic motion with a displacement amplitude of 0.03 in and a frequency of 50 cps has an acceleration amplitude of 7.7 g; harmonic motion with a displacement amplitude of 0.10 in and a frequency of 40 cps has a velocity amplitude of 25.1 in/s.[19]

sensitive to some extent. Therefore the position they are mounted on the machine may be critical. Portable probes are used by field troubleshooters to help point out where problems may occur. They may be velocity or acceleration probes that are also direction sensitive.

Displacement or proximity probes are generally used where it is critical that a machine or a machine component, such as a shaft, has its growth or relative movement measured. For measuring shaft motion relative to a bearing, the probe can be mounted within the bearing itself.

Most proximity probes are eddy current probes. Eddy probes generate an eddy current field, the strength of which is proportional to the distance between the probe

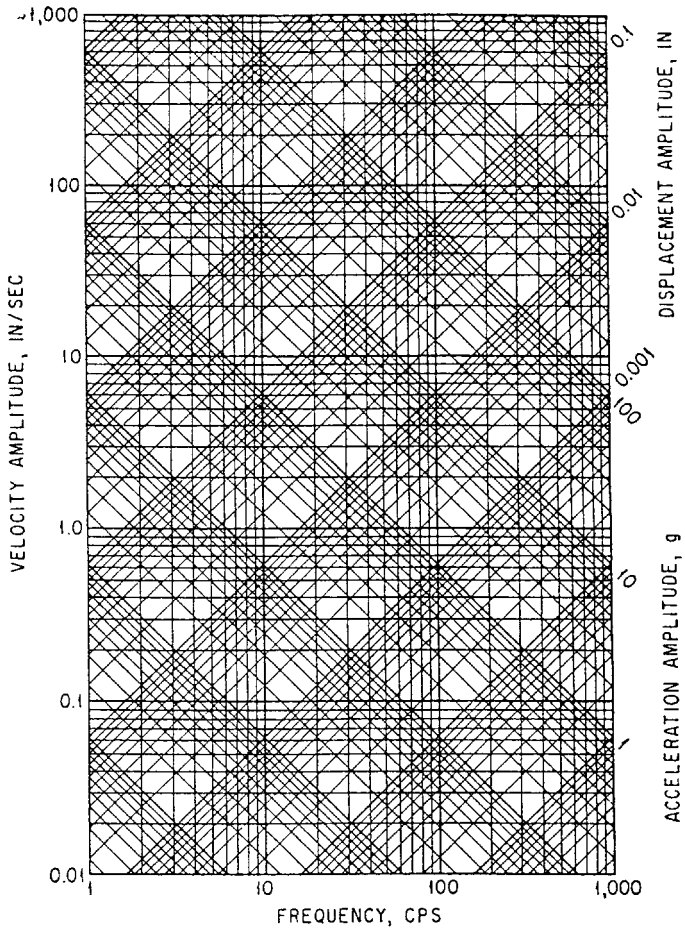


FIG. C-290 Sample nomogram relation of frequency to the amplitudes of displacement, velocity, and acceleration in harmonic motion.[15]

and the surface that is being monitored for thrust movement. They are inexpensive. However, shaft surface conditions and electrical runout can result in false signals. Shaft runout should be less than the smallest value that the probe can measure. Precision turbomachinery, such as gas turbines, have demagnetized, precision ground shafts that maintain optimum runout values.

Velocity probes are common when the application is nonprototype, generally smooth running, and continuously monitored. Their flat frequency response (useful linear bandwidth) is quite limited, which also limits their useful application.

Generally velocity transducers are made of an armature mounted in a magnet. The velocity of the armature creates a proportional voltage output. A high force (noise, N) is required to create a measurable signal (S). These probes are much more expensive than proximity probes, although they are rugged. They are generally mounted on bearing housings, so high noise is available to cause the signal. The main advantage of this probe is its excellent signal-to-noise (S/N) ratio because of its high output and low impedance characteristics.

Their low-frequency response is limited to about 10 Hz, due to the damping component in the transfer function characteristic of the armature-magnet assembly. The resonant peak of the probe limits the high end of the frequency range.

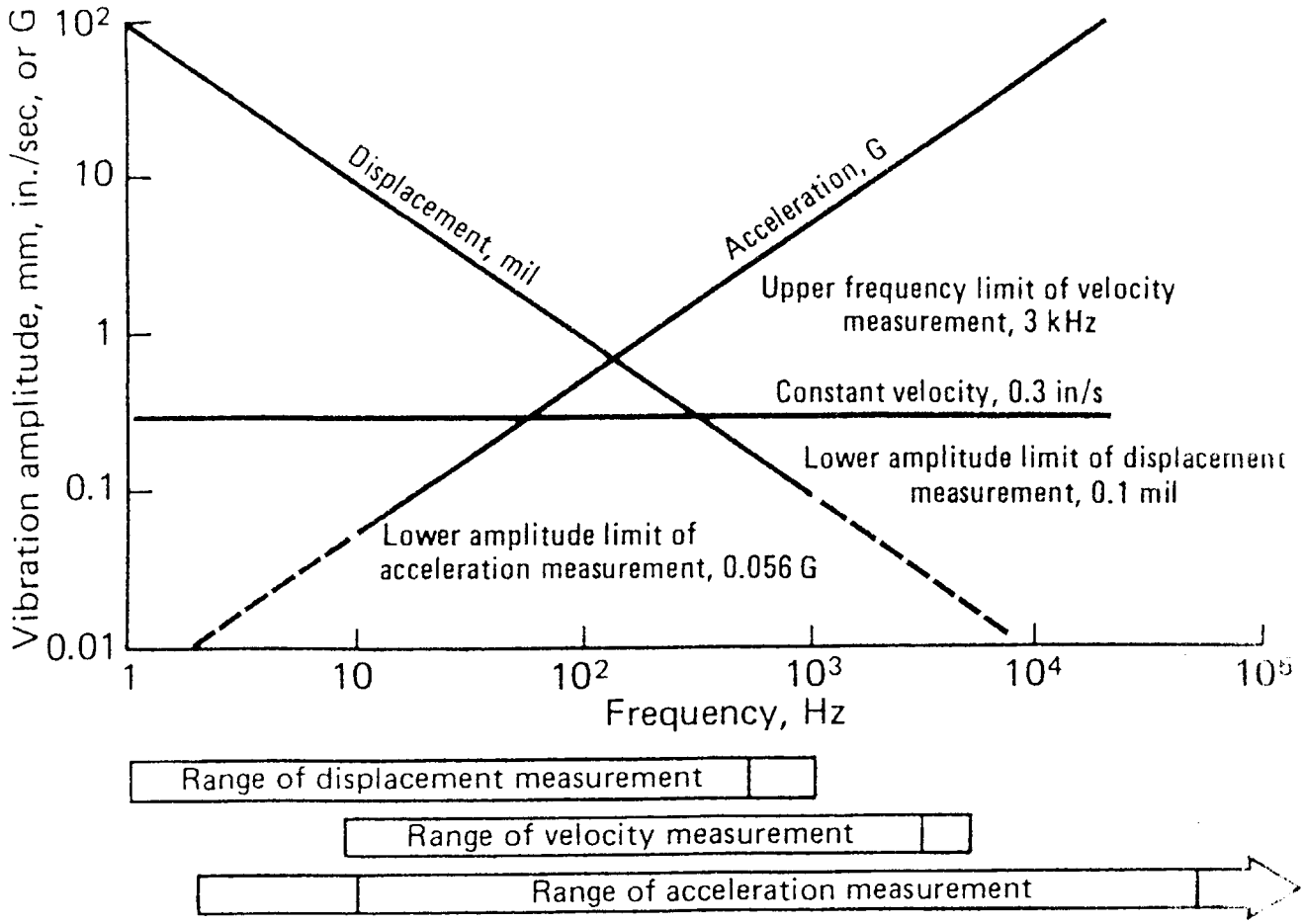


FIG. C-291 Limitations on machinery vibrations analysis.[13]

Accelerometers measure rate of change of velocity or the signal proportional to any force causing degradation. Like velocity probes, they are direction sensitive, so the potential problems to be monitored need to be kept in mind when choosing a location for them to be installed.

In an accelerometer, the acceleration caused by the exciting force acts on a mass, which it transmits to the piezoelectric crystal it is mounted on. This in turn causes a voltage.

Accelerometers have a wide frequency response range. They are not too expensive and are temperature resistant.

Their disadvantages are twofold. First, they are low output/high impedance devices that require loading impedance. Long cables are therefore not an option, unless an amplifier is built into the pickup to provide an amplified signal at low impedance. A power supply is required to do this, which adds expense and weight. Second, typically an acceleration of 1 g at 0.5 Hz produces a displacement of about 9 ft. In other words, poor S/N ratio characterizes the low end of its otherwise wide band response (typically 0.1 Hz to 15 kHz).

With probes in general, choice of one with the right flat frequency response range is critical. It makes the difference between detecting a problem and not knowing what the problem really is. Each probe type has its advantages and limitations, so

despite a theoretical ability to get velocity and displacement by integration(s) of an acceleration reading, more than one type of probe may be required for thorough analysis. Also see Table C-19.

Vibration monitoring of prototypes. Faced with an application that is new and untried, it may be preferable to have a combination of all three types of probes. Accelerometers may be permanently installed with a flat frequency response range that covers the possible combinations for higher frequencies with the components in question. There should also be provision to use a tracking filter for troubleshooting. Velocity probes cover mid-frequency ranges well and problems such as imbalance. Displacement probes cover the lower frequency end of the spectrum and will indicate unforeseen shaft growth.

In the early days of the RB211 (aeroengine model, later a highly successful aeroderivative as well), these tracking filters were critical for getting advance warning of catastrophic failure, such as cracks starting on shafts. These cracks would have progressed to failure quicker than any other monitoring technique in existence could have caught. The aeroengine model is what was used to develop the land-based aeroderivative model. This gives land-based users the added advantage that much of the prototype troubleshooting was already one.

Vibration monitoring selection for remote applications. This may occur on offshore platforms, stations that a company may not want to man extensively. Selections are as before, but the level of monitoring must be stepped up. For instance, if proximity probes and accelerometers were satisfactory in a manned application, all three types of probe should be considered for a remote application.

A qualified individual should visit the site at intervals to ensure that the probes are still measuring what they were intended to monitor. Dirt and debris accumulation, probes being knocked out of position, and other incidents can and will occur.

Readings from these probes can be fed into a computer and transferred to an onshore or head office site for observation and analysis.

Recorders and parameters for measuring vibration

Some machinery problems are detected more easily when certain data manipulation techniques are used. Among the more common techniques are:

I. Amplitude versus frequency (spectrum analyzer)
   Noise attenuation by averaging
   Waterfall graphs

II. Amplitude versus frequency versus time (amplitude versus frequency data observed repeatedly, quickly)

III. Amplitude and phase versus machine rpm (Bode, Nyquist plots)

IV. Time waveform
   Orbits (sometimes called Lissajous patterns)

V. Amplitude versus time (strip chart recorders or electronic data loggers)

Each of these will be looked at in turn.

I. The most common method in use today is *spectrum analysis* (*amplitude versus frequency*) in combination with a tracking filter. Spectrum analysis can be conducted online, as needed, or real time. Real time is very useful if dealing with highly complex, random, or transient vibrations.

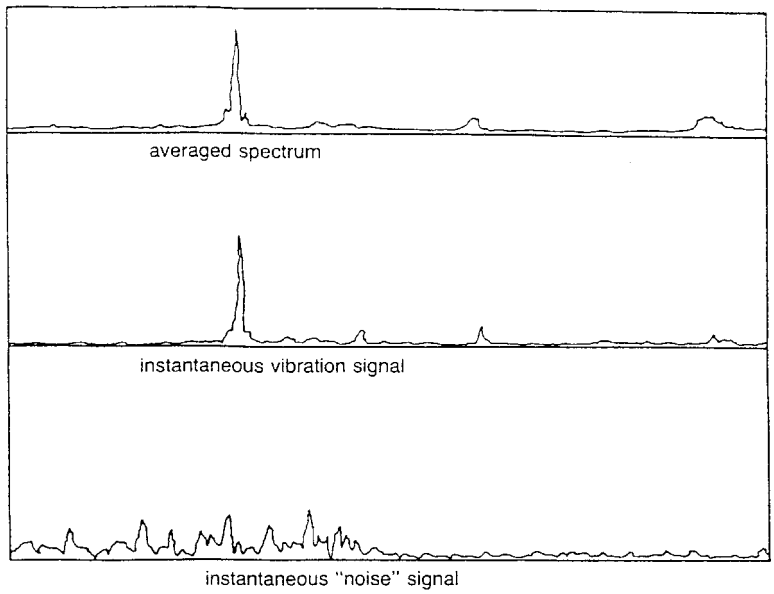


FIG. C-292 Noise attenuation by averaging.[13]

The spectrum analyzer portrays the frequency content of each time domain instant. However, the time domain as well as the frequency domain of a continuous signal change with time. Averaging is used to show which amplitudes predominate in a given signal. Frequently stationary signals are noted (stationary in that statistical properties do not change with time, regardless of what time period is used to measure the average). An example of a condition that gives a stationary signal is a machine running at constant speed and load. Averages may be used to diagnose problems during machinery startup and load change.

As noted in Fig. C-292, averaging is a technique to improve the S/N ratio. The contribution of the random frequencies does not show in the average signal. The large peak in the average signal is the running frequency. Nonaverage signals may be required for diagnosing random problems.

Figure C-293 is a diagrammatic representation of a time signal into a sum of oscillatory functions from which a spectrum is obtained. It helps provide an understanding of what makes up a spectrum signal.

Spectrum analysis is generally advisable if working with a prototype installation. This is because observing actual vibration or noise spectra when a machine is undergoing changes in various operating parameters will provide information that would be difficult or impossible to get in any other way.

A real-time spectrum analyzer can be set up to average incoming data. This feature is useful if:

1. Maximum repeatability of a machine's spectrum is required.
2. The steady-state spectrum of a machine is changing rapidly and unpredictably.
3. Important discrete frequencies are masked by random vibration or noise.

The spectrum analyzer can also be set up for *transient data capture*. Transients are vibration or noise that are one-time events, and are not part of the steady-state spectrum component. To study the transient properly, the signals containing information on the transients will need to be stored in memory. Therefore newer

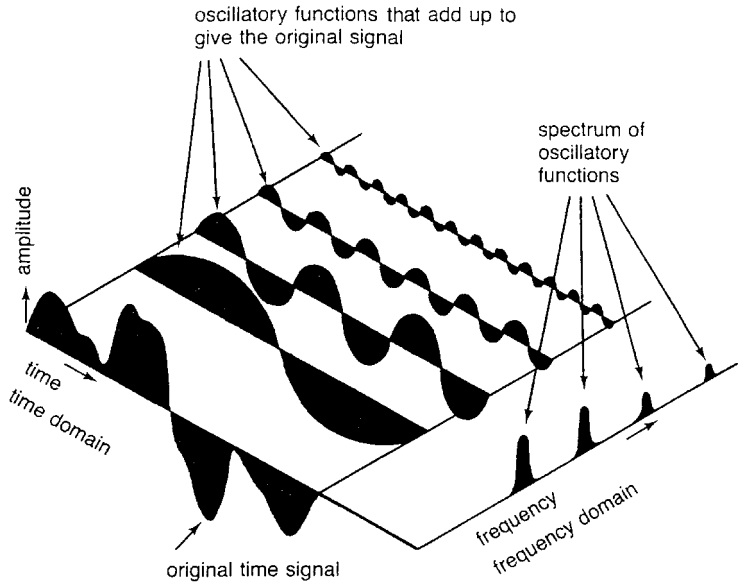


FIG. C-293 Decomposition of a time signal into a sum of oscillatory functions from which a spectrum can be obtained.[13]

spectrum analyzers incorporate *automatic peak hold* features. Automated online systems generally store all peak transients for a predesignated amount of time.

Manual capture of transient data at maximum amplitude may be fine if:

1. The time of the event is known.
2. The event is slow enough, so the operator can push the hold button before the event is over.

In contemporary circumstances, however, this is generally not the case.

II. *Amplitude versus frequency versus time* (generally short-term averages) can be used in *waterfall* graphs to show the growth of certain frequency patterns at, for example, run up, as shown in Fig. C-294. They are used when amplitude versus frequency data need to be recorded repeatedly (or continuously) and quickly. The plots are also often referred to as *cascade* diagrams. Such circumstances include:

1. Resonant conditions or critical speeds during machine startup.
2. Determining the nature of the spectrum during load changes, sudden temperature excursions, and other changes in operating variables.

III. *Amplitude versus machine rpm* diagrams are called Bode plots:

1. Tracking filters and key phasor markings are required.
2. Resonant frequencies are evident by peak amplitudes and 180° phase shift.
3. Machine rpm must avoid coinciding with excitation forces arising from unbalance, misalignment, and other machinery problems.

Figure C-295 is an example of a typical scan on a machine.

An example of the plot being used to troubleshoot is as follows. It is noted there is an amplitude peak as well as 180° phase change at 500 rpm in Fig. C-296. There

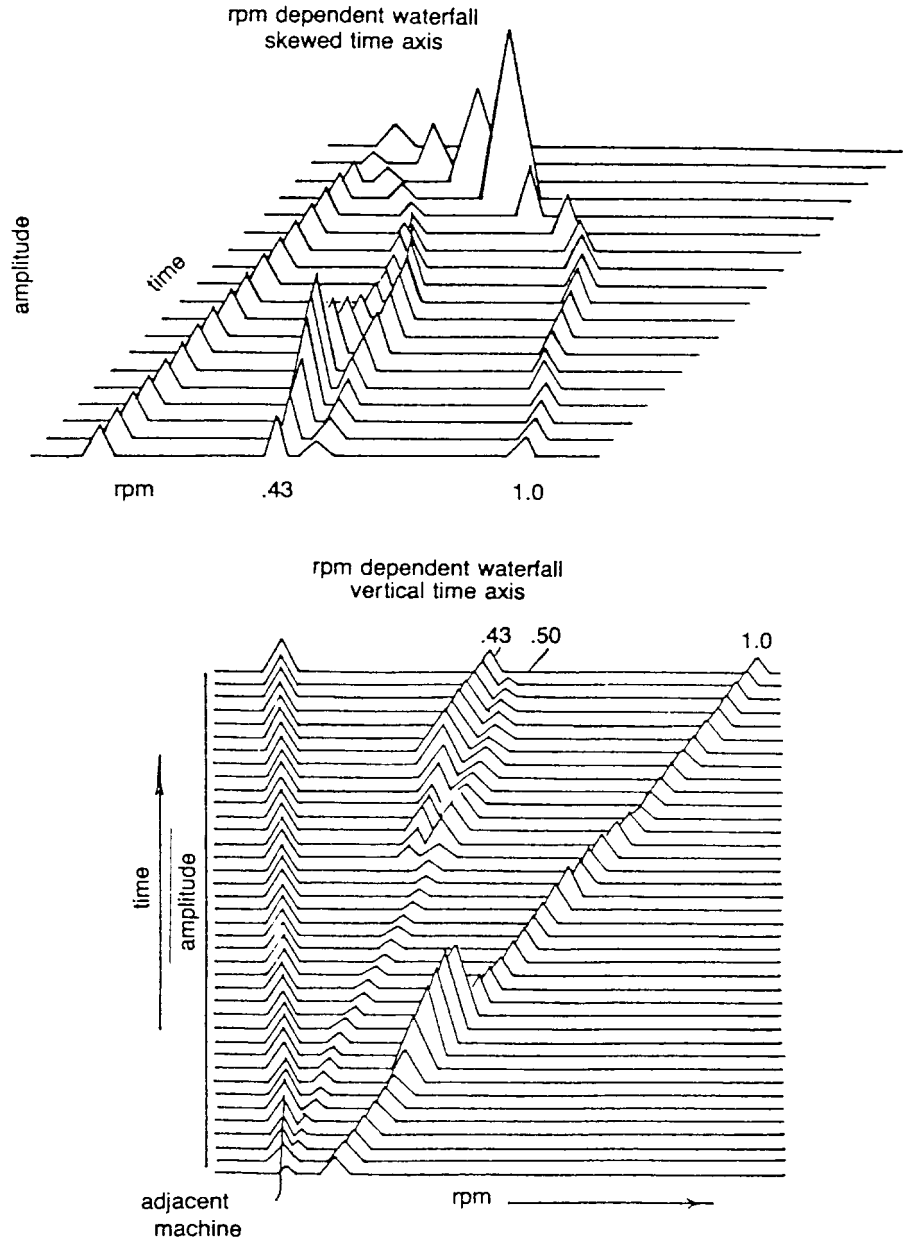


FIG. C-294 Waterfall graph of increasing rpm.[13]

is also a 1200 rpm peak noted but with no corresponding 180° phase shift. The peak at 1200 rpm is therefore unrelated to resonance condition and could be caused by a neighboring machine.

In Fig. C-297 a 180° phase shift is noted at 1400 rpm but without a corresponding amplitude peak. Reasons could include:

- The excitation force at 1400 rpm being extremely low
- A very heavily damped system
- A pickup (sensor) located at the nodal point

In Fig. C-298 an amplitude peak is accompanied by a 360° phase shift. Reasons could include:

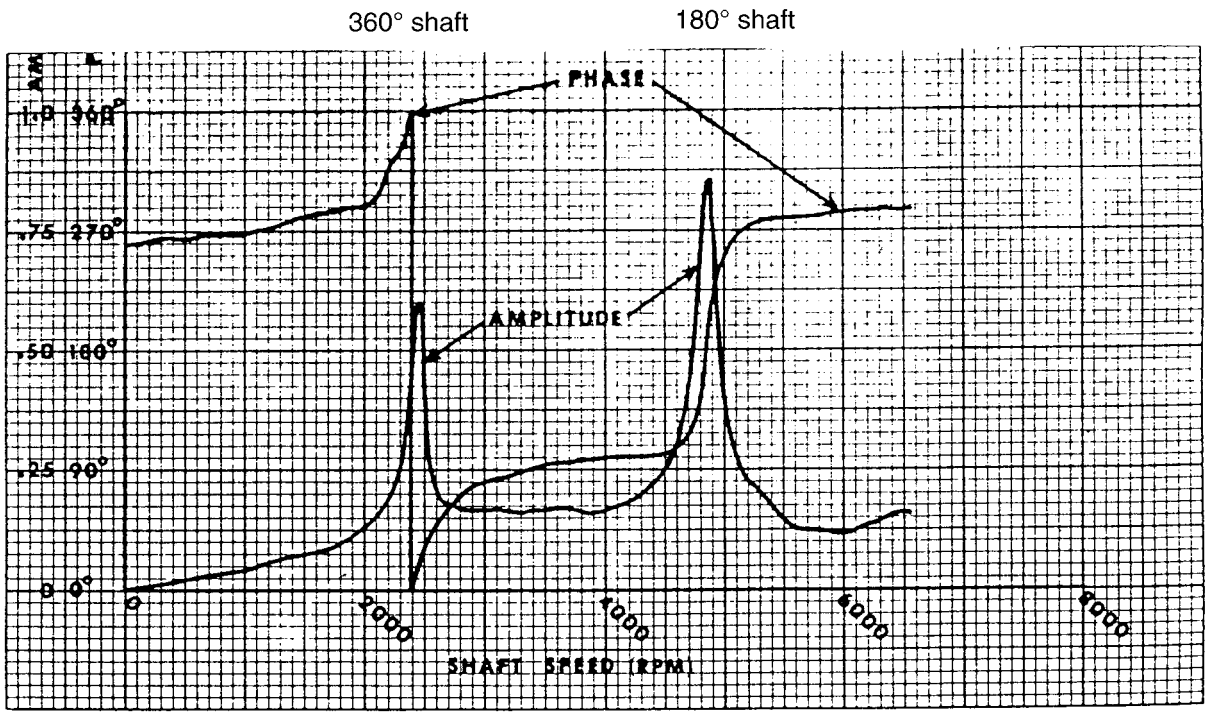


FIG. C-295 Plots of amplitude and phase versus machine rpm clearly identify critical speeds and resonant conditions.[15]

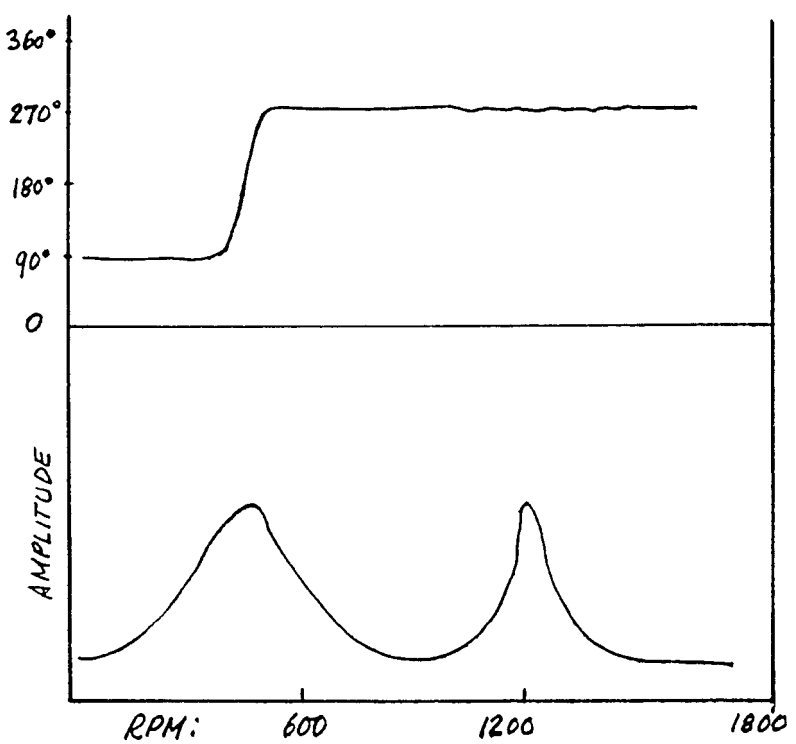


FIG. C-296 Amplitude peak and phase change at 500 rpm.[15]

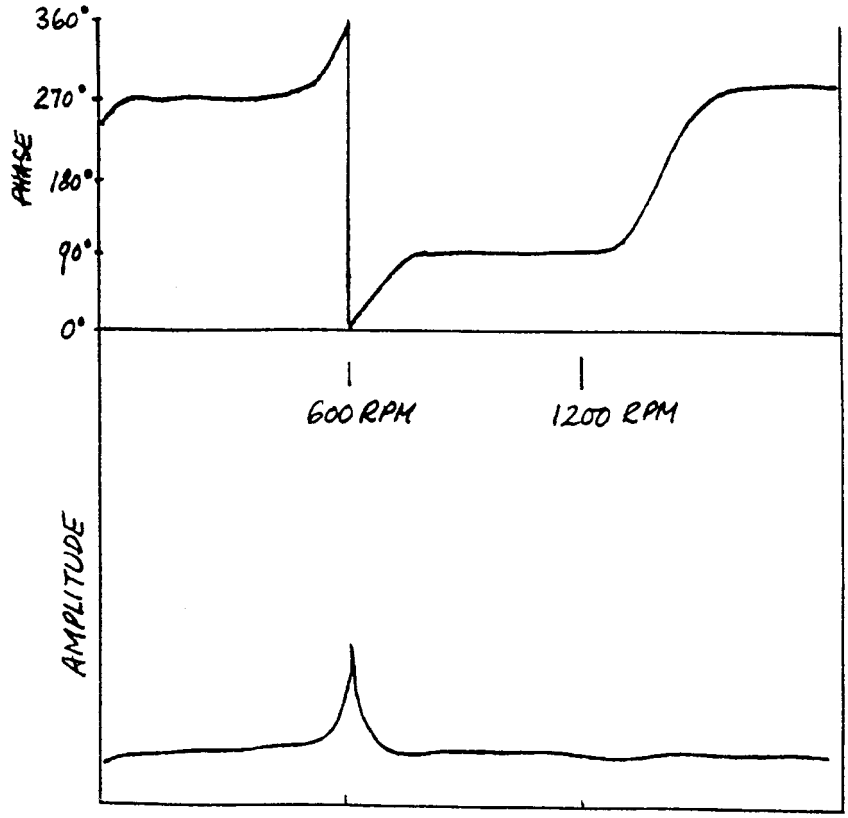


FIG. C-297 180° phase shift at 1400 rpm with no corresponding amplitude peak.[15]

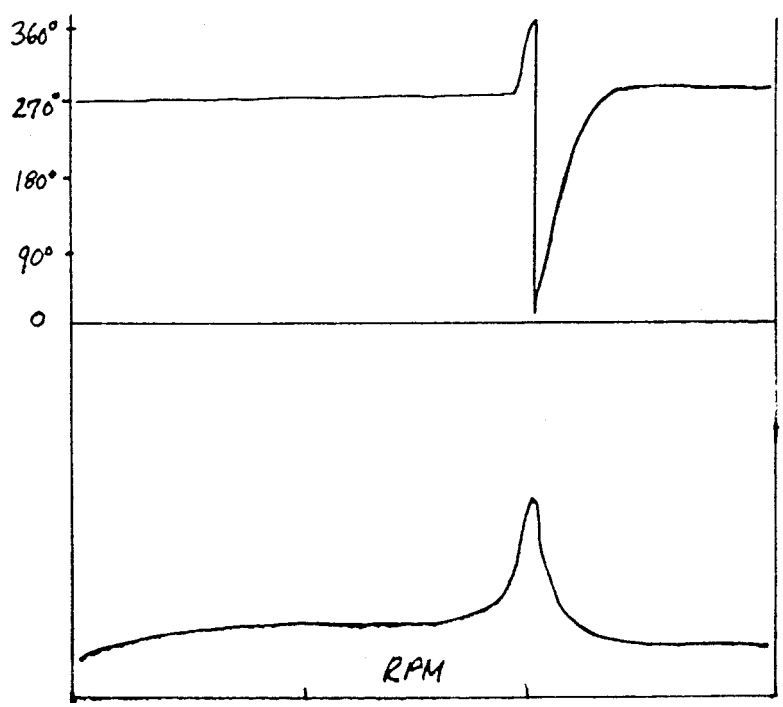


FIG. C-298 An amplitude peak corresponds to a 360° phase shift.[15]

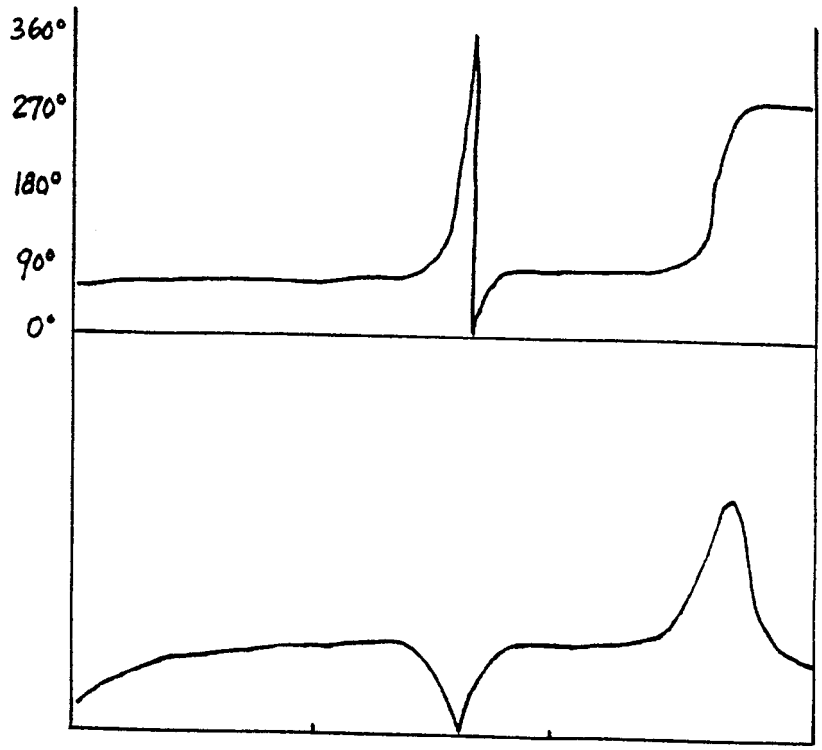


FIG. C-299 A 360° phase change corresponds to an amplitude dip at 1000 rpm.[15]

- Two systems in resonance at or near the same frequency
- Correcting only one cause of resonance may not solve the problem entirely

In Fig. C-299 a dip is seen at 1000 rpm together with a 360° phase change. This generally means there are two systems in resonance at, or very near, the same frequency. An *antinode* is created that then shows up as an amplitude dip.

This principle can be used to control resonance. *Dynamic absorbers* are resonant systems deliberately added to a machine.

When eddy current or proximity probes are used for electric motor vibration monitoring, it needs to be noted that the probe cannot distinguish between actual shaft vibration and any runout or eccentricity of the shaft journal. Excessive mechanical or electrical runout (in the zone that the probe is studying) can cause the dip in Fig. C-300.

Instead of canceling each other, the runout(s) and shaft vibration may add vectorially to give a plot like the one in Fig. C-301. The result is that the viewer may think the machine is actually in much worse or much better shape than it really is. Thorough runout readings should be conducted, therefore, before response measurements are taken. Electrical/physical runouts should be eliminated before operation, if at all possible.

Figure C-302 may mean that the voltage pulse from the reference pickup is lost or has a very small amplitude near 6500 rpm.

Figure C-303 may mean that phase shift indications between 1° and 359° may cause extreme travel of the plotter pen.

When *filter-in/filter-out* amplitude versus rpm is applied to the vibration scans, further observations can be made.

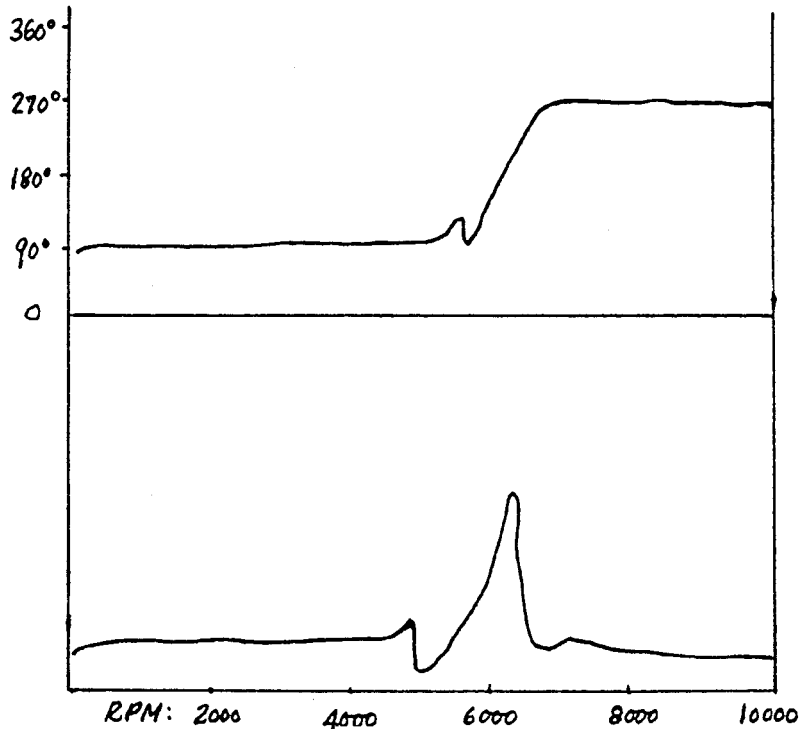


FIG. C-300 Excessive mechanical or electrical runout can cause dips noted here.[15]

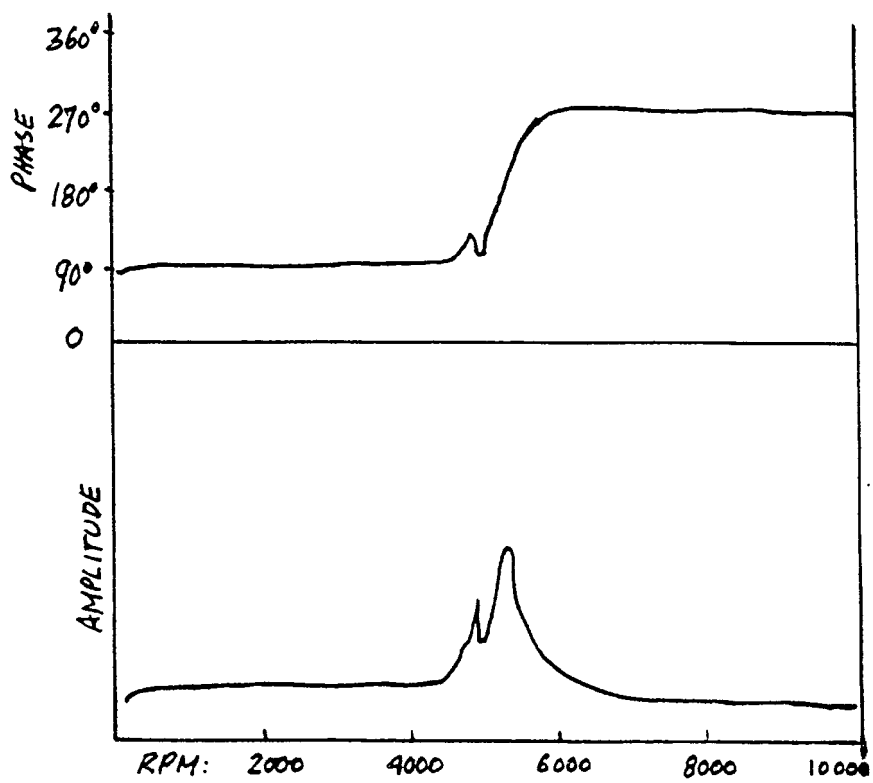


FIG. C-301 Runout and shaft vibration may add up vectorially.[15]

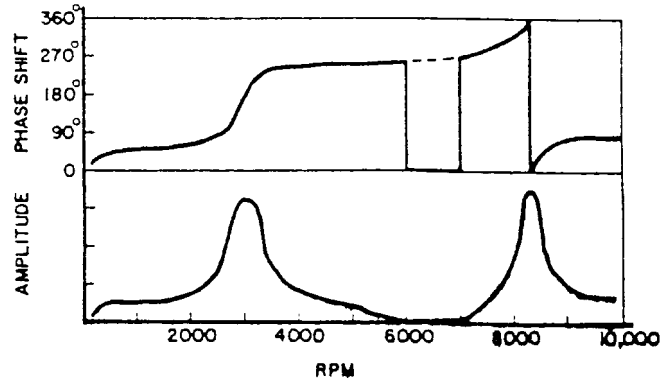


FIG. C-302 The "blank" section may mean the voltage pulse from the reference pickup is lost or has a very small amplitude near 6500 rpm.[15]

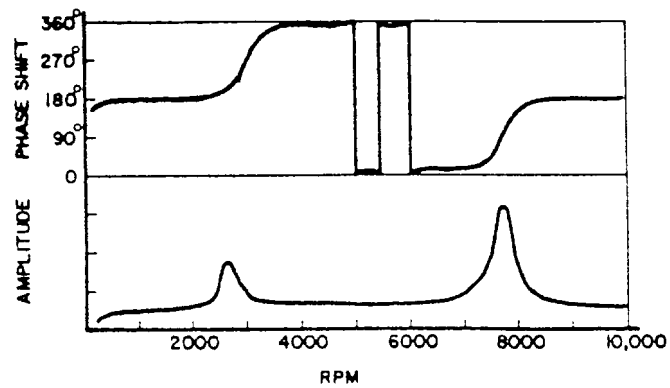


FIG. C-303 The "steps" noted may mean the phase shift indications between 1° and 359° can cause extreme travel of the plotter pen.[15]

Consider the example in Fig. C-304 for a pump operating at 3600 rpm. There are vibration peaks at 1× machine rpm and 2× machine rpm. If we assume the pump has a resonant frequency of 2000 cpm, on coastdown, the 2000 cpm resonance will be excited by what is probably unbalance at 1× rpm as it goes through 2000 rpm, and what is probably misalignment at 2× rpm when it goes through 1000 rpm.

The excited resonance does not appear in filter-in plots because in this mode the filter is tuned to a specific frequency, for example the 1× rpm component, and rejects all other frequencies. A filter-in amplitude versus rpm plot does not therefore give a total picture of a system. A filter-out plot is also required. (See Fig. C-305.)

For *phase analysis and in-place balancing, strobe lights* are commonly used. With orbits, damping is not taken into account. This method only works for lightly damped systems. Also, the balance weights are only correct for a certain speed. The optimum balancing plane considered is the plane containing the center of gravity of the rotor system or any plane that allows for the orbit to be shrunk to a spot.

The strobe light and analyzer frequency are triggered by the same vibration signal. Therefore, if the measured vibration frequency and the machine rpm (or some multiple) are the same, the strobe makes the rotor appear to stand still. If the vibration source is another part of the machine or from another machine, the strobe image may appear erratic, or the reference mark may appear to rotate slowly.

Some examples of when this can occur include:

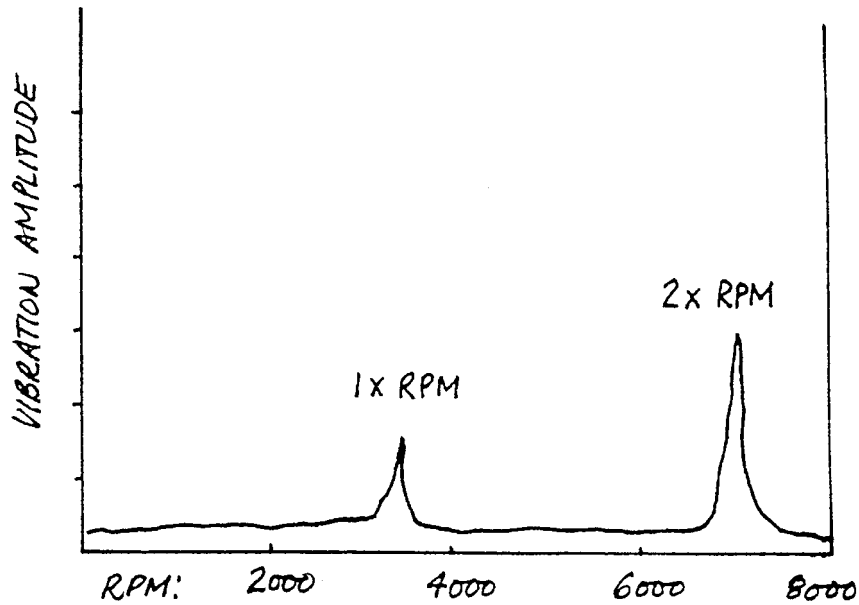


FIG. C-304 Pump at 3600 rpm, resonant frequency at 2000 cpm produce peaks of 1× rpm and 2× rpm on coastdown.[15]

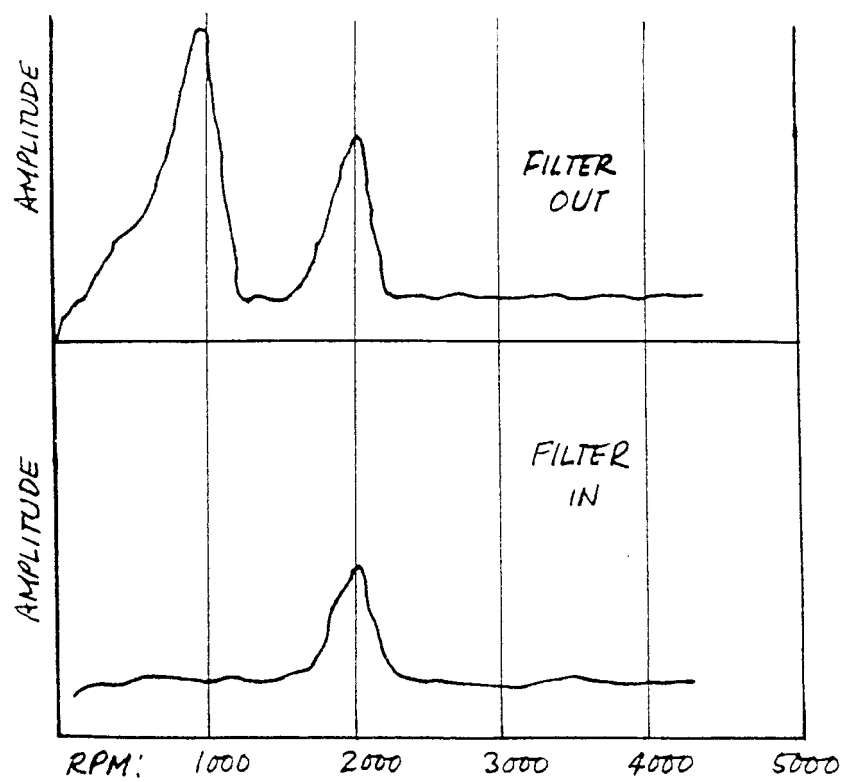


FIG. C-305 Filter-in plot may tune out all but the specific "tune-in" frequency.[15]

- On belt-driven fans, vibration may originate with the belts. These belts operate near but not quite at fan rpm, so they may indicate a marginally different speed when observed with a stroboscope.
- Different roll speeds in paper manufacturing machinery.

Amplitude versus phase (Nyquist plots). Nyquist plots depict the unbalance amplitude versus phase vector on polar coordinate paper. The data are collected with a spectrum analyzer and a tracking filter with polar amplitude and phase output signals for driving a standard X-Y recorder.

The Nyquist plot is a very limited technique in that it:

- Does not provide a speed reference to indicate the rpm at which resonance occurs, even though it does prove the presence and significance of resonant conditions during runup or coastdown, so machine rpm has to be manually recorded
- Only can measure the vibration component at 1× rpm

Besides these disadvantages, it has advantages, which include:

- An immediate indication of the unbalance vector without the need to compare separate amplitude and phase plots
- Elimination of confusing phase discontinuities that appear on Bode plots as phase changes from 0° to 360°
- Elimination of phase discontinuities that appear on Bode plots where the vibration amplitude has reduced to a low enough level that there is not enough vibration signal to maintain a phase indication
- Amplitude and phase data are obtained in a single runup or rundown of a machine with only one recording pen
- When proximity probes are being used, mechanical and electrical runout can be vectorially eliminated by recalibrating the recorder amplitude zero control to the origin of the polar graph while the runout vector is being displayed with the machine on slow roll, as in Fig. C-306
- Nyquist plots for each machine bearing can be compared to provide information on the mode shape of each resonance detected

IV. *Time waveform analysis* deals with time versus amplitude information. The vibration signal is applied to the scope vertical input. The vertical axis on the cathode ray tube is scaled in amplitude. The horizontal axis of the scope depicts time (seconds, milliseconds, and so forth).

The time waveform signal in Fig. C-307 shows transient vibration resulting from the startup of an electric motor. A vibration signal of this nature occurs for a very short span of time, but true peak amplitude is easily noted. Most vibration analyzers and meters damp the meter movement to limit erratic pointer movement. Often the damping slows the meter response enough that true peak amplitude of the signal is displayed at a reduced value.

Time waveform analysis picks up mechanical problems that may have identical frequencies but differ in terms of dynamic behavior.

Figure C-308 shows a case with a gear with one chipped or broken tooth that impacts the mating gear once per revolution. If there is also an unbalance problem present, that would cause a signal of a more regular sinusoidal shape.

Figure C-309 shows a vibration signal with and without electric current applied. The difference is the effect of electrical current.

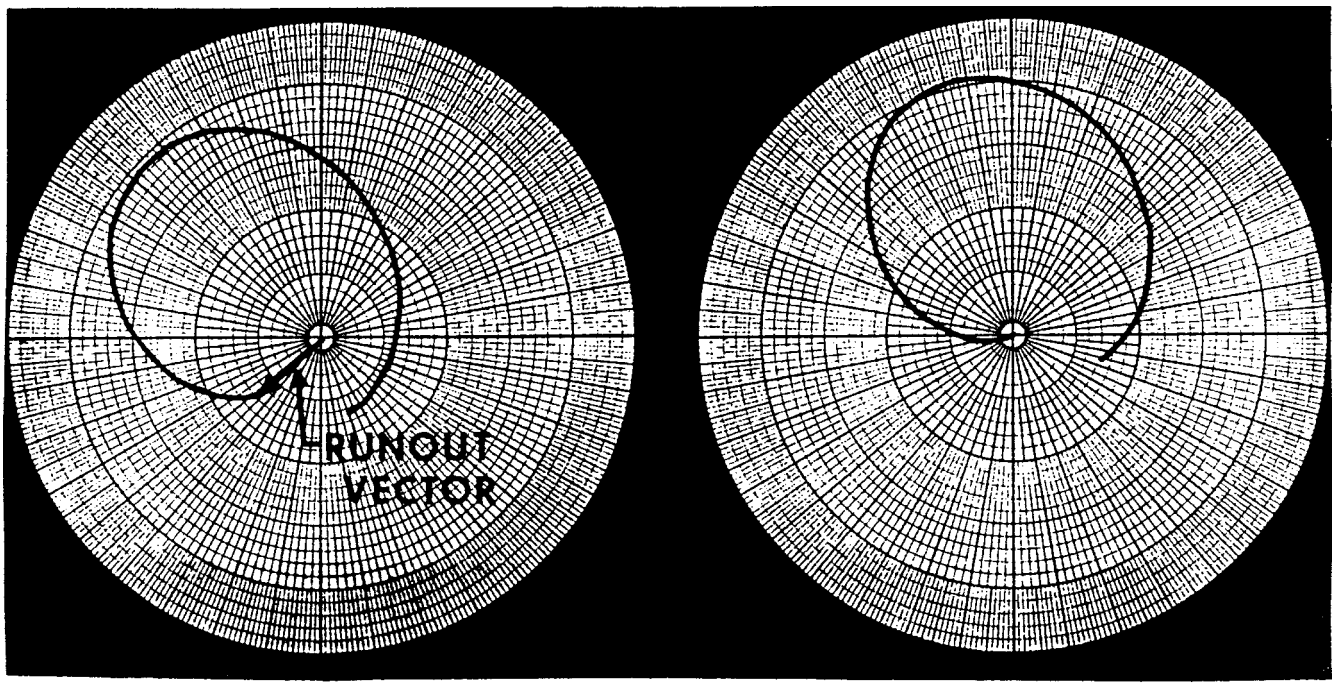


FIG. C-306 Shaft runout compensation.[15]



FIG. C-307 Transient vibration resulting from startup of an electric motor.[15]

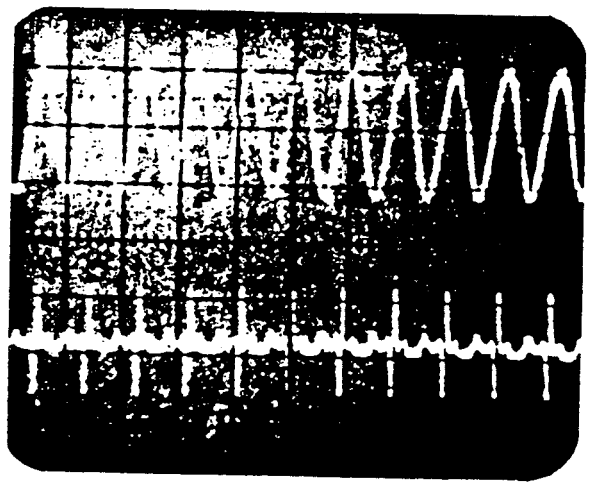


FIG. C-308 The difference between unbalance vibration and that generated by a defective gear tooth would be clearly indicated by observing the vibration time waveform.[15]

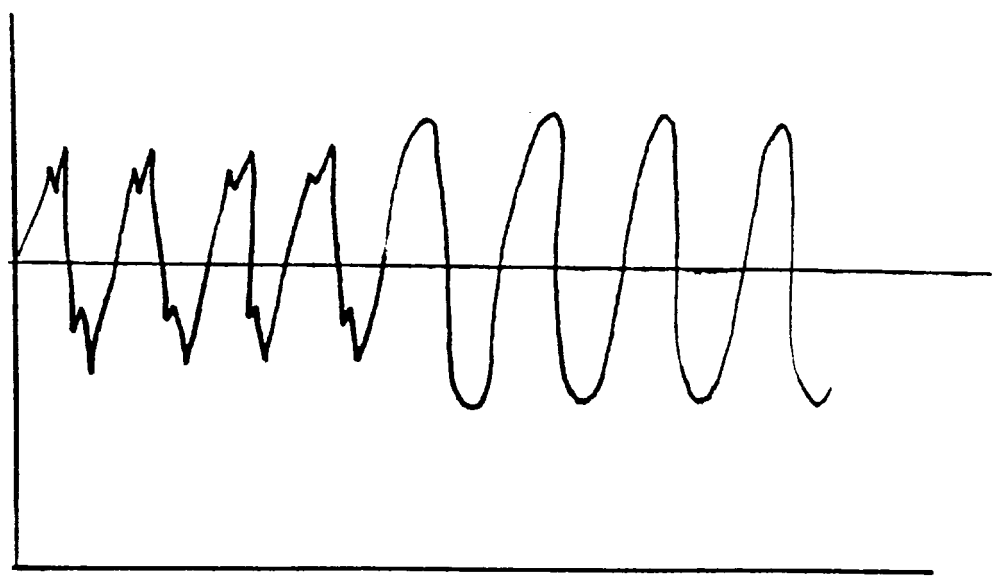


FIG. C-309 Vibration signal with and without electric current.[15]

Harmonic relationship of two separate frequencies is depicted in Fig. C-310.

Frequency analysis of a drive motor reveals 1× and 2× rpm peaks in Fig. C-311.

A lower frequency is read in Fig. C-312, as well as a higher frequency. The latter is due to 60 Hz or 3600 cpm forcing frequency. The two are not harmonically related.

Lissajous patterns are time waveform orbits. Dual-direction probes are used for this analysis. See Fig. C-313.

Orbits can be measured using stroboscopes. Three signal pickups are used to measure vibration in two mutually perpendicular planes (x and y axes, for example). The third probe is used to register the once per revolution reference point and is

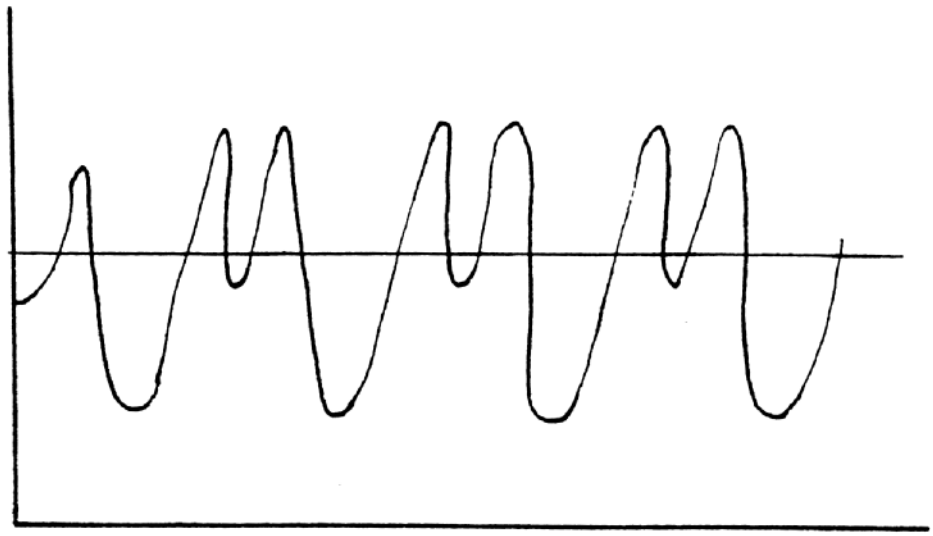


FIG. C-310 Harmonic relationships of two separate frequencies.[15]

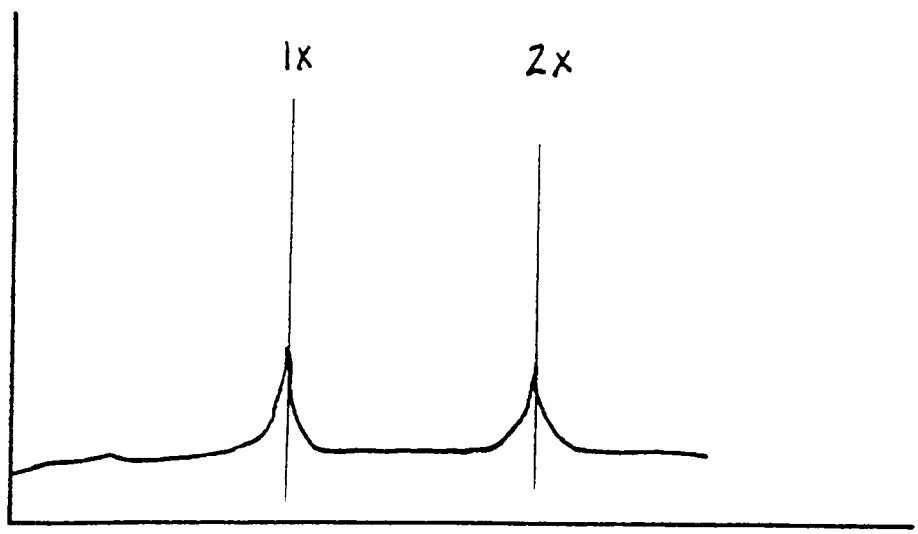


FIG. C-311 1× and 2× peaks in drive motor scan.[15]

called the key phasor (z axis). The key phasor appears as a bright spot on the screen. When the orbit is completely circular, the maximum vibration occurs in the direction of the key phasor. The correction mass is arrived at by trial and error. When the balance is perfect the orbit becomes a bright point again.

If the orbit is an ellipse, a perpendicular is dropped on the major axis of the ellipse to intersect its circumcircle. The intersecting point defines the phase angle of the force.

For speeds above the first critical, the key phasor will appear opposite the heavy point.

In Fig. C-314, the shaft motion is occurring at 0.5× rpm. Superimposing a reference signal on the Lissajous pattern aids in identifying the frequency of shaft vibration.

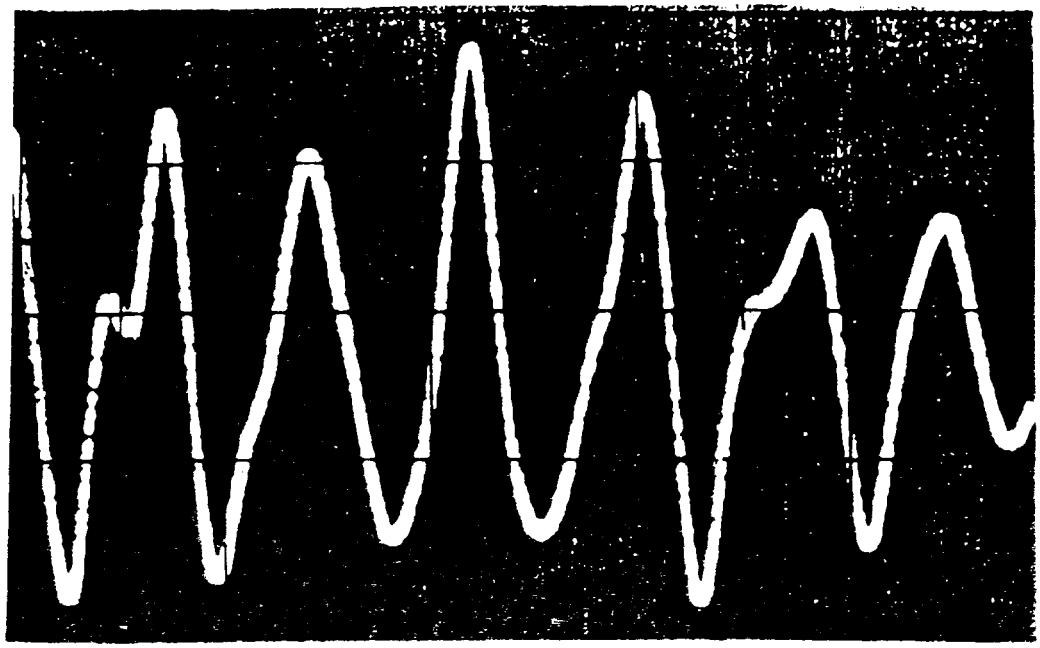


FIG. C-312 The higher frequency observed is due to 60-Hz forcing frequency and is not harmonically related to the lower forcing frequency.[15]

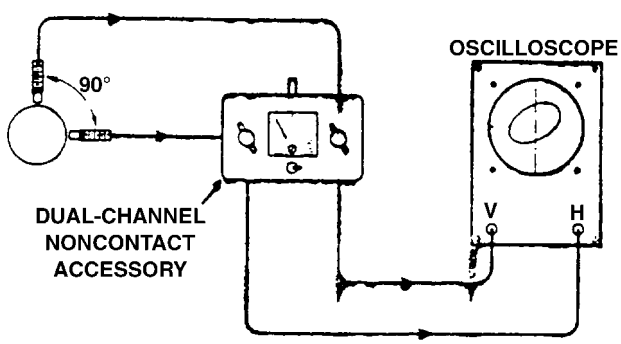


FIG. C-313 Dual-direction probe use.[15]

In the field, a handheld vibration probe can provide useful relative information even though these probes are more prone to direction sensitivity to some extent. Particularly revealing are the relative amplitudes of the triaxial readings.

For instance, note the 1× rpm vibration components for the following problems (see Figs. C-315 through C-318):

	Radial	Axial
Unbalance—1× rpm	High	Low
Misalignment—1× rpm	High	High
Bent shaft—1× rpm	High	High

Typical horizontal to vertical ratios are 2 : 1 through 5 : 1. If the ratio is much larger, resonance or looseness may be indicated. However with a vertical to horizontal

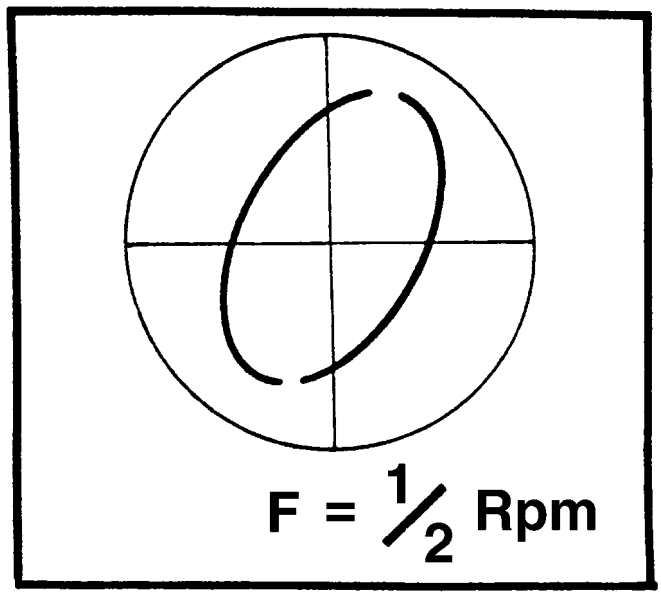


FIG. C-314 Superimposing a reference signal on the Lissajous pattern aids in identifying the frequency of shaft vibration. This shaft motion is occurring at $^1/_2$ rpm.[15]

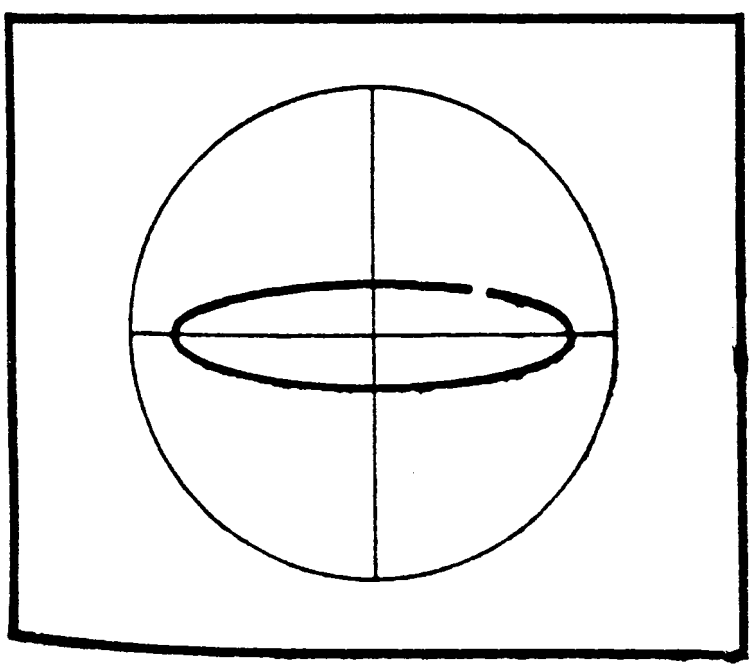


FIG. C-315 A highly elliptical pattern may indicate misalignment, bearing wear, or possible resonance.[15]

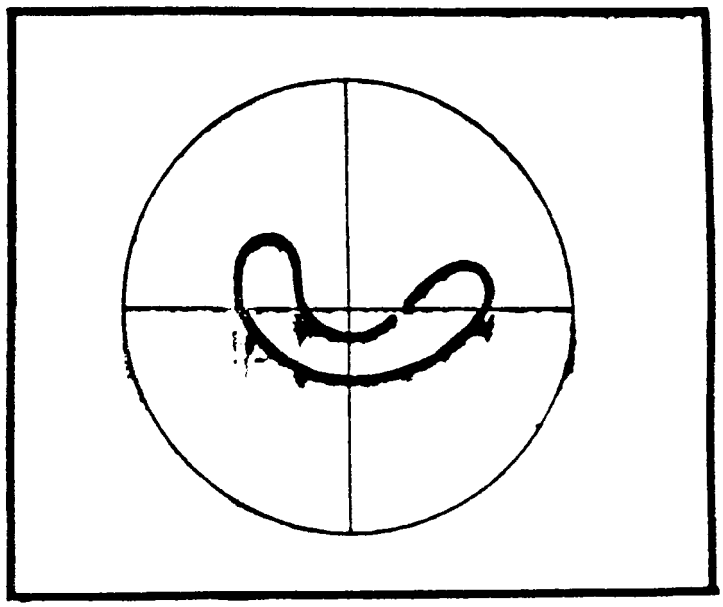


FIG. C-316 Misalignment will sometimes reveal a banana-shaped pattern.[15]

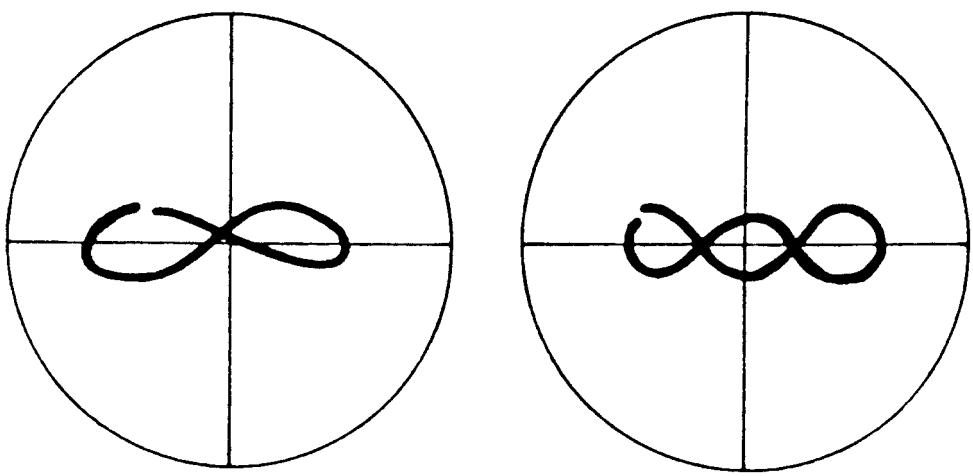


FIG. C-317 Additional misalignment pattern possibilities.[15]

reading ratio of greater than 1, a wiped bearing, loose bearing cap, or excessive clearance may be indicated.

Monitoring system packages (measurement equipment)

There are many kinds of monitoring packages available today. A good start for initial OEM product information is the Internet. However, to ultimately select an appropriate system, much more research is required on items such as product

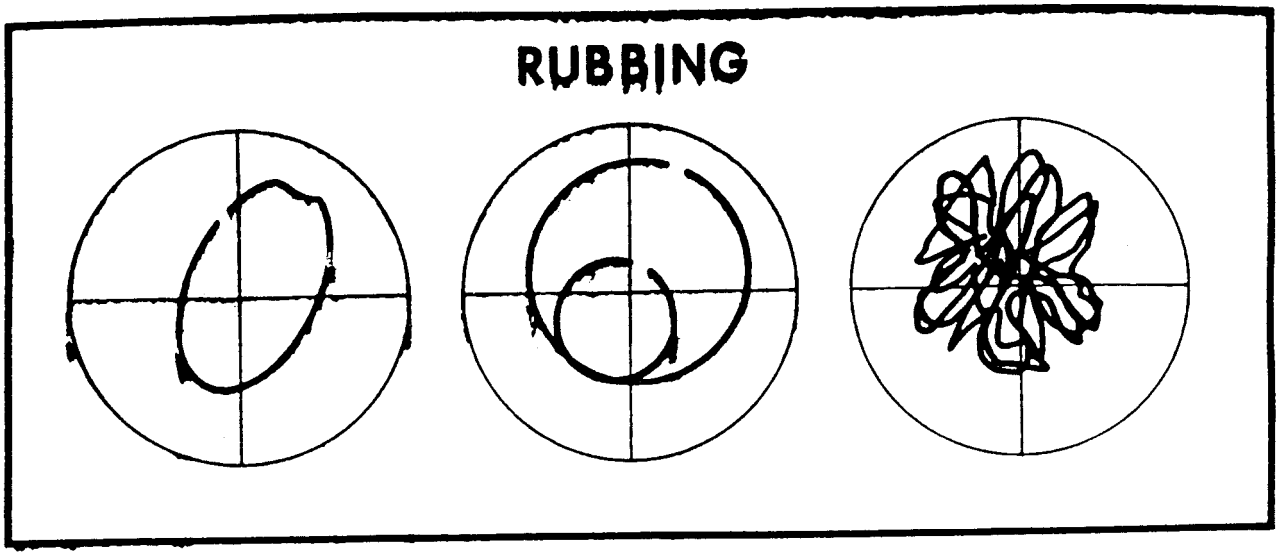


FIG. C-318 Typical Lissajous patterns caused by rubbing.[15]

ruggedness and frequency response range, as well as knowledge accumulation on the problems the end-user's machinery may give, other end-users' experience with vendors, after-sales service, and so forth.

Frequently, the OEM will have already chosen the monitoring package and included it, installed, with the price of the machinery system. If it does not perform to expectations or leaves out parameters that may later prove to be useful, monitoring equipment and systems or partial systems may be specified. An example of this is LCA counters installed as a retrofit on the Rolls Royce Olympus and Spey fleets that resulted in saving expected life on some critical parts by a factor of 200 percent or more.

Troubleshooting

A vibration signal is measured in mils, in/s or in/s^2 (displacement, velocity, and acceleration, respectively), by a vibration probe. The signal can be fed into a spectrum analyzer and split into different frequency components. There may be contributory components of:

- Less than 1× rpm, where rpm = rotational speed of the machine
- 1× rpm
- 2× rpm
- Number of blades, buckets, and vanes × rpm
- Number of gear teeth × rpm
- Other multiples thereof

Each of these components is a symptom of one or more machine problems. Vibration analysis personnel have tabulated this information in various formats. Engineers generally pick the one they like best for everyday reference.

Vibration severity charts have been drawn up by industry. The base parameter they use may be displacement, velocity, or acceleration. These tables should be used

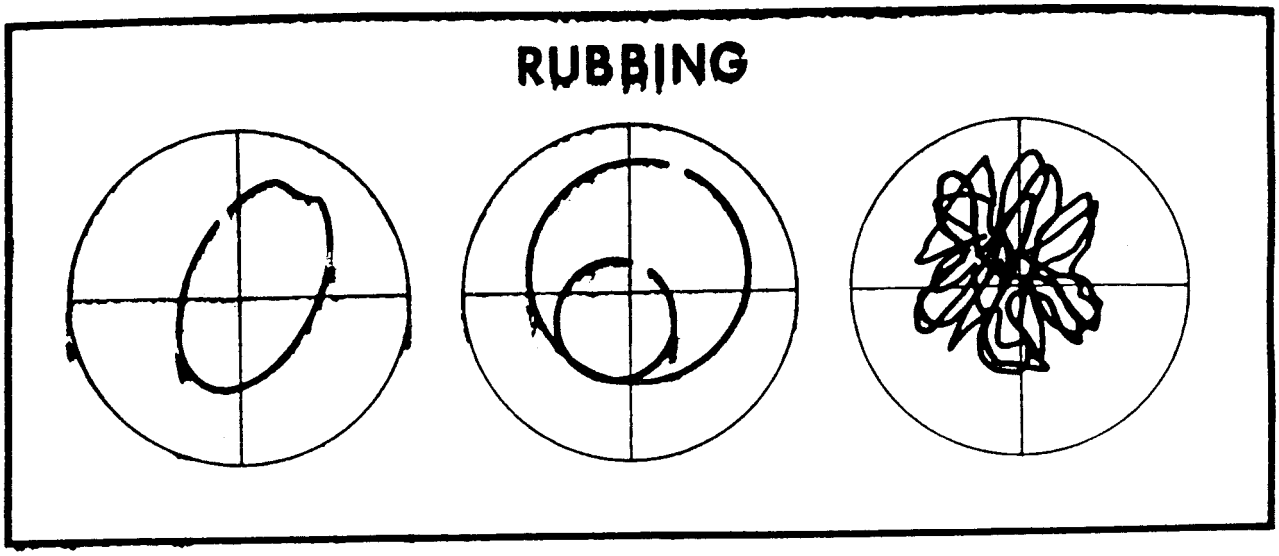


FIG. C-318 Typical Lissajous patterns caused by rubbing.[15]

ruggedness and frequency response range, as well as knowledge accumulation on the problems the end-user's machinery may give, other end-users' experience with vendors, after-sales service, and so forth.

Frequently, the OEM will have already chosen the monitoring package and included it, installed, with the price of the machinery system. If it does not perform to expectations or leaves out parameters that may later prove to be useful, monitoring equipment and systems or partial systems may be specified. An example of this is LCA counters installed as a retrofit on the Rolls Royce Olympus and Spey fleets that resulted in saving expected life on some critical parts by a factor of 200 percent or more.

Troubleshooting

A vibration signal is measured in mils, in/s or in/s^2 (displacement, velocity, and acceleration, respectively), by a vibration probe. The signal can be fed into a spectrum analyzer and split into different frequency components. There may be contributory components of:

- Less than 1× rpm, where rpm = rotational speed of the machine
- 1× rpm
- 2× rpm
- Number of blades, buckets, and vanes × rpm
- Number of gear teeth × rpm
- Other multiples thereof

Each of these components is a symptom of one or more machine problems. Vibration analysis personnel have tabulated this information in various formats. Engineers generally pick the one they like best for everyday reference.

Vibration severity charts have been drawn up by industry. The base parameter they use may be displacement, velocity, or acceleration. These tables should be used

as a guide only, as applications may change the relative severity of a vibration value. Each OEM has its own guide chart, sometimes for different models. An OEM may have one chart for installed field values for new machines and another chart for overhauled machines. Readings done in OEM test facilities may not provide good indicators of operational values at all. OEM tests are rarely run at full load or at full running speed (operational rpm).

Consider also a case where velocity measured is relatively low and in an area you might consider "safe." Corresponding acceleration values may be high however. This means that the "safe" velocity reading may deteriorate into a high reading. The normal rpm of the machine in question will affect the acceleration signal. Acceleration is a factor of speed squared. So, since force = mass × acceleration, if rpm is doubled, the force causing the vibration is quadrupled.

Imbalance. Typically, a high vibration signal at 1× machinery rotational frequency in the radial plane indicates imbalance. Note however that a 1× rpm signal may also be caused by misalignment.

Imbalance may be caused by factors internal to, and external to (bearings and so forth), the rotor.

Causes of imbalance. Some common reasons for unbalance to occur in an operating rotating machine are:

a. Lack of symmetry due to limitations in:
   - Design
   - Manufacture (including inclusions and voids in material. Blow holes and porous sections in castings are common causes); also, the fixtures used for machining and lathe processes are a common source of error
   - Reassembly during overhaul (e.g., cases where the axis of rotation as located by bearing journals is off-center from the axis that contains the center of gravity of the rotor assembly (this may also contribute to misalignment); looseness between mating parts, caused by, for instance, different diameters, can also be a problem source

b. Nonhomogeneous material (e.g., faulty raw material)

c. Eccentricity due to various problems including:
   - Bent shaft, sometimes caused by faulty runup or rundown procedures (may also cause a misalignment signal)
   - Machining errors

d. Misalignment of bearings (e.g., couple unbalance caused by machining errors—more material off at one end of locating bore diameter, more material off at 180° to that position at opposite end of rotor assembly); note that unbalance moments are calculated by multiplying the force causing them by the radius at which the force is at

e. Shift due to plastic (permanent) deformation of components

f. Temperature differential (thermal gradients)

g. Hydraulic or aerodynamic unbalance

Some of these contributors can be corrected before balancing procedures start, such as (a) through (c). An unbalance may be either continually distributed throughout the rotor or a few discrete forces that occur along the length of the rotor.

A typical unbalance signature plot is shown in Fig. C-319.

There are two kinds of rotors from the perspective of vibration:

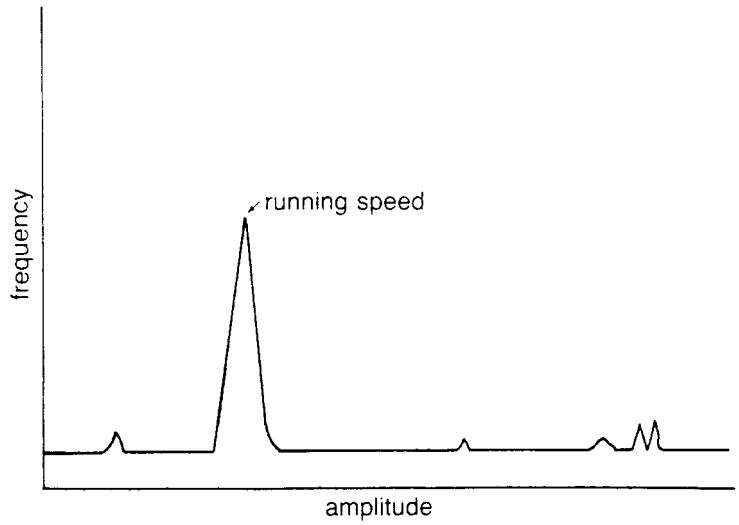


FIG. C-319 A typical unbalance signature plot.[13]

- Inflexible rotors
- Flexible rotors

An example of an inflexible rotor is a gear wheel. Most of the rotors we will consider are those that are not rigid but can be balanced in a low-speed balancing machine. These divide into rotors with:

- Axial distribution of unbalance known
- Axial distribution of unbalance unknown

Most turbomachinery rotors are flexible rotors. Flexible rotors are designed to operate at speeds above those corresponding to their first natural frequency of transverse vibrations. Two plane balancing is normally adequate to consider flexible rotors in instances when the vibration is not in phase with the force causing the vibration.

Balancing process. When a machine is overhauled, its rotor undergoes a balance procedure. The most accurate balance is obtained by balancing the rotor at the same speed at which it will operate. The only way this is possible, unless the rotor is a low-speed rotor, is if the rotor is balanced in a vacuum chamber. This is expensive. Generally, balancing the rotor at way below operational speed is acceptable if sufficiently fine tolerances are used to compensate for the fact that, at higher speeds, the unbalance force will increase as a function of the speed squared.

Besides inaccuracies brought on by balancing the rotor at other than operating speed, other inaccuracies can be caused by:

1. Fits and clearances in tooling
2. Changes of individual blades (moment weight differs from the blade being replaced)
3. Lack of or insufficient allowance for appropriate assembly-related components such as shaft keys

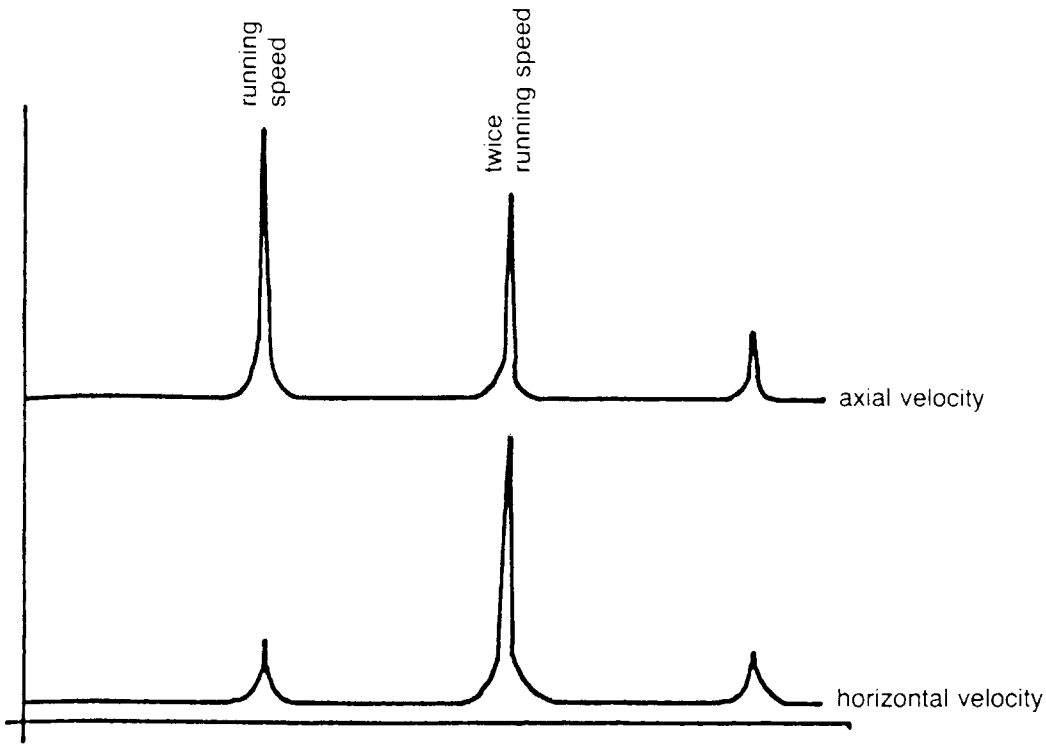


FIG. C-320 A typical misalignment signature plot.[13]

For precision-engineered, high-speed rotors it therefore is essential that:

- Customized tooling for the balance process be engineered with the same precision as the rotor components
- Individual blades be moment weighed
- Initial static balance procedures place suitable blades opposite each other
- Individual wheels (stages) are balanced separately before rotor assembly
- Single plane and two plane balancing is done when appropriate

The procedures are repeated for each rotor involved.

Misalignment. Misalignment commonly causes a signal of 2× rotational speed, or rpm. However, it has also been known to cause signals at 0.5× rpm or 1× rpm, and less frequently higher orders (3×, 4×, and so forth) as well. Imbalance has often been blamed for the 1× rpm signal that misalignment often causes.

A typical misalignment plot is shown in Fig. C-320.

How common is misalignment? Like everything else, it depends on the quality of your equipment, the instructions provided with it, the training and turnover of maintenance staff, and so forth. The reverse dial method is commonly used in the field today. "Cold" use of the method is the most common. However, if the machine train is run up and the readings taken quickly on shutdown, the "hot" readings would apply more to a running situation.

There are also optical and laser alignment techniques that have been proven successful. They recover initial cost of overhead and training quickly, particularly when used with large complex turbomachinery. This course, however, is not meant to cover alignment techniques, which are a separate course on their own.

When monitoring the results of misalignment on a bearing housing with a vertical and a horizontal reading, the size of the vertical versus the horizontal reading may be a factor of frame damping in either direction, as opposed to an indicator of how the misalignment-causing force actually subdivides into components. The nature of the imbalance vector is also influenced by what kind of misalignment is involved—angular (shafts of driver and driven at an angle to each other) or parallel (centerline of shafts of driver and driven not in the same line). For angular misalignment, the axial vibration component is higher than parallel misalignment.

Whether other vibration-causing forces are involved also plays a part. If the coupling between the driver and driven equipment has teeth, a component of vibration equal to the number of teeth × rpm might show up. In other words, separating the components due to misalignment in a vibration spectrum might not be as easy as with imbalance.

In other words there is no anticipated relationship between the x and y components of misalignment-caused vibration. With imbalance one might expect them to be roughly the same. Nonetheless a large (2× rpm) component may be a good indicator that one needs to check alignment on the train in question, depending on the phase angles involved.

Part of the reason that misalignment vibration may be hard to pinpoint a source for is that it may be caused by a combination of the following causes and a few others:

1. Temperature growth of a machine or faulty growth calculations
2. Static pipe strain and pipe strain increases under running conditions (pipes deflecting, growing; elbow movement during operation)
3. Coupling wear of gear teeth or girds
4. Temperature growth of concrete or metal bases
5. Hydraulic thrust at pipe elbows
6. Foundation settling

It is important not to blame the mechanics and millwrights if it appears that they reported perfect cold alignment readings only to have severe misalignment when the machine was run. It is at this point that the engineer needs the mechanics more than ever.

A phase angle reading will help confirm the presence of misalignment. With coupling misalignment the axial and radial readings at either end of the machine are generally exactly 180° out of phase.

For a bent shaft, for instance, the radial readings would be in phase and the axial readings would be 180° out of phase. For both a bent shaft and misalignment, the radial and axial readings are relatively high.

Subsynchronous frequency orders. Previously, a common problem was oil whip or whirl. Contemporary design makes this problem a rarity today. Oil whirl occurs as a consequence of the rotational center of a rotor being different from its geometric center. This means there is a difference in the thickness of the oil fill in bearings. Because of a narrower gap, the oil pressure builds in this area until it lifts the shaft. The pressure escapes, the shaft “falls” back into place, and the oil pressure build-up starts again.

Other problems can cause subsynchronous vibration, including clearance problems with bushing and labyrinth seals. Aerodynamic components such as impellers, shrink fits, and shaft hysteresis can also cause problems.

Figures C-321 through C-324 illustrate the occurrence of such a problem. In Fig. C-321, operating machine speed is 20,000 rpm, 500 psig suction pressure, and

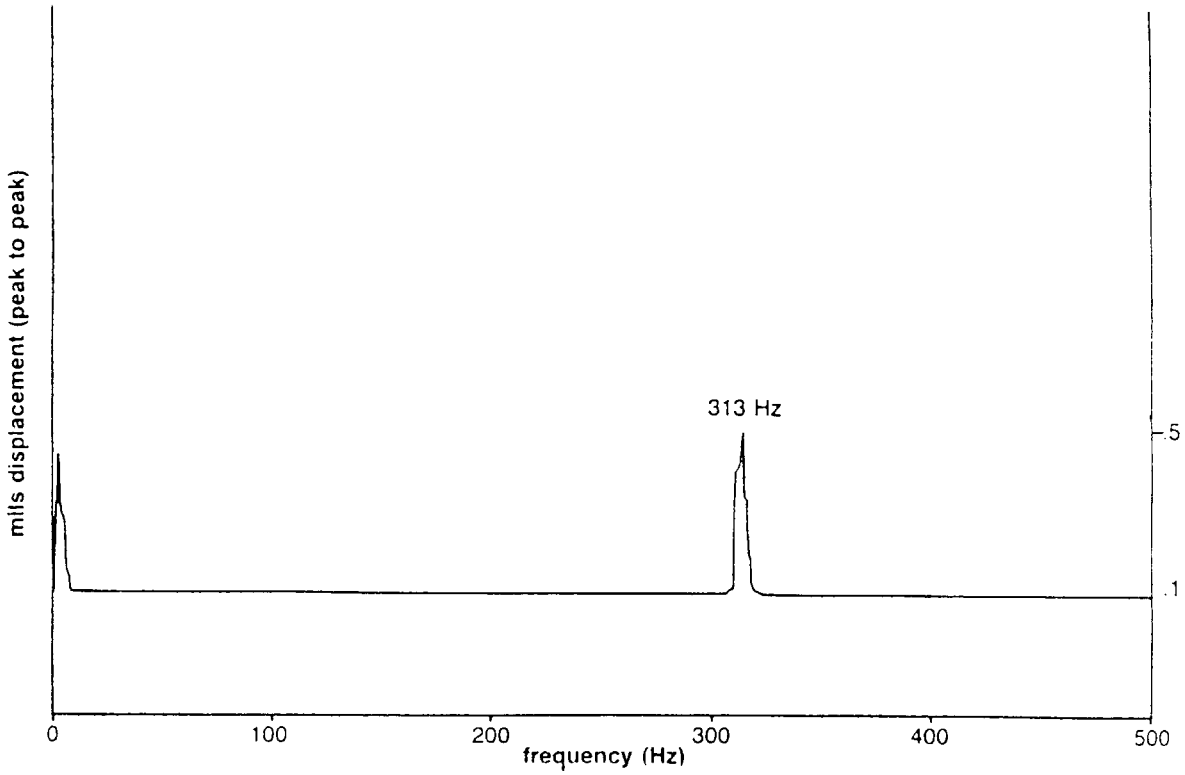


FIG. C-321 Vibration spectrum (rpm = 20,000, P_d = 1200 psig).[13]

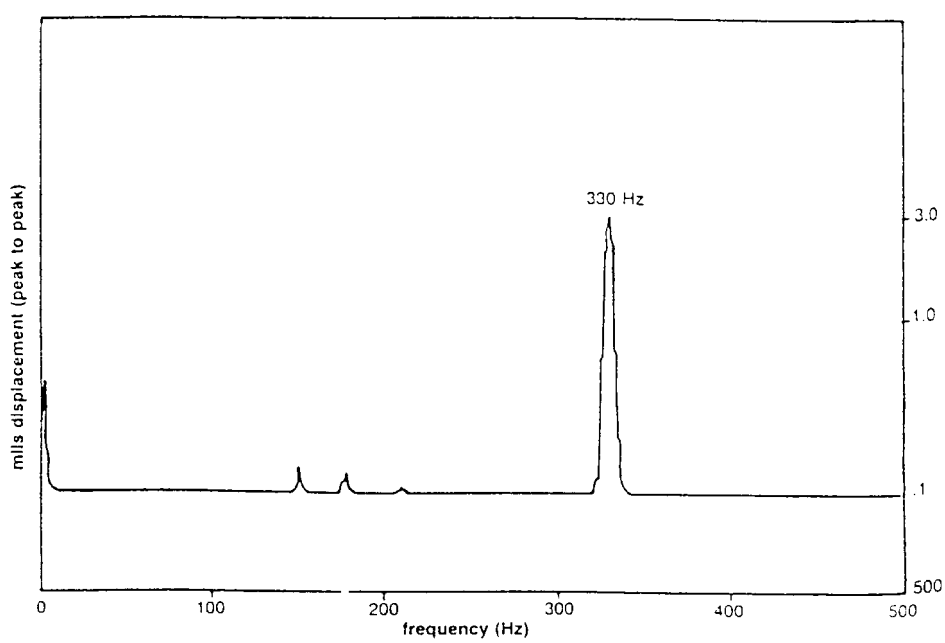


FIG. C-322 Vibration spectrum (rpm = 20,000, P_d = 1250 psig).[13]

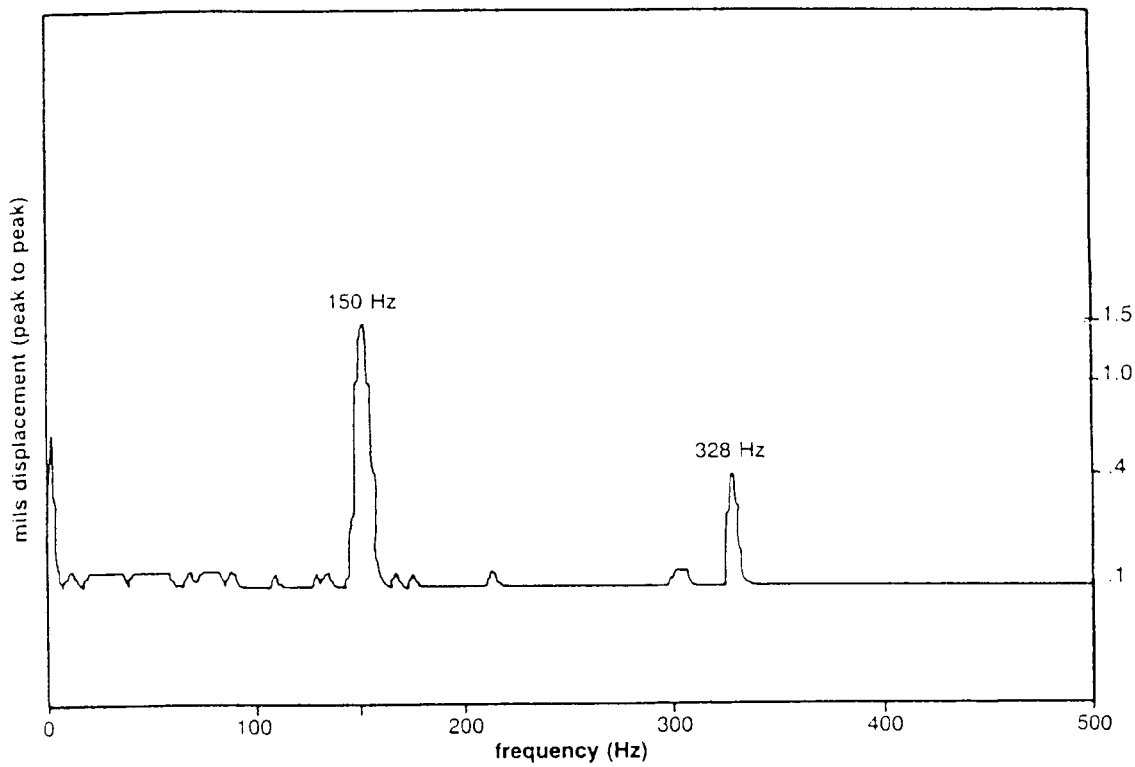


FIG. C-323 Vibration spectrum (rpm = 20,000, P_d = 1270 psig).[13]

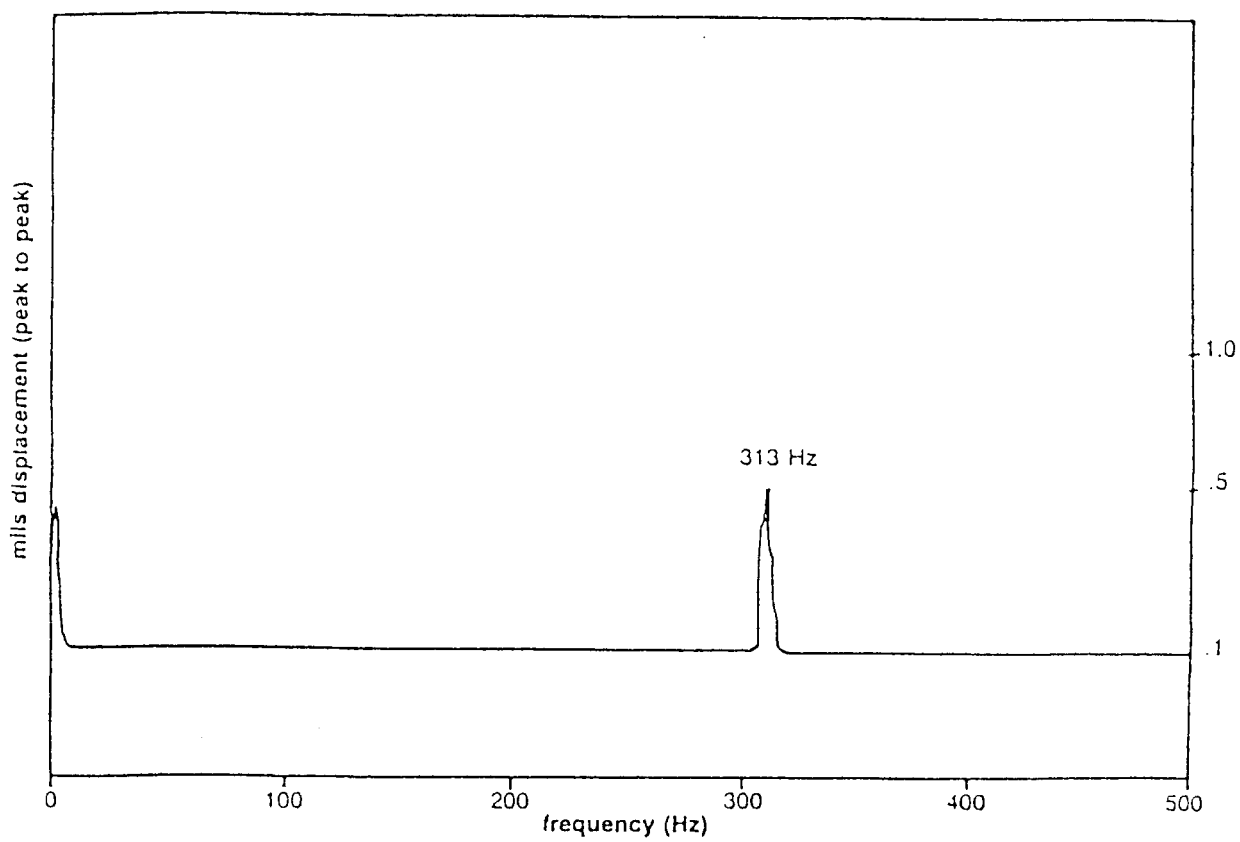


FIG. C-324 Vibration spectrum (rpm = 20,000, P_d = 1320 psig).[13]

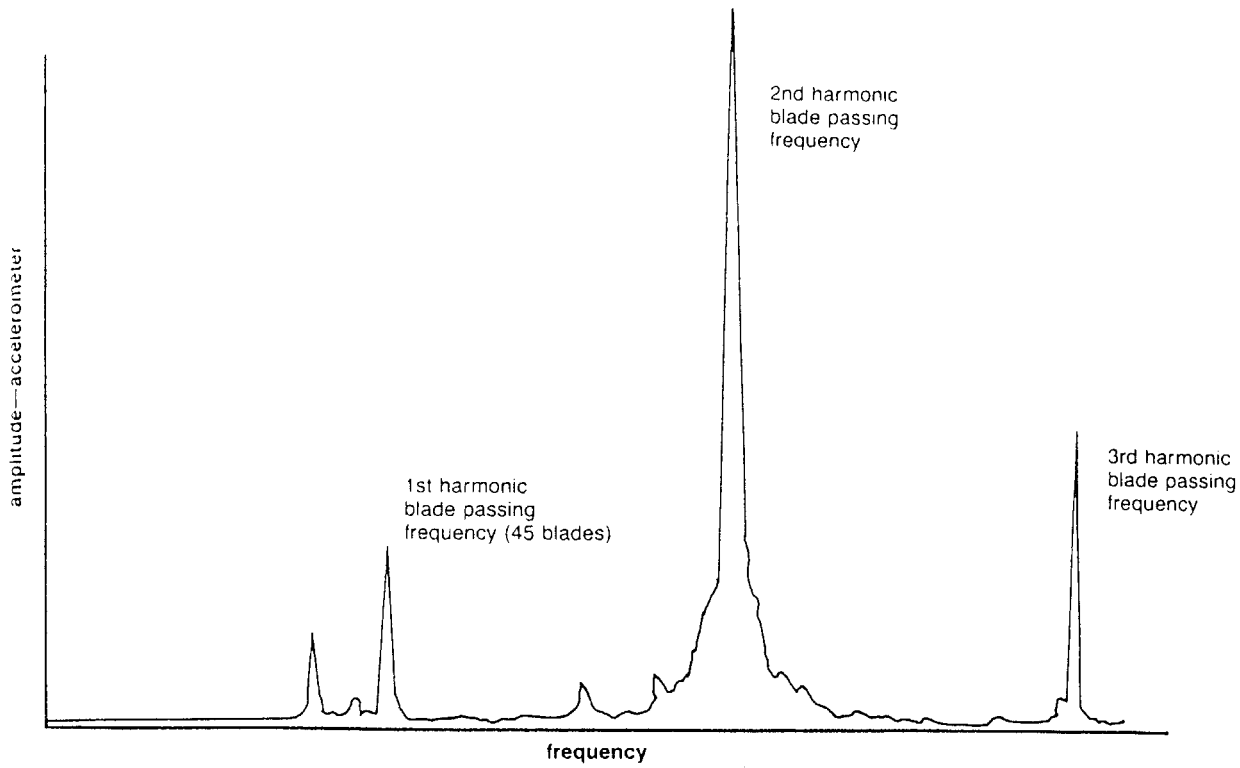


FIG. C-325 Axial-flow compressor spectrum showing blade-passing frequency.[13]

1200 psig discharge pressure. Note a synchronous peak of 0.5 mil at 20,000 rpm (possibly due to unbalance).

In Fig. C-324 machine rpm and suction pressure stay unaltered. Discharge pressure has been raised to 1250 psig. Now a 0.2 mil subsynchronous component shows up at 9000 rpm. Frequently such components may be intermittent and hard to capture without the use of the peak hold mode on the analyzer.

In Fig. C-323 with all other conditions remaining unchanged, it is noted that just a 20 psig increase in discharge pressure raised the 9000 rpm component from 0.2 mil to 1.5 mil.

It took an increase in suction pressure by 50 psig while maintaining the same discharge pressure to allow the unit to regain stability.

This shows the importance of keeping track of process conditions in addition to mechanical items that change within the machine system.

Other frequency orders. When originated within the machine, these are generally related to meeting contact surfaces.

Blades. For instance, blade tip rub would show up a signal of "number of teeth in the 'contacting' stage × rpm."

Figure C-325 shows an accelerometer's signature from an axial flow compressor with strong frequency component of the first, second, and third harmonic of the fifth-stage stator blade row. An inspection of this stator row indicated cracks caused by high-cycle fatigue.

Gears. A frequency of "number of gear teeth × rpm" may indicate resonance with the natural frequency of the concrete foundation, a fabricated base, or supporting beams.

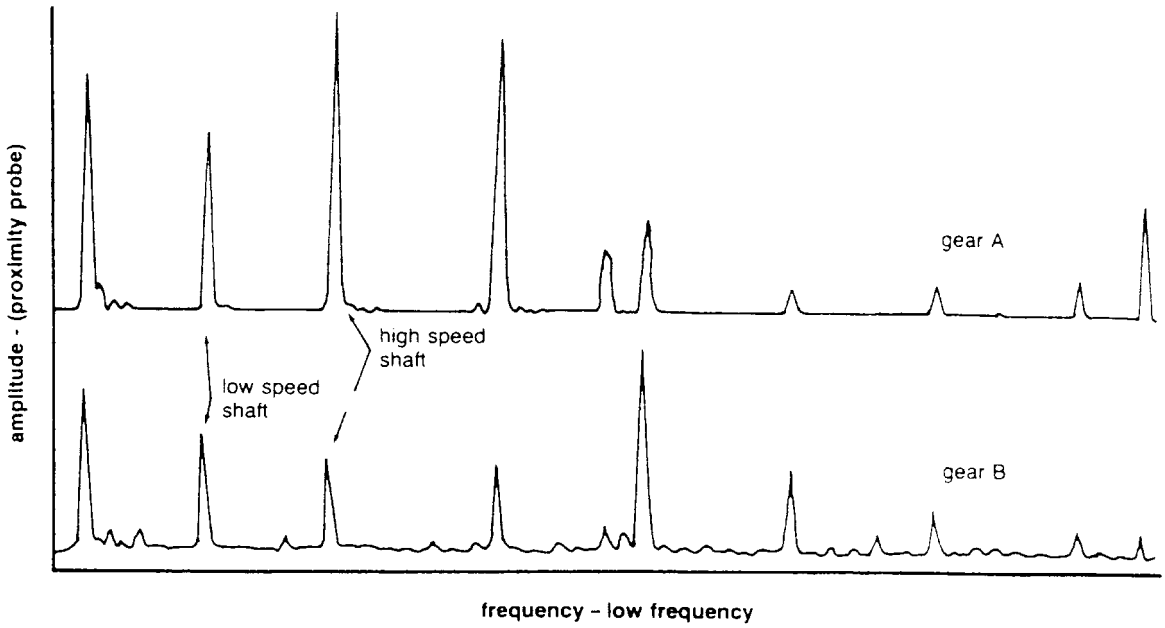


FIG. C-326 Gearbox signature (low-frequency end).[13]

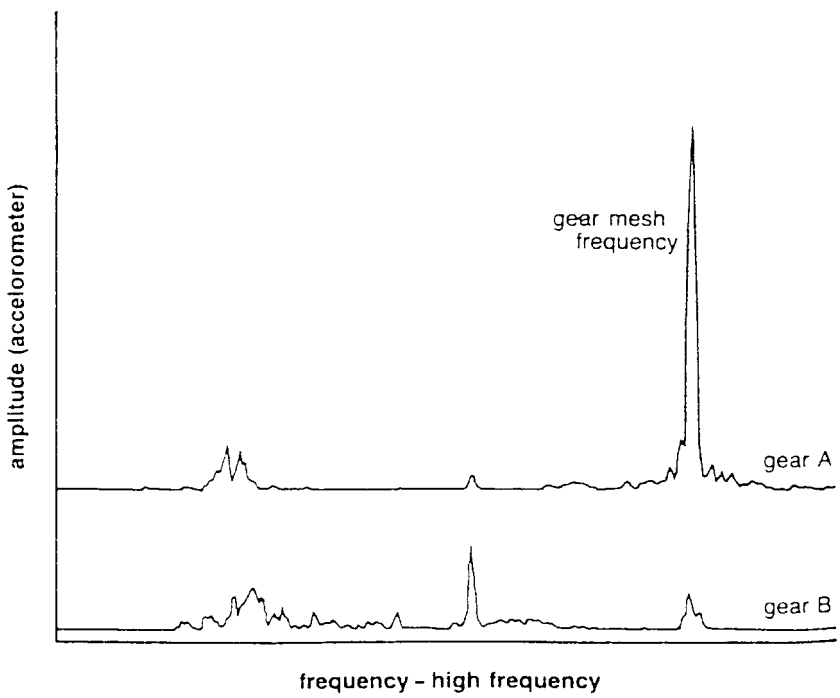


FIG. C-327 Gearbox signature (high-frequency end). Potential for damaged tooth.[13]

Figures C-326 and C-327 are typical of the kind of information that can be provided by accelerometers. These data would not be possible in the low-frequency spectra provided by proximity probes. Figure C-326 shows two gears in good condition (accelerometer is at the low-frequency end of the gearbox).

Figure C-327 shows a problem with gear A that may be a chipped or cracked tooth.

A frequency of 1× rpm may be observed when a 1× rpm signal elsewhere in the machine (e.g., an unbalanced orbit) is transmitted to a gear and gets it to run eccentrically. This eccentric running in turn produces a 1× rpm signal with a high amplitude in the direction of the imaginary line joining the centers of the two mating gears. At 90° to this position, the 1× rpm signal would be lower, producing an orbit that is a flat ellipse.

This orbit in turn magnifies any resonance to gear mesh frequency if present.

When troubleshooting resonance, remove sources for lower order frequencies first, to help analyze the higher frequency vibration.

If gear misalignment occurs, a vibration at 2× rpm would probably show up. To prove misalignment, use Prussian blue to coat the gear surfaces and run for a few minutes. A clear indication of contact surfaces and wear pattern shows when the gears are examined.

Loose assembly. Looseness of a part generally causes vibration at 2× the frequency of the rotating part. To visualize this, consider the case of a loose machine base and compare it to a bench that has two uneven supports. First one support touches in a cycle, then the other. So the frequency is 2× rpm.

Drive belts. If drive belts are loose, a vibration is caused. This can be observed with a strobe shone on a reference mark on the belt. This is often confused with an unbalanced belt and belts are thrown away unnecessarily.

Sources of a vibration-causing force from outside the machine include:

1. Piping stresses—static, cantilevered
2. Foundation problems
   - Foundation settling
   - Frost melting/permafrost problems
   - Moving soil (muskeg or other shifting soil insufficiently removed)
   - Foundation inclusions (grout problems, soft feet, and so forth)
3. Extreme climatic change

Contingency measures in a mature model (old or approaching scheduled overhaul) and retrofit. In the case of a mature operating model, the problems prevalent with a new train or prototype model can give way to those posed by:

1. Changing field composition
2. Changing environmental regulations: new burner designs, water and steam injection to reduce NO_x, and so forth

These changes may or may not have anything to do with the aerodynamic and mechanical behavior of the machine in question. At any rate, they will have to be analyzed as they come up and an attempt should be made to approximate some budget figures for retrofit items that are anticipated due to tightening legislation.

Vibration signatures may also be a good indicator of when a machine is approaching the point of requiring overhaul. See Figs. C-328 and C-329. Figure C-328 compares baseline signature with one taken after two years of operation. The increase in high-frequency levels was found attributable to blade flutter caused by cracked blades.

Figure C-329 is similar. It shows an increase in the component due to one stator stage's resonant frequency, indicating high blade flutter, that was found to be caused by cracks in that stator.

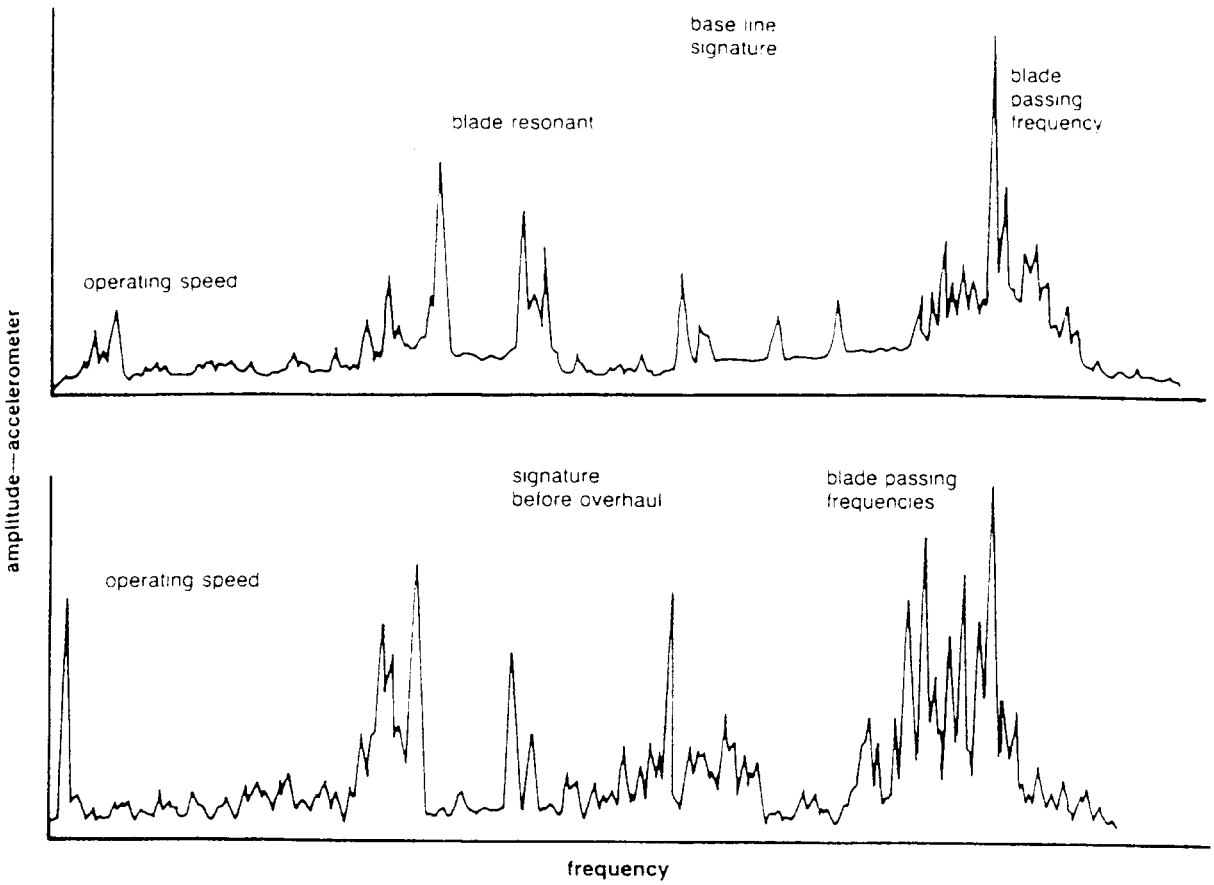


FIG. C-328 Machinery analyses showing comparison of baseline signature to signature before overhaul.[13]

Precautions on new turbomachinery. To help avoid problems on new turbomachinery:

1. Ask that vibration specifications be included with preliminary information—prior to formal quote request stage.
2. Make sure vibration specifications include data on allowable vibration levels, types of probes to be used, and whether the probes are seismic or proximity.
3. Arrange to be present for in-factory balance tests as well as final (full or partial load) tests. Final assembly should also be witnessed if at all possible, particularly in the case of a prototype. Baseline signatures should be taken.
4. Be aware of machinery shipping plans.
5. During commissioning, run the drive unit alone and take vibration readings.
6. Run the drive unit with the machine coupled, but not in complete process loop.
7. Record readings during machine rundown. Audit surrounding systems, piping, and so forth.
8. Before the machine arrives on site, check piping, grouting, and so forth.
9. Do hot and cold alignments on the train.
10. With electric motors, ensure half key is allowed for during balance.

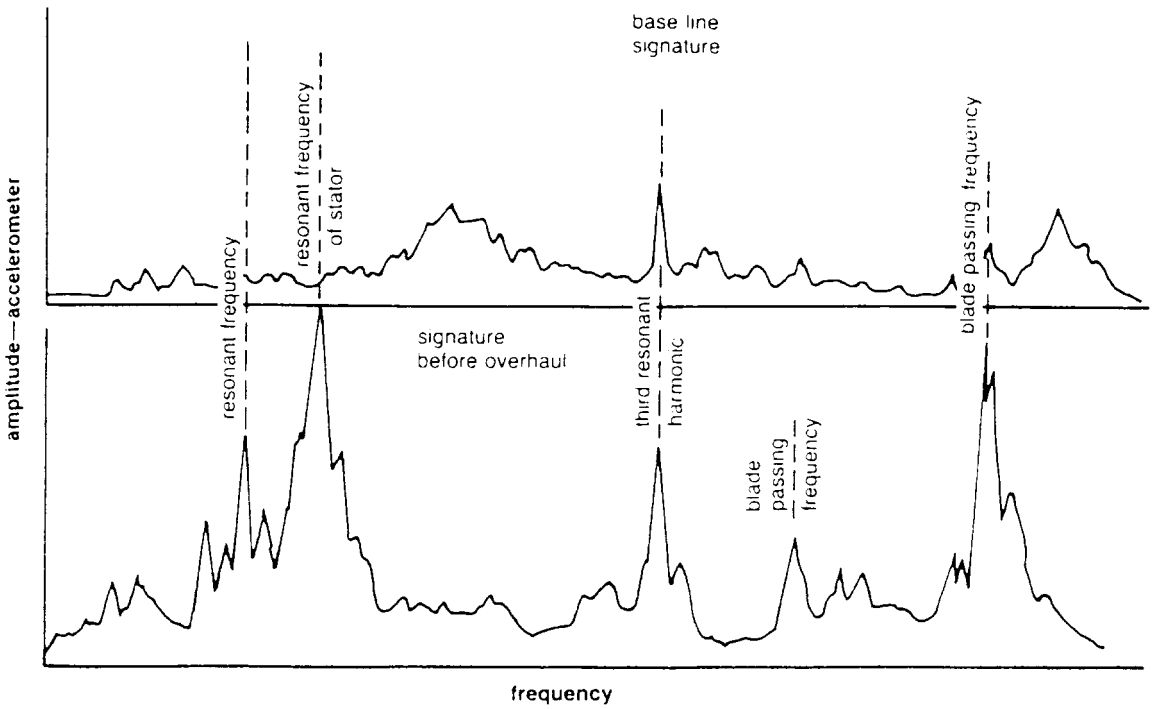


FIG. C-329 Machinery analyses showing comparison of baseline signature to signature before overhaul.[13]

Troubleshooting philosophy

In any problem situation, the indicators may include vibration readings as well as gas path parameters. There may also be other indicators, such as bleed valve behavior, bearing cavity temperature, and so forth. These other measured quantities may not be conveniently available (although they may be monitored at some intervals) in a nonexpert comprehensive system. However, most problems—over 95 percent for nonprototype applications—that occur with turbomachinery can be solved with good VA and PA data.

A basic philosophy for troubleshooting is as follows:

1. Spend money on diagnostic equipment only if you can use and interpret the data. If you are new to troubleshooting, VA, and so forth, get help, but with a view to learning how to do all this yourself. A good troubleshooter has the right
- Physical equipment
- Mental knowledge
- Relevant training

2. One of the things that is really useful to have is a portable spectrum analyzer. If you have a vibration system already installed, but you need to see if it would benefit you to
- Retrofit more probes on your installation
- Work out how many you need for similar future installations

then a portable system with:
- A portable probe or probes (velocity or acceleration transducer)
- A spectrum analyzer, including storage capacity to store successive plots and a chart recorder to make a hard copy of the spectrum

is very useful. You can now build up your own store of information on every item of machinery you are responsible for.

3. You should study the instrumentation—OEM-supplied or otherwise—on your installation and learn about its accuracy, usefulness, and ability to have its signal fed into a retrofitted PLC (programmable logic controller) or a computer. Consider what additional instrumentation, if any, might be useful.

4. Concentrate on gas path monitoring parameters as these are the most useful. Generally, most systems, however basic, supplied by an OEM will have enough data for you to fit a PA system. This is useful for

- Determining the health of the gas path
- Helping diagnose failed blades, combustion liners, crossover tubes, and so forth
- Determining when a module (compressor or turbine) needs to be washed
- Determining if premature shutdown/maintenance is required

For further discussion of PA systems, *see* Life-Cycle Assessment (LCA).

5. Consider what the return on investment (ROI) might be if you were to get a comprehensive online (perhaps real-time) condition-monitoring system. Consider also if it would ever make life trouble-free for the operator.

Problem diagnosis. Let us assume that a problem has occurred. Ask these questions with reference to the occurrence:

- What?
- When?
- How?
- Why?

Then ask:

- What needs to be done?
- When?
- How?
- Why?
- Will this affect anything else?
- What is the cost of doing nothing?
- How much production will we lose meanwhile?
- Can we correct anything else while we correct this problem?
- What can we learn for future installations?

Summary rules

1. There is no one consistently right answer for any symptom in condition monitoring.
2. Separate the elements of plant, process, and personnel.
3. Do not spend money to get more data than you can thoroughly understand or be taught to understand.
4. Fully automated intelligent systems might not be worth the money.
5. When you think you know all the answers, see rule 1.

TABLE C-21 "Normal" Vibration Levels on BRG Housings in IPS (Peak) Highest Noted on Smooth Machine[20]

Machine Type	1	2	3	4	VP1	VP2	GMF1	GMF2	BP1	BP2
Blowers (6000 rpm max)	.05	.02	.01	.01	.04	.01				
Horizontal centrifugal compressors	.05	.02	.01	.01	.04	.01				
Barrel compressor	.03	.01	.005	.005	.05	.005				
Gears Parallel Shaft	.1	.05	.02	.02			.05	.02		
Epicyclic	.05	.02	.02	.02			.05	.01		
Steam turbines	.1	.02	.02	.02					.05	.01
Gas turbines/axial compressor	.2	.02	.01	.01					.05	.01
Pumps	.1	.05	.01	.01						
Motors	.1	.1	.05	.05						
			LP1	LP2	LP3	LP4	LP5			
Screw compressor	.1	.01	.1	.1	.1	.05	.05			

VP = vane pass.
BP = blade pass.
GMF = gear mesh frequency.
LP = lobe pass.

Tables C-21 through C-27 give commonly accepted guideline limits for vibration readings. These limits apply to turboexpanders and all associated machinery in the process train.

Figures C-330 through C-333 are a few of the diagnostic charts available in industry. They are not new but then neither is much of the machinery being monitored in older plants. There is no hard-and-fast rule about which is best. Knowledge of a particular machine and process determines which are appropriate.

Guide. Note that the limits expressed in Fig. C-332 are based on experience in refineries. This guide reflects the typical proximity probe installation close to and supported by the bearing housing and assumes the main vibration component to be of 1× rpm frequency. The seemingly high allowable vibration levels above 20,000 rpm reflect the experience of high-speed air compressors (up to 50,000 rpm) and jet engine–type gas turbines with their light rotors and light bearing loads.

Readings must be taken on machined surfaces with runout less than 0.5 mil up to 20,000 rpm and less than 0.25 mil above 12,000 rpm.

Warning. Judgment must be used especially when experiencing frequencies in multiples of operating rpm on machines with standard bearing loads. Such machines cannot operate at the indicated limits for frequencies higher than 1× rpm. In such cases, enter the graph with the predominant frequency of vibration instead of the operating speed.

TABLE C-22 Maximum Allowable Vibration Limits on BRG Housing in IPS (Peak) for Operation Up to Earliest Possible Corrective Shutdown[20]

Machine Type	1	2	3	4	VP1	VP2	GMF1	GMF2	BP1	BP2
Blowers (6000 rpm max)	.5	.4	.25	.25	.1	.05				
Horizontal centrifugal compressors	.25	.2	.15	.15	.1	.05				
Barrel compressor	.15	.1	.1	.1	.05	.025				
Gears										
Parallel Shaft	.25	.2	.15	.15			.1	.05		
Epicyclic	.15	.1	.1	.1			.1	.05		
Steam turbines	.25	.2	.15	.15					.1	.05
Gas turbines/axial compressors	.5	.4	.25	.25					.1	.05
Pumps	.25	.2	.15	.15	.1	.05				
Motors	.25	.2	.15	.15						
			LP1	LP2	LP3	LP4	LP5			
Screw compressors	.25	.2	.2	.2	.2	.2	.2			

VP = vane pass.
BP = blade pass.
GMF = gear mesh frequency.
LP = lobe pass.

Introduction to rotor dynamics: vibration theory

The two main categories of vibration systems are:

- Forced systems
- Free systems

A free system operates under forces that are inherent in the system, so it operates at one or more of the natural frequencies of the system.

A forced vibration system operates under the influence of an external force impressed on the system. Vibration occurs at the frequency of the exciting force, which has nothing to do with the natural frequencies of the system. When the exciting force frequency and the natural frequency coincide, we have what is termed resonance. Large and dangerous amplitudes occur. Fortunately, practical systems have damping, which includes frictional forces.

A degree of freedom is the term given to an independent coordinate that describes the motion of the system. Figure C-334 depicts a one degree of freedom system: the classical spring mass system.

If a system has two or more degrees of freedom, then frequency and amplitude have no definite relationship. Among many types of disorderly motion, there will be a few where each point in the vibration system follows a definite pattern and

TABLE C-23 Machinery Lateral Vibrations, Less Than 0.5× rpm[20]

Symptom	Probable Cause	Examples
Appears suddenly at a frequency of 40–50% of rpm. Often disappears when speed is reduced. Lube oil temperature increase will often eliminate or reduce vibration severity.	Bearing oil whirl	■ Bearing clearance too large ■ Bearing lightly loaded ■ Oil viscosity too high ■ Improper bearing design
Same symptoms as bearing oil whirl except a decrease in lube oil temperature will often eliminate or reduce vibration severity.	Seal ring oil whirl	■ Oil ring seal acting as lightly loaded bearing
Same symptoms as bearing oil whirl; however, vibration frequency is constant even though speed is changed.	Resonant whirl	■ Same as oil whirl except triggered by resonant component such as rotor, casing, piping, etc.
Appears suddenly at or above rotor critical speed when critical is below one-half operating speed. Increasing speed increases vibration amplitude but whirl frequency remains constant. When speed is decreased, vibration disappears below where it first appeared.	Friction-induced rotor whirl	■ Encountered in builtup rotor or rotors with shrink fits or rotor disassembly to inspect fits, increase shrink fits ■ Coupling friction has been known to induce whirl
Vibration appears/disappears suddenly with increasing/decreasing speed, respectively.	Loose component	■ Rotor sleeves/impellers become loose as speed is increased ■ Bearing liners, housings, or self-aligning spherical casts have loose fits ■ Loose casing or supports
Vibration peaks at specific speeds and high axial vibration is often present.	Subharmonic resonance	■ Usually occurs as a result of loose components or as a result of aerodynamic or hydrodynamic excitations, areas to be investigated for correction are seals, thrust clearance, couplings, and rotor stator clearance

the frequency is common; these are called principal modes of vibration. Such a system might be a shaft between two supports, as drawn in Fig. C-335.

Most vibration occurs in periodic motion, which means it has cycles that repeat themselves. The simplest form of periodic motion is harmonic motion, which can be represented by a sine or cosine function. Periodic motion, however, is not always harmonic, although it may have harmonic components. See Fig. C-336.

The harmonic motion can be represented as:

$$\text{Displacement} = x = A \sin \omega t \tag{C-1}$$

Velocity and acceleration are obtained by differentiating displacement.

TABLE C-24 Machinery Lateral Vibrations, 1× Operating Frequency[20]

Symptom	Probable Cause	Examples
Increasing vibration amplitude with speed, behavior repeats for successive runs.	Unbalance	■ Loose rotor component ■ Foreign object lodged in rotor ■ Fouling ■ Poor balance [initial startup of rotor or components added to rotor (coupling, etc.)] ■ Off-center journal
Vibration peaks at specific speeds, peaks can usually be shifted with change in oil temperature	Rotor critical speeds	
Vibration peaks at specific speeds, oil temperature changes will generally not change the speed at which the peak occurs.	Structural resonance	■ Any machine part or supporting structure could have its natural frequency in the operating speed range
Increasing vibration with speed.	Casing distortion	■ Uneven casing warmup ■ External forces
Increasing vibration with speed, may or may not repeat for successive runs (prime 1×, also up to 5× present).	Bowed or bent rotor	■ Temporary heat bow ■ Permanent (no change with time)
Vibration amplitude varies with time in a definite rhythm (beat).	Beat frequency	■ Occurs when two or more machines mounted to common foundations operate at nearly the same speed ■ Occasionally a beat can develop in one machine if its operating speed is close to a structural component resonance
Increasing vibration with speed, other frequencies and axial vibration usually present.	Thrust-bearing damage	■ Usually result of off-design operation (surge, liquid slugging) or balance piston problem (design, plugged, or worn labyrinths)
Increasing vibration with speed. Shaft bearing housing amplitude about equal.	Sleeve bearing damage	■ Increased clearance ■ Wiped bearing
Increasing vibration with speed (prime frequency is 1× rpm plus many other frequencies).	Seal rub	■ Rub usually relieves itself and therefore appears as erratic vibration
High axial vibration, vibration erratic (prime frequency is 1×, also 2× present).	V or other drive belts, component eccentricity	■ Mismatched V belts ■ Drive and driven pulley not aligned ■ Gear pitchline has excessive runout ■ Thrust collar may be cocked
Electric motors. Vibration behaves like unbalance but disappears when power off.	Electric motor armature	■ Correct eccentricity
Appears on gears like rotor critical speed. Vibration peaks at specific speeds.	Torsional resonance	■ Usually occurs only during startup or drastic load-speed change

TABLE C-25 Machinery Lateral Vibrations, 2× Operating Frequency[20]

Symptom	Probable Cause	Examples
Increasing vibration with speed (prime 2×, also 1× and/or 5× present often accompanied by high axial vibration).	Misalignment of coupling or bearing	■ Thermal casing growth ■ Piping forces ■ Bearing housing alignment
Increasing vibration with speed.	Loose rotor components	■ Loose coupling hub ■ Loose impellers or sleeves
Increasing vibration with speed. Appears on adjacent rotors.	Coupling machining inaccuracies	■ Replace coupling
Increasing vibration with speed. Vibration appears/disappears suddenly. Appears on adjacent rotors.	Coupling damage	■ Pitting of coupling teeth ■ Loose coupling spacer ■ Dirt in coupling teeth
Increasing vibration with speed.	Unbalanced reciprocating part	■ Crankshaft- or piston-type machinery ■ Loose piston or rod
Vibration peaks at specific speed.	Harmonic resonance	■ Same as critical or resonance

TABLE C-26 Machinery Lateral Vibrations, Frequencies >2× rpm[20]

Symptom	Probable Cause	Examples
Erratic high-frequency vibration amplitude and possibly an audible sound.	Rotor rub	■ Labyrinth rubs generally self-correct. ■ Disc rubs due to thrust-bearing failure often self-correct temporarily through wear, steel-on-steel shrill noise during wear. ■ Rotor deflection is critical speed.
rpm × no. of vanes/blades (always present). rpm × no. of guide vanes/nozzles.	Vane/blade aerodynamic or hydraulic forces.	■ No concern for normal operations. Record signal for machine as new condition to permit identification of possible future problem. Also record harmonics.
rpm × no. of gear teeth (always present).	Gear mesh frequency	■ Record signal for reference. ■ Distress noisy and shows increase in GMF and harmonics. ■ $^1/_2$ GMF—even no. of teeth with machining error.
rpm × no. of lobes (always present).	Lobe pass frequency	■ Record for future reference.
rpm × no. of pads (always present).	Journal tilt pad bearing	■ Increased vibration with increasing clearance.
High-frequency, destructive vibrations. Unaffected by operating speed.	Steam turbine valve vibration	■ Rare occurrence; change valve plug, seat shape, or increase valve gear rigidity.
Multiples of running frequency.	Harmonic resonance	■ Multiples of component natural frequencies (rotor casing, foundation, bearing housing, diaphragms, etc.).

TABLE C-27 Vibration Diagnosis[20]

Usual Predominant Frequency*	Cause of Vibration
Running frequency at 0–40%	Loose assembly of bearing liner, bearing casing, or casing and support Loose rotor shrink fits Friction-induced whirl Thrust-bearing damage
Running frequency at 40–50%	Bearing-support excitation Loose assembly of bearing liner, bearing case, or casing and support Oil whirl Resonant whirl Clearance-induced vibration
Running frequency	Initial unbalance Rotor bow Lost rotor parts Casing distortion Foundation distortion Misalignment Piping forces Journal and bearing eccentricity Bearing damage Rotor-bearing system critical Coupling critical Structural resonances Thrust-bearing damage
Odd frequency	Loose casing and support Pressure pulsations Vibration transmission Gear inaccuracy Valve vibration
Very high frequency	Dry whirl Blade passage

* Occurs in most cases predominantly at this frequency; harmonics may or may not exist.

$$\text{Velocity} = \frac{dx}{dt} = A\omega\cos\omega t = A\omega\sin\left(\omega t + \frac{\pi}{2}\right) \tag{C-2}$$

$$\text{Acceleration} = \frac{d^2x}{dt^2} = -A\omega^2\sin\omega t = A\omega^2\sin\left(\omega t + \frac{\pi}{2}\right) \tag{C-3}$$

Note the phase angle difference between displacement, velocity, and acceleration. (See Fig. C-337.) Velocity leads displacement by 90° and acceleration leads displacement by 180°.

In an *undamped free system*, shown in Fig. C-338, if the mass is pulled downward and then released, the force of the spring, equal to its stiffness coefficient × distance displaced, tends to restore equilibrium.

This motion is described by

$$m\ddot{x} = -kx \tag{C-4}$$

where $-kx$ = restoring force or

$$\ddot{x} + \frac{k}{m}x = 0 \tag{C-5}$$

Maximum Allowable (But Possibly Conservative) Vibration Limits for Operation until Earliest Possible Corrective Shutdown (All Measurements in Inches per Second, Peak)

	Speed Harmonics								
Machine Type	1× rpm	2× rpm	3× rpm	4× rpm	1× Vane Pass	2× Vane Pass	1× Gear Mesh Freq.	2× Gear Mesh Freq.	1× Blade Pass
Blowers, up to 6000 rpm maximum	.50	.40	.25	.25	.10	.050			
Contrifugal compressors									
Horizontal	.25	.20	.15	.15	.10	.050			
Barrel	.15	.10	.10	.10	.05	.025			
Steam turbines									
Special purpose	.30	.25	.15	.15					.10
General purpose	.50	.40	.25	.25					.10
Gas turbines and axial compressors	.50	.40	.25	.25				.10	
Contrifugal pumps									
Between brgs.	.25	.20	.15	.15	.10	.05			
Overhung type	.50	.40	.25	.25	.10	.05			
Electric motors	.25	.20	.15	.15					
Gear units					Intermed. Freq.				
Parallel,					1000–3000 Hz				
spec. purp.	.25	.20	.15	.15	.15		.10	.05	
Parallel, gen. purp.	.50	.40	.25	.25	.15		.10	.05	
Epicyclic	.15	.10	.10	.10			.10	.05	

			1× Lobe Pass	2× Lobe Pass	3× Lobe Pass	4× Lobe Pass	5× Lobe Pass
Screw compressors	.25	.20	.20	.20	.20	.20	.20

Note that filtered components add up to unfiltered total amplitude of vibration.
(1) The significance of vane, blade, and lobe pass frequencies is not yet fully understood. More field data must be evaluated to arrive at universally meaningful maximum levels.
(2) Vane, blade, lobe pass, and gear mesh frequency amplitudes vary with load and/or speed change. The actual sensitivity should be part of the database.

FIG. C-330 Vibration measured on bearing housing.[15]

If we assume a harmonic solution to the equation, then we can have the following solution

$$x = c_1 \sin \omega t + c_2 \cos \omega t \tag{C-6}$$

If we use this in the previous equation, then

$$\left(-\omega^2 + \frac{k}{m}\right)x = 0$$

Frequency in Terms of Revolutions per Minute (rpm)	Most Likely Causes	Other Possible Causes and Remarks
1× rpm	Unbalance	■ Eccentric journals, gears, or pulleys ■ Misalignment or bent shaft—if high axial vibration ■ Bad belts if rpm of belt ■ Resonance ■ Reciprocating forces ■ Electrical problems
2× rpm	Mechanical looseness	■ Misalignment if high axial vibration ■ Reciprocating forces ■ Resonance ■ Bad belts if 2× rpm of belt
3× rpm	Misalignment	Usually a combination of misalignment and excessive axial clearances (looseness)
Less than 1× rpm	Oil whirl (less than $^1/_2$ rpm)	■ Bad drive belts ■ Background vibration ■ Subharmonic resonance ■ Beat vibration
Synchronous (ac line frequency)	Electrical problems	Common electrical problems include broken rotor bars, eccentric refer, unbalanced phases in polyphase systems, unequal air gap
2× synchronous frequency	Torque pulses	Rare as a problem unless resonance is excited
Many times rpm (harmonically related frequency)	Bad gears Aerodynamic forces Hydraulic forces Mechanical looseness Reciprocating forces	Gear teeth × rpm of bad gear Number of fan blades × rpm Number of impeller vanes × rpm May occur at 2, 3, 4, and sometimes higher harmonics if severe looseness
High frequency (not harmonically related)	Bad antifriction bearings	■ Bearing vibration may be unsteady—amplitude and frequency ■ Cavitation, recirculation, and flow turbulence cause random, high-frequency vibration ■ Improper lubrication of journal bearings (friction-excited vibration) ■ Rubbing

FIG. C-331 Vibration frequencies and the likely causes.[15]

which gives us the value of the system's single natural frequency for any x:

$$\omega = \sqrt{\frac{k}{m}} \tag{C-7}$$

Damped system. Types of damping include viscous damping, friction (coulomb) damping, and solid damping (or structural damping within the material itself). Figure C-339 depicts free vibration with viscous damping.

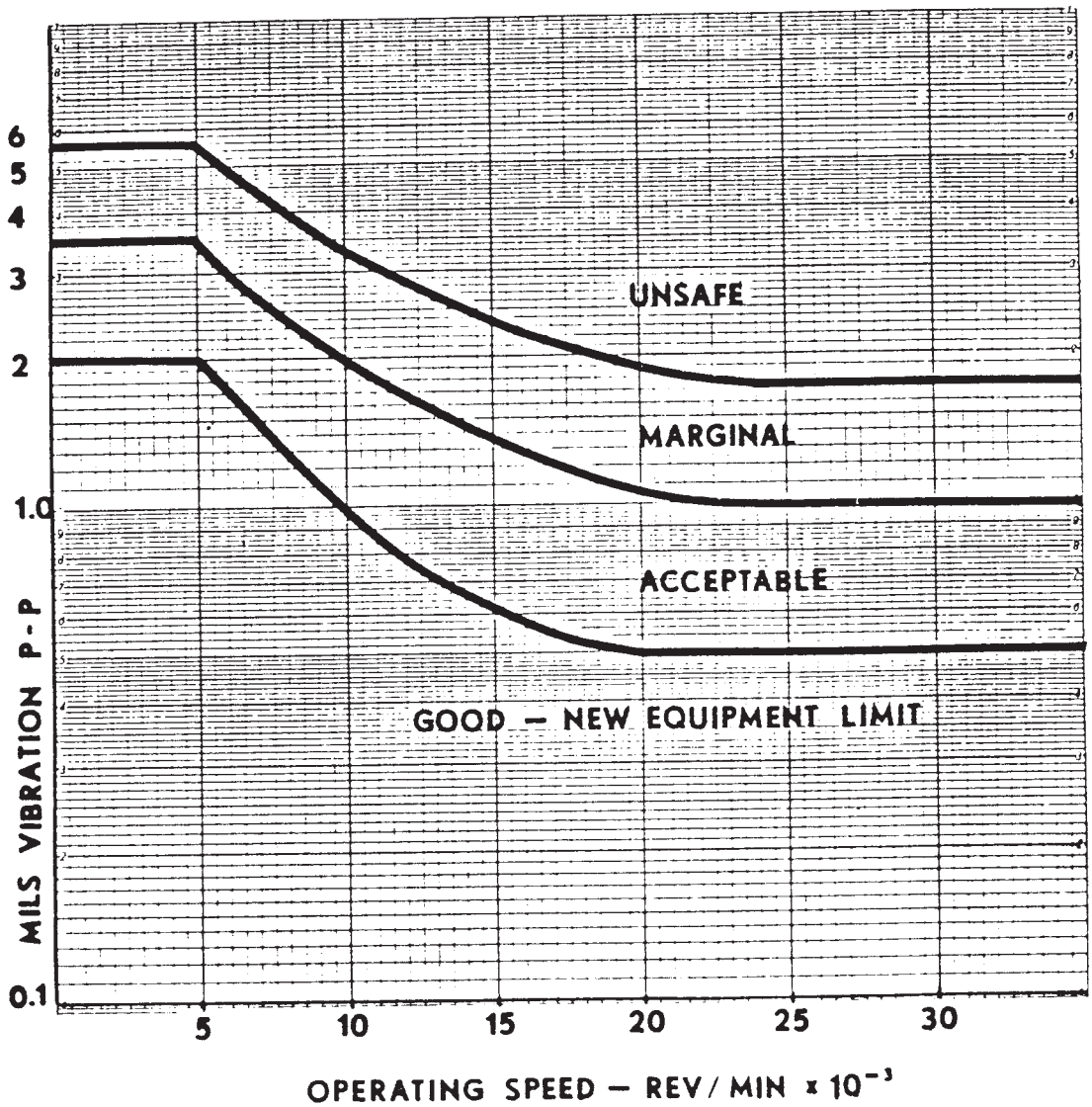


FIG. C-332 Turbomachinery shaft vibration guide.[15]

Viscous damping force is proportional to the velocity, so if c is the coefficient of viscous damping,

$$F_{\text{damp}} = -c\dot{x}$$

or

$$m\ddot{x} = -kx - c\dot{x} \tag{C-8}$$

or

$$m\ddot{x} + c\dot{x} + kx = 0$$

If we use the trial solution

$$x = c(e^{rt}) \tag{C-9}$$

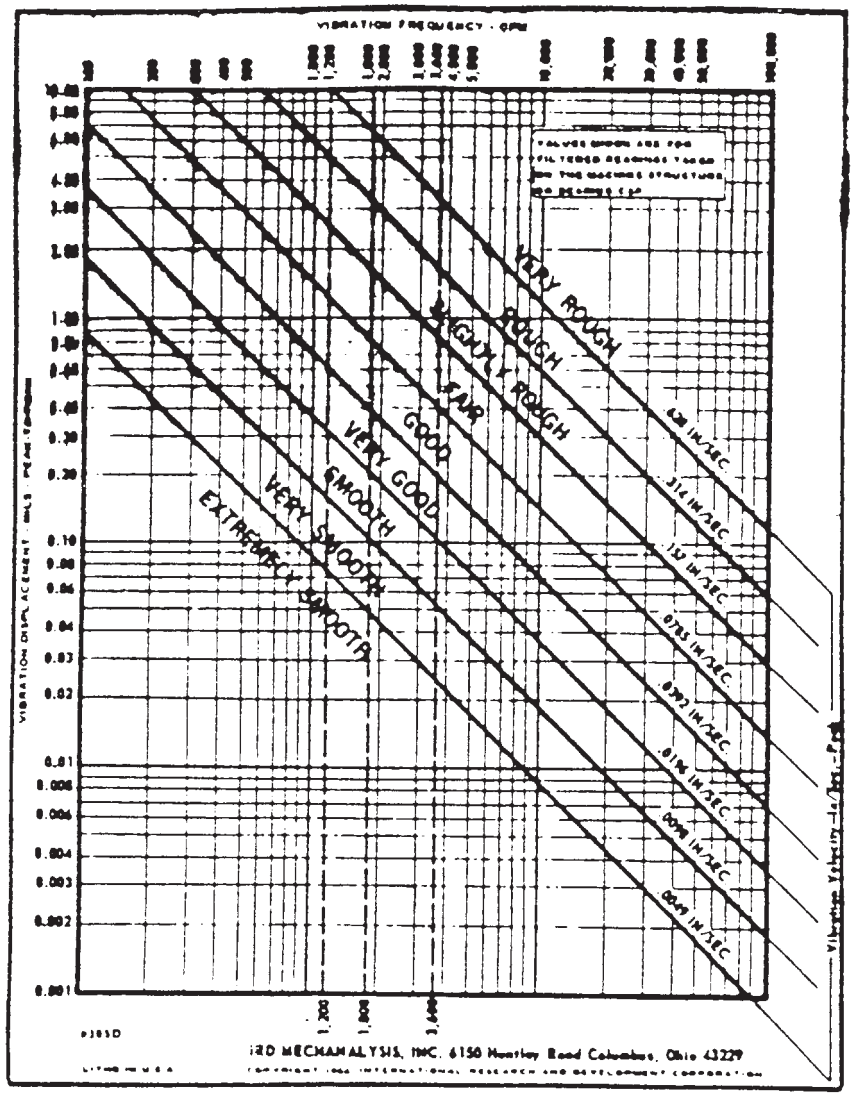


FIG. C-333 General machinery vibration severity chart.[15]

which, if we substituted into the previous equation, we get

$$\left(r^2 + \frac{c}{m}r + \frac{k}{m}\right)e^{rt} = 0 \tag{C-10}$$

which, if we solve for t

$$r_{1,2} = \frac{-c}{2m} \pm \sqrt{\frac{c^2}{4m^2} - \frac{k}{m}} \tag{C-11}$$

or in more general terms

$$x = e^{(-c/2m)t}\left[c_1 e^{\left[\sqrt{c^2/(4m^2)-(k/m)}\right]t} + c_2 e^{-\left[\sqrt{c^2/(4m^2)-(k/m)}\right]t}\right] \tag{C-12}$$

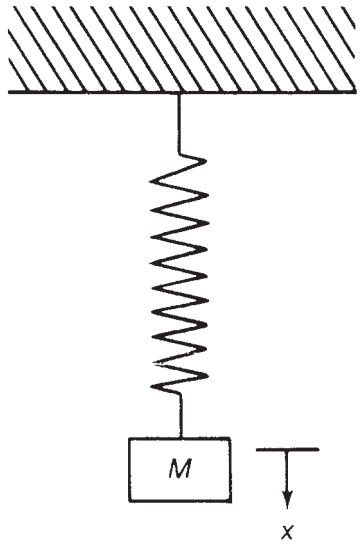


FIG. C-334 System with single degree of freedom.[13]

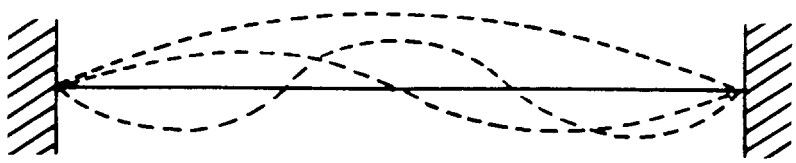


FIG. C-335 System with infinite number of degrees of freedom.[13]

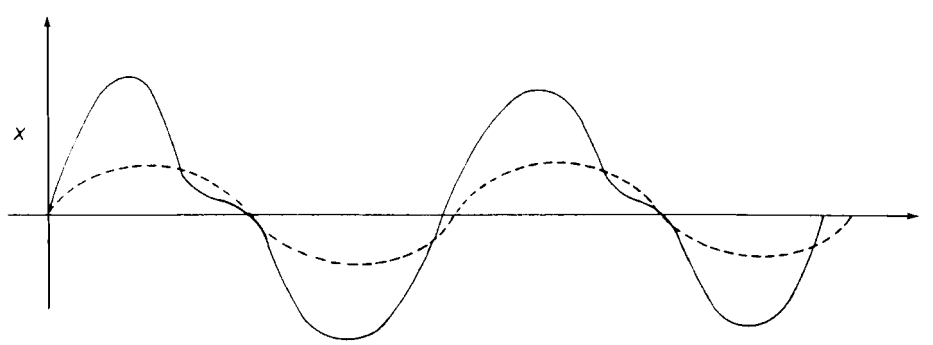


FIG. C-336 Periodic motion with harmonic components.[13]

The solution observed in the latter equation will depend on whether the roots are real, imaginary, or zero. If the critical damping coefficient required to make the root zero is c_c then

$$\frac{c^2}{4m^2} = \frac{k}{m}$$

or

$$\frac{c}{2m} = \sqrt{\frac{k}{m}} = \omega_n \tag{C-13}$$

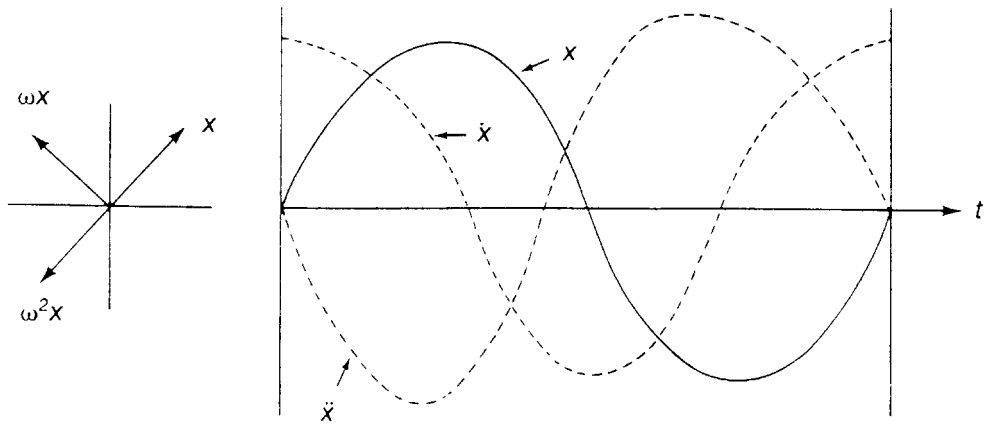


FIG. C-337 Harmonic motion of displacement, velocity, and acceleration.[13]

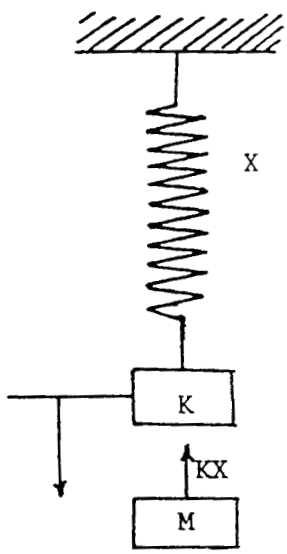


FIG. C-338 Single degree of freedom system (spring mass system).[13]

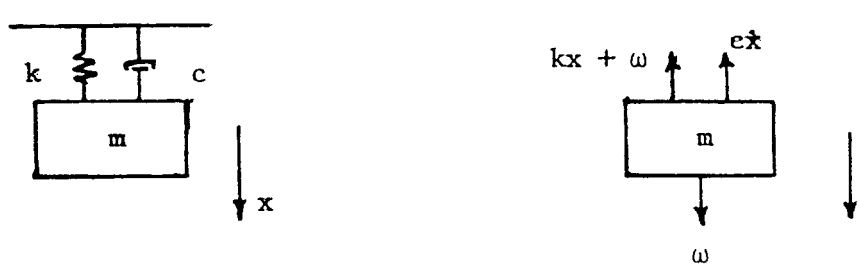


FIG. C-339 Free vibration with viscous damping.[13]

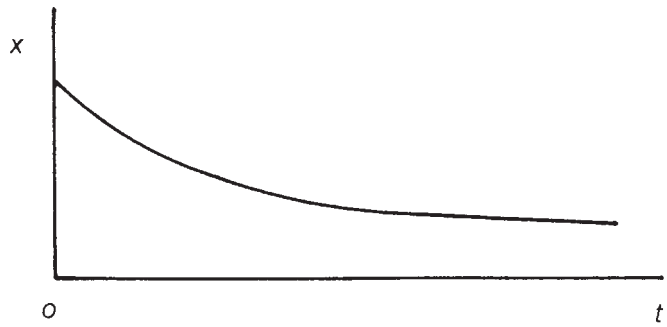


FIG. C-340 Overdamped decay.[13]

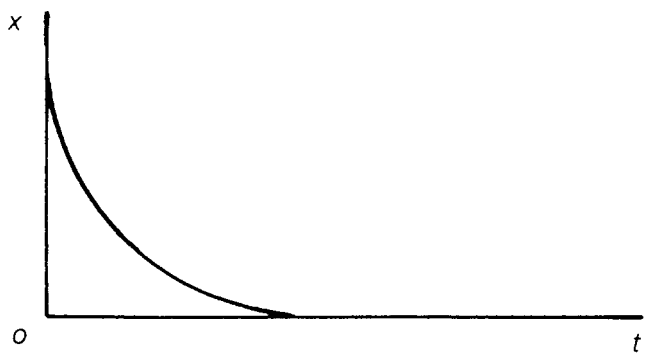


FIG. C-341 Critical damping decay.[13]

We now define the damping factor

$$\tau = \frac{c}{c_c} \tag{C-14}$$

to specify the amount of damping in any system.

In an overdamped system

$$\frac{c^2}{4m^2} > \frac{k}{m}$$

and the roots are real. Figure C-340 describes overdamped motion. Figures C-341 and C-342 show critical damping and underdamped decay.

With critical damping

$$\frac{c_{cr}^2}{4m^2} = \frac{k}{m}$$

or

$$c_{cr}^2 = 4m^2 \frac{k}{m} = 4mk$$

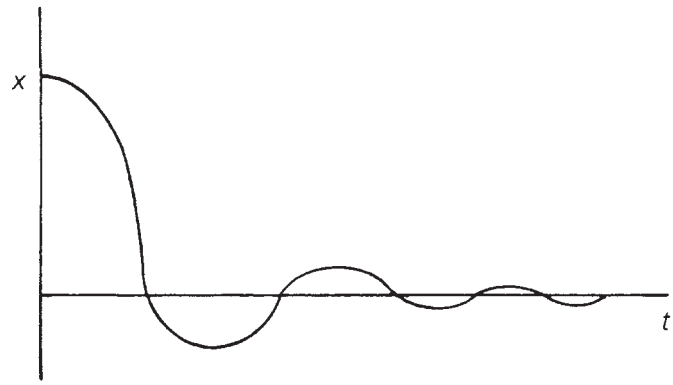


FIG. C-342 Underdamped decay.[13]

Thus

$$c_{\text{cr}} = \sqrt{4mk} = 2m\sqrt{\frac{k}{m}} = 2m\omega_n$$

i.e., the roots are equal.

Underdamped systems have imaginary root solutions. With an underdamped system,

$$\frac{c^2}{4m^2} < \frac{k}{m}$$

so the imaginary roots are given by

$$r_{1,2} = \pm i\sqrt{\frac{k}{m} - \frac{c^2}{4m^2}} \tag{C-15}$$

Then the response becomes

$$x = e^{(-c/2m)t}\left[c_1 e^{i\sqrt{(k/m)-(c^2/4m^2)}} + c_2 e^{-i\sqrt{(k/m)-(c^2/4m^2)}}\right]$$

which can be written

$$x = e^{-(c/2m)t}[A\cos\omega_d t + \beta\sin\omega_d t] \tag{C-16}$$

where x is the response amplitude.

Forced vibration. In forced vibration there is an external excitation force. See Figs. C-343 and C-344.

Now the equation of motion is

$$m\ddot{x} = F\sin\omega t - kx - c\dot{x} \tag{C-17}$$

or

$$m\ddot{x} + c\dot{x} + kx = F\sin\omega t$$

Let us say that the steady-state oscillation of this system is given by

$$x = D\sin(\omega t - \theta) \tag{C-18}$$

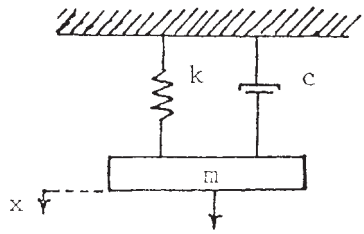


FIG. C-343 Forced vibration system.[13]

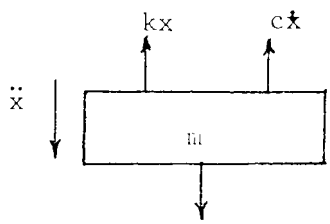


FIG. C-344 Free body diagram of mass (M).[13]

where D is displacement of the steady-state oscillation. Motion lags force by θ. So for velocity and acceleration, we have

$$\theta = \dot{x} = D\omega\cos(\omega t - \theta) = D\omega\sin\left(\omega t - \theta + \frac{\pi}{2}\right) \tag{C-19}$$

$$a = \ddot{x} = D\omega^2\sin(\omega t - \theta) = D\omega^2\sin\left(\omega t - \theta + \frac{\pi}{2}\right) \tag{C-20}$$

If we substitute the preceding two equations into the initial equation of motion (C-17) we get

$$mD\omega^2\sin(\omega t - \theta) - cD\omega\sin\left(\omega t - \theta + \frac{\pi}{2}\right) - D\sin(\omega t - \theta) + F\sin\omega t = 0 \tag{C-21}$$

In other words, the sum of all the system forces:

- Inertia
- Damping
- Spring
- Impressed force

is zero or,

Inertia force + damping force + spring force + impressed force = 0

From the previous equation, we see that displacement lags impressed force by the phase angle, θ, and the spring force acts opposite in direction to displacement. Damping force lags displacement by 90° and therefore acts in the opposite direction to velocity. Inertia force is in phase with displacement and acts in the opposite direction to acceleration. This agrees with the physical interpretation of harmonic motion. See Fig. C-345 for vectorial representation of the system.

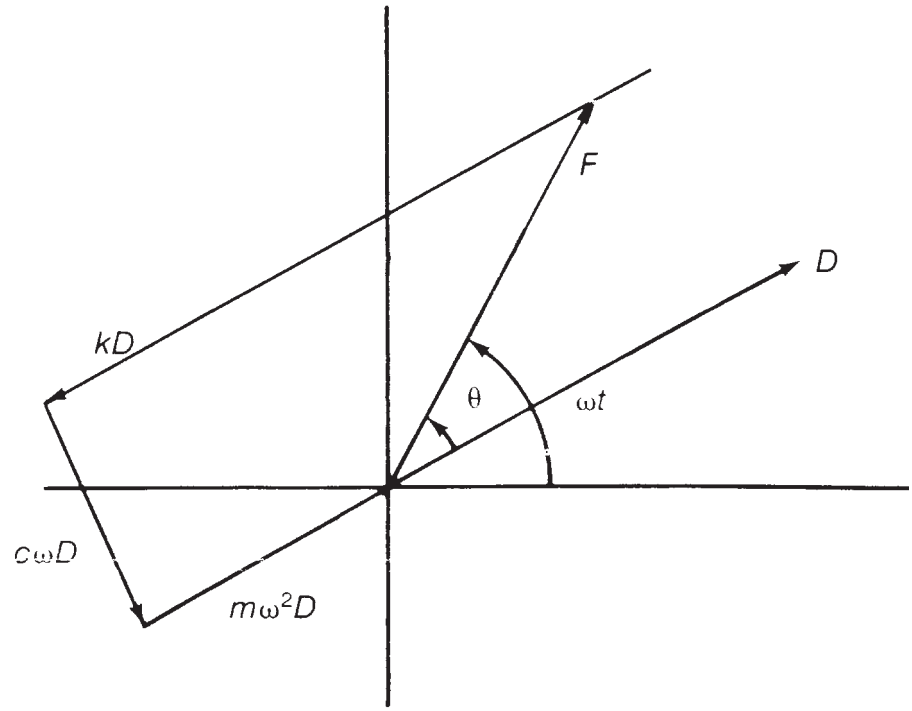


FIG. C-345 Vector diagram of forced vibration with viscous damping.[13]

- d lags F by θ
- kD acts opposite D
- Damping force lags D by 90°
- Damping force acts opposite velocity

From the vector diagram we get the phase angle and the amplitude

$$D = \frac{F}{\sqrt{(k - m\omega^2)^2 + c\omega^2}} \tag{C-22}$$

$$\tan\theta = \frac{c\omega}{k - m\omega^2} \tag{C-23}$$

Removing all units (nondimensional), we get (note $c_c = 2\,m\omega_n$ = critical damping coefficient)

$$D = \frac{F/k}{\left(1 - \frac{\omega^2}{\omega_n^2}\right) + \left(2\zeta\frac{\omega}{\omega_n}\right)^2} \tag{C-24}$$

$$\tan\theta = \frac{2\zeta\frac{\omega}{\omega_n}}{1 - \left(\frac{\omega}{\omega_n}\right)^2} \tag{C-25}$$

where $\omega_n = \sqrt{\frac{k}{m}}$ = natural frequency and $\zeta = \frac{c}{c_c}$ = damping factor.

We can see that damping has a large part to play in determining amplitude and phase angle in the zone around resonance. Amplitude at resonance tends to infinity. The effect of the magnification factor ($D/F/k$) and the phase angle θ is mainly a function of frequency ratio, the damping factor ζ. This is illustrated in Figs. C-346A and C-346B.

$$\text{For } \omega/\omega_n = 1.0, \text{ the phase angle is } 90^\circ$$

$$\text{For } \omega/\omega_n \ll 1.0, \text{ the phase angle is } s\text{mall, as are inertia and damping force}$$

Design considerations then include taking into account *natural frequency*. As we have seen, for a single degree of freedom, this parameter is given by

$$\omega_n = \sqrt{k/m}$$

Increasing the mass reduces ω_n, and increasing the spring constant k increases ω_n. From a study of damped system theory, the damped natural frequency

$$\omega_d = \omega_n \sqrt{1-\zeta^2}$$

is lower than ω_n.

Figures C-346A and C-346B also show that characteristics of forced vibration or resonance in rotating machinery reinforce the concept of avoiding extended running at or near critical speeds.

Simplification of above theory applied to actual machinery. We have seen that for a single degree of freedom system natural frequency is given by

$$\omega_n = \sqrt{k/m}$$

So if we increase mass, we decrease ω_n, and if we increase k, we increase ω_n.

We have also seen that

$$\omega_d = \omega_n \sqrt{1-\zeta^2} \text{ (lower than } \omega_n)$$

All rotating machinery has some unbalance and this produces excitation at the rotational speed. The system's natural frequency is also the critical speed of the shaft. The forced damped system study shows us that amplitude rate is maximized at

$$\omega_m = \omega_n \sqrt{1-2\zeta^2}$$

Note that the natural frequency is more important than the damped natural frequency.

If there is no damping, amplitude tends to infinity at the natural frequency. So critical speeds are avoided when operating machinery.

With small machinery, mass is small and large k is easy to arrange; so small, low-speed machines can operate safely in the range below critical speeds.

With large machinery, compressors, gas turbines, and team turbines included, the mass is large and speeds are high.

Therefore, a system with a very low critical speed is designed and the machine operated at above this critical speed. Due care must be observed moving through these speed range(s), and enough damping must be designed into bearings and foundations.

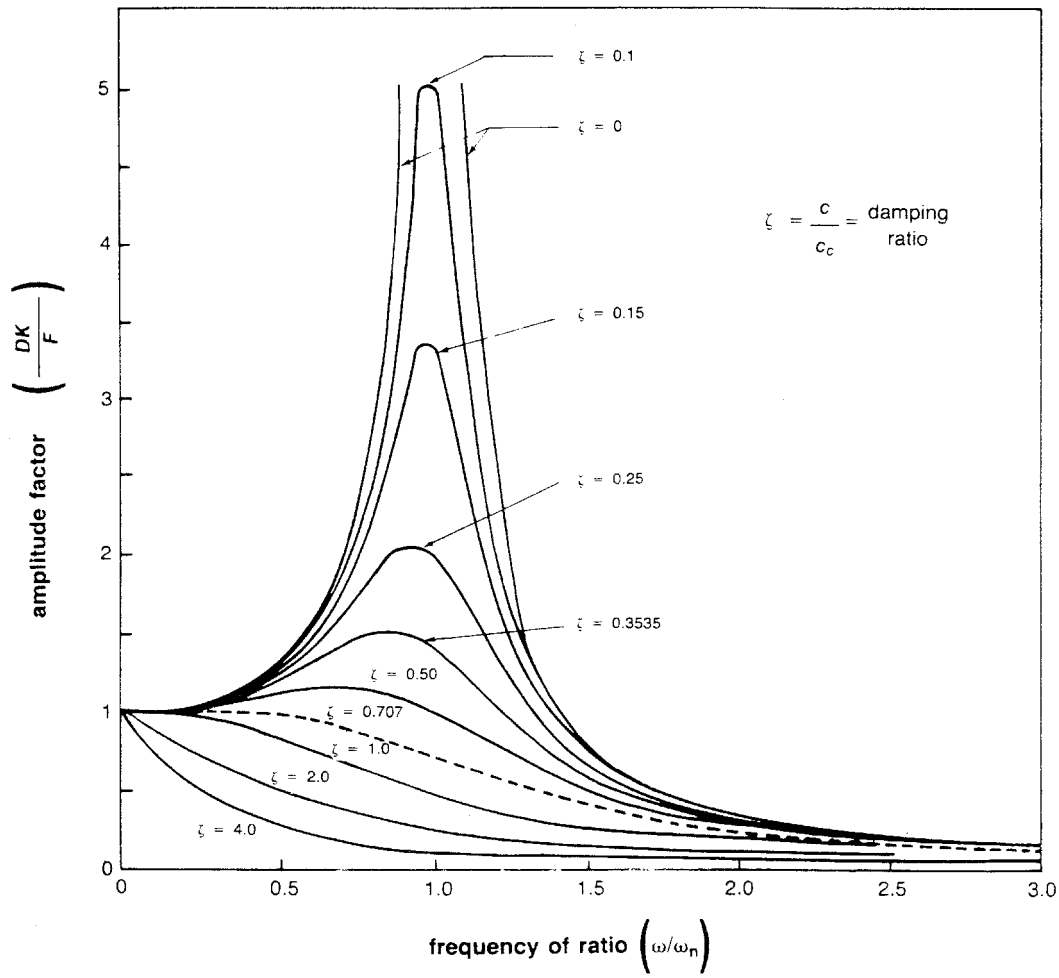


FIG. C-346A Amplitude factor as a function of the frequency ratio *r* for various amounts of viscous damping.[13]

The structural natural frequencies of most large systems are in the low-frequency range, so resonant couplings between structure and foundation must also be avoided.

Vibration-causing forces in rotating machinery comes from rotating unbalanced masses. These occur due to:

1. The center of mass and the center of rotation not being the same (uneven distribution of mass).
2. Deflection of a shaft due to its own weight aggravates the condition in 1.
3. 1 and 2 are static eccentricities. When rotation starts about a geometric center, they are amplified.
4. If the shaft is journal bearing supported, the shaft may move in an orbit, so the rotational is also rotated about geometric center.

Unbalanced forces increase as a function of the velocity squared. Balancing technology is the only method available to reduce these forces.

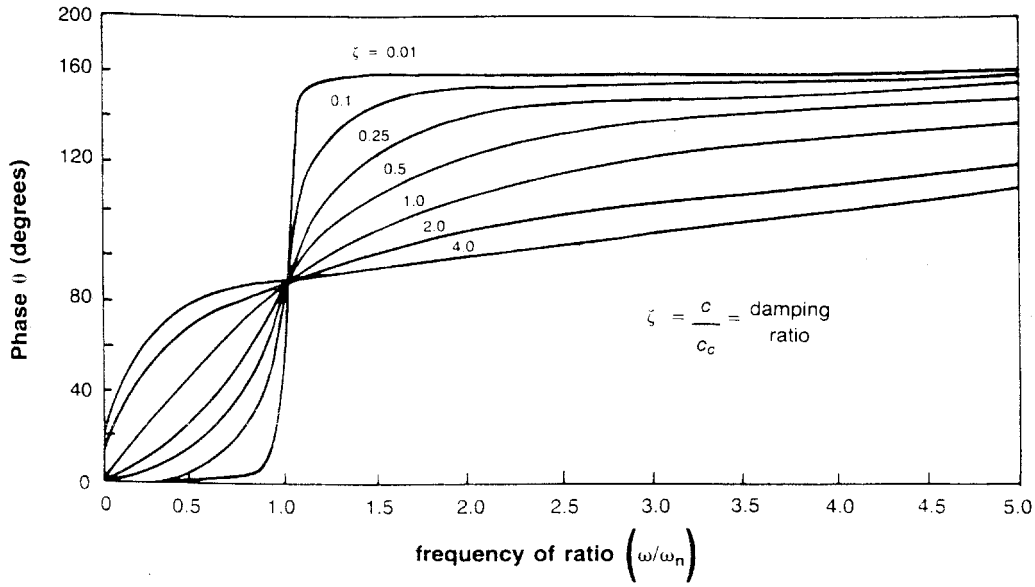


FIG. C-346B Phase angle as a function of the frequency ratio for various amounts of viscous damping.[13]

Further mathematical analysis is beyond the scope of this text. However, Table C-28 describes forces acting on rotor-bearing systems, an important basic aspect of rotor dynamics.

Thorough design analysis should also include consideration of the following factors:

1. Casing and foundation forces
2. Rotor-motion-prompted forces due to:
   - Mechanical and material properties
   - Various loadings of the system (viscous and hydrodynamic forces in the bearings and with blade loading)
3. Forces applied to the rotor (source: couplings, gears, misalignment, balance piston and axial thrust imbalance)
4. Rotor bearing instabilities caused by
   - External forces that cause resonance of a frequency dependent on the external mechanisms
   - Internal forces that cause instabilities
5. Forced vibration caused by
   - Unbalance
   - Bent shaft (2× rpm)
   - Misalignment

Rigid supports. The simplest model of a rotating machine consists of a large flat disc mounted on a flexible shaft with the ends mounted in rigid supports. These rigid supports constrain a rotating machine from any lateral movement but allow free angular movement. Figures C-347A and C-347B illustrate such a shaft that operates above its first critical. The mass center of the disc "e" is displaced from the shaft centerline or geometric center of the disc due to manufacturing and material

TABLE C-28 Forces Acting on Rotor-Bearing Systems[13]

Source of Force	Description	Application
1. Forces transmitted to foundations, casing, or bearing pedestals	Constant, unidirectional force	Constant linear acceleration.
	Constant force, rotational	Rotation in gravitational or magnetic field.
	Variable, unidirectional	Impressed cyclic ground or foundation-motion.
	Impulsive forces	Air blast, explosion, or earthquake. Nearby unbalanced machinery. Blows, impact.
	Random forces	
2. Forces generated by rotor motion	Rotating unbalance: residual, or bent shaft	Present in all rotating machinery.
	Coriolis forces	Motion around curve of varying radius. Space applications. Rotary-coordinated analyses.
	Elastic hysteresis of rotor	Property of rotor material that appears when rotor is cyclically deformed in bending, torsionally or axially.
	Coulomb friction	Construction damping arising from relative motion between shrunk-fitted assemblies. Dry-friction bearing whirl.
	Fluid friction	Viscous shear of bearings. Fluid entrainment in turbomachinery. Windage.
	Hydrodynamic forces, static	Bearing load capacity. Volute pressure forces.
	Hydrodynamic forces, dynamic	Bearing stiffness and damping properties.
	Dissimilar elastic beam	Rotors with differing rotor lateral stiffnesses. Slotted rotors, electrical machinery. Keyway. Abrupt speed change conditions.
	Stiffness reaction forces	
	Gyroscopic moments	Significant in high-speed flexible rotors with discs.
3. Applied to rotor	Drive torque	Accelerating or constant-speed operation.
	Cyclic forces	Internal combustion engine torque and force components.
	Oscillating torques	Misaligned couplings. Propellers. Fans. Internal combustion engine drive.
	Transient torques	Gears with indexing or positioning errors.
	Heavy applied rotor force	Drive gear forces. Misaligned 3-or-more rotor-bearing assembly.
	Gravity	Nonvertical machines. Nonspatial applications.
	Magnetic field: stationary or rotating	Rotating electrical machinery.
	Axial forces	Turbomachine balance piston. Cyclic forces from propeller, or fan. Self-excited bearing forces. Pneumatic hammer.

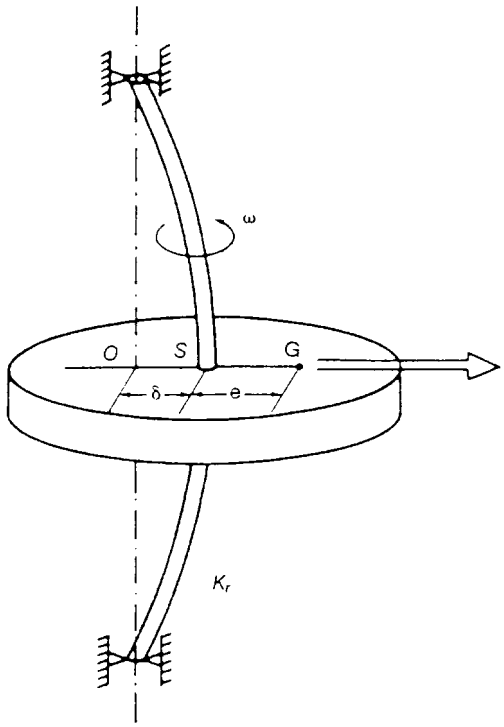


FIG. C-347A Rigid supports.[13]

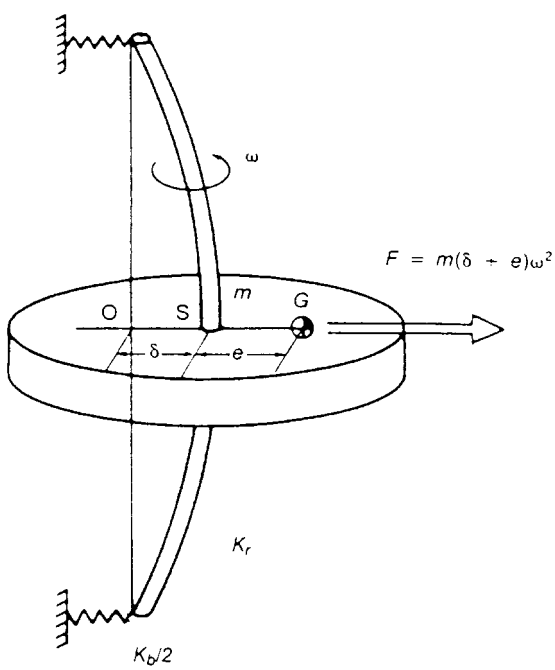


FIG. C-347B Flexible supports.[13]

imperfections. When the disc is rotated at rotational speed ω, the mass causes it to be displaced so that the center of the disc describes an orbit of radius δ_r from the center of the bearing centerline.

If the shaft flexibility is represented by radial stiffness K_r, that will balance the centrifugal force. Thus restoring force equals centrifugal force, or

$$K_r\delta = m\omega^2(\delta_r + e)$$

$$\delta_r = \frac{m\omega^2 e}{K_r - m\omega^2} = \frac{(\omega/\omega_n)^2 e}{1-(\omega/\omega_n)^2} \tag{C-26}$$

where

$$\omega_n = k_r/m$$

the natural frequency of the lateral vibration of the shaft and disc at zero speed.

When $\omega < \omega_n$, δ_r is positive.

When operating below the critical speed, the system operates with the center of mass on the outside of the geometric center.

When operating above the critical speed (i.e., $\omega > \omega_n$), shaft deflection δ_r tends to infinity. In real life, outside forces provide damping.

For very high speeds ($\omega \gg \omega_n$), amplitude δ_r equals $-e$, i.e., the disc rotates about its center of gravity.

Flexible supports. In actual life, all bearing systems have some flexibility. Let us assume each support has flexibility $K_{b/2}$ and the total system has flexibility K_b.

The flexibility of the entire lateral system is given by

$$\frac{1}{K_t} = \frac{1}{K_r} + \frac{1}{K_b} = \frac{K_b + K_r}{K_r K_b} \tag{C-27}$$

or

$$K_t = \frac{K_r K_b}{K_b + K_r}$$

Natural frequency is given by

$$\begin{aligned}\omega_{nt} &= \sqrt{\frac{K_t}{m}} = \sqrt{\left(\frac{K_r K_b}{K_b + K_r}\right)\Big/ m} \\ &= \sqrt{\frac{K_r}{m}}\sqrt{\left(\frac{K_b}{K_b + K_r}\right)} \\ &= \omega_n\sqrt{\frac{K_b}{K_b + K_r}}\end{aligned} \tag{C-28}$$

In a very rigid support

$$K_b << K_r$$

then

$$\omega_{nt} = \omega_n$$

With finite stiffness, or

$$K_b \gtrless K_r$$

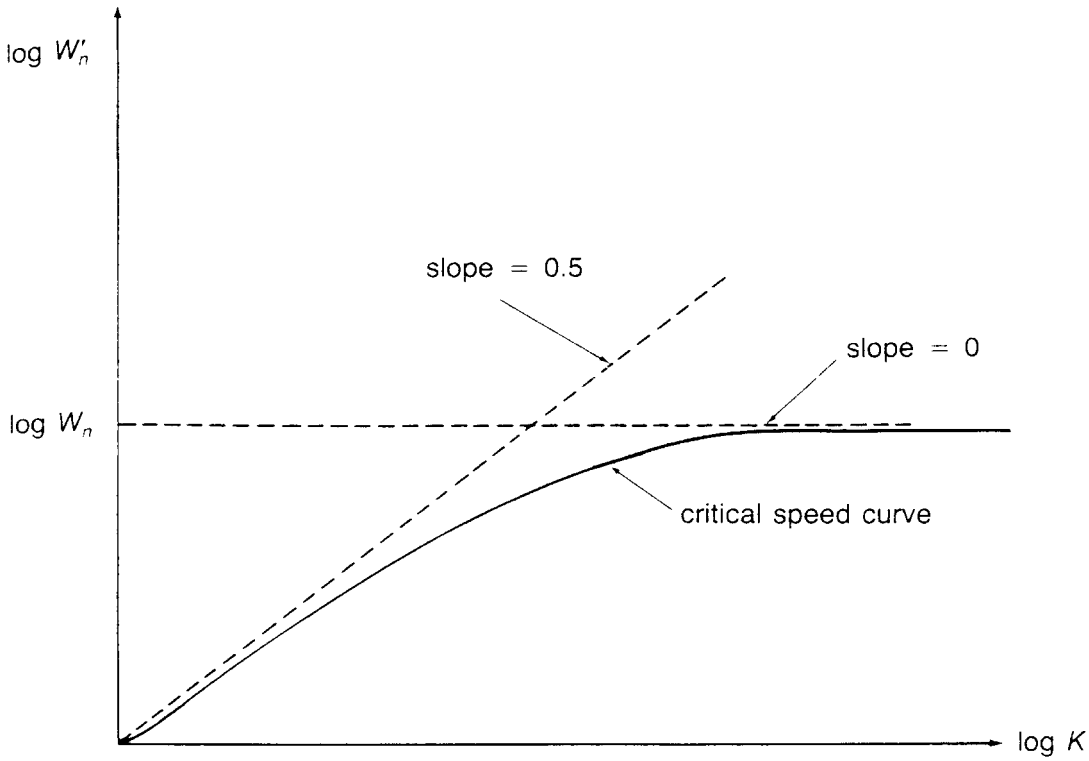


FIG. C-348 Critical speed map.[13]

then

$$\omega_n < \omega_{nt}$$

Therefore, we see that flexibility lowers the natural frequency of a system. This is diagrammatically represented in Fig. C-348, which can be extended to include 2nd, 3rd, and higher critical speeds.

Figure C-349A shows vibration modes of a uniform flexible shaft supported at both ends by rigid supports.

Figure C-349B shows a rigid rotor supported by flexible supports.

The critical speed concept theory outlines the identification of the operating range of a rotor-bearing system, probable mode shapes, and approximate locations of peak amplitudes.

Forced vibration. In forced vibration the usual driving force in rotating machinery is the shaft speed or multiples of this speed. Note the characteristics of forced and self-excited vibration in Table C-29.

The speed becomes critical when excitation frequency is equal to one of the natural frequencies of the vibration.

In forced vibration the system is a function of frequencies that can be multiples of rotor speed excited by frequencies other than 1× rpm. They might be blade-passing frequency, gear-teeth frequency, and so forth.

For forced vibration, critical frequency remains the same for any shaft speed.

Damping reduces the amplitude of forced vibration but does not change critical speed values.

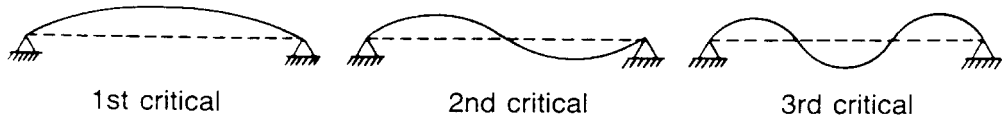


FIG. C-349A Rigid supports and a flexible rotor.[13]

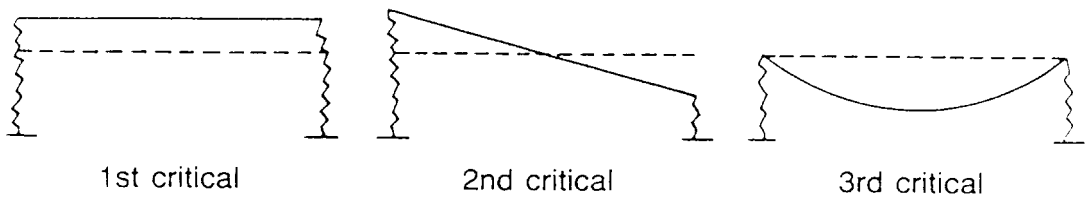


FIG. C-349B Flexible supports and rigid rotors.[13]

TABLE C-29 Characteristics of Forced and Self-Excited Vibration[13]

	Forced or Resonant Vibration	Self-excited or Instability Vibration
Frequency/rpm relationship	$N_F = N_{rpm}$ or N or rational fraction	Constant and relatively independent of rotating speed
Amplitude/rpm relationship	Peak in narrow bands of rpm	Blossoming at onset and continues to increase with increasing rpm
Influence of damping	Additional damping Reduce amplitude No change in rpm at which it occurs	Additional damping may defer to a higher rpm Will not materially affect amplitude
System geometry	Lack of axial symmetry External forces	Independent of symmetry Small deflection to an axisymmetric system Amplitude will self-propogate
Vibration frequency	At or near shaft critical or natural frequency	Same
Avoidance	Critical frequency above running speed Axisymmetric Damping	Operating rpm below onset Eliminates instability Introduce damping

Campbell diagram. Figure C-350 is a representation of a Campbell diagram. The Campbell diagram is an overall view of all the vibration excitations that can occur on an operating system. The numbered lines (numbers with a circle around them) are engine order lines. For illustration purposes, the second stage compressor blade has been theoretically represented. We see that operating the compressor at a specific speed range—12,000 rpm—will excite the 200-Hz first harmonic (200 Hz = 200 × 60 = 1200 rpm). This range should be avoided.

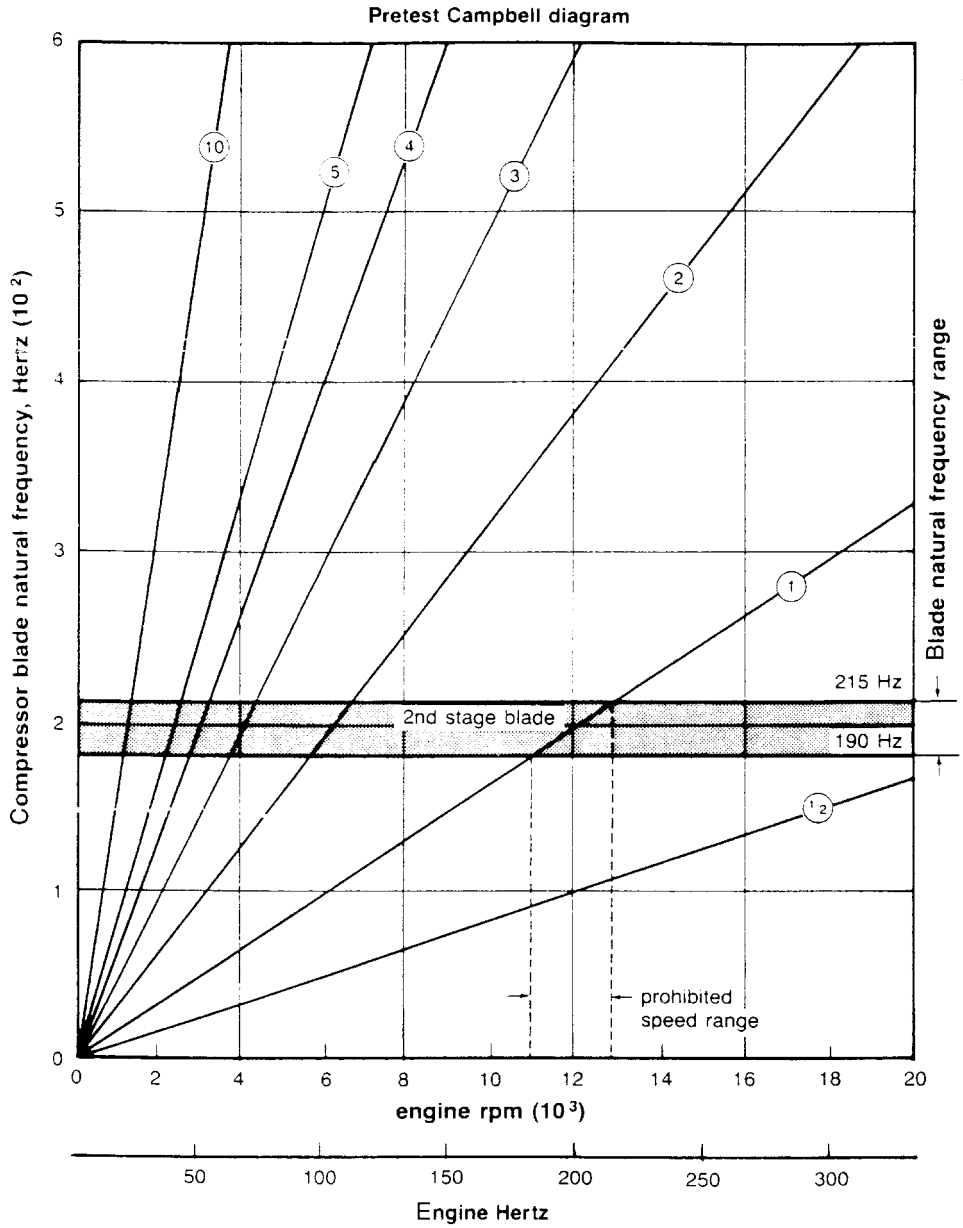


FIG. C-350 Campbell diagram.[13]

To ensure that blade stress levels are within fatigue life requirements, one might strain gauge the blading on prototype test models to measure stress levels. An impeller might also be mounted on a shaker table with variable frequency output (0–10,000 Hz). Accelerometers can be mounted at various positions on the impeller and used with a spectrum analyzer to record frequency response. See Figs. C-351 and C-352.

The results of all these tests may be plotted on a Campbell diagram, such as Fig. C-353. Lines of excitation frequency are vertical lines on the diagram. The design speed is represented by a horizontal line. When lines of excitation frequencies and multiples of running speed intersect near the line of design running speed, there may be a problem.

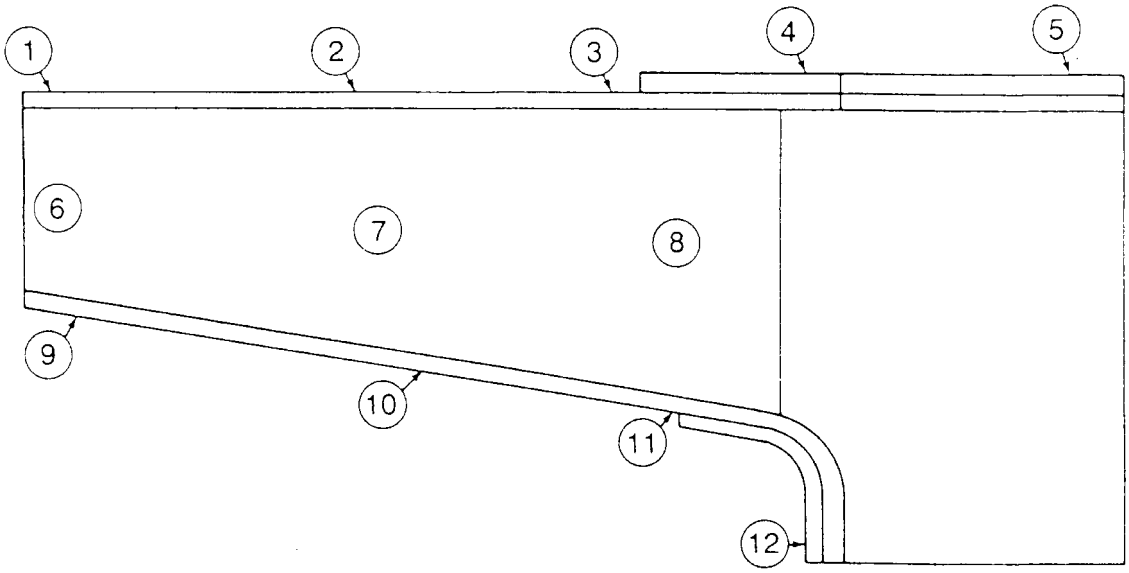


FIG. C-351 Accelerometer locations on impeller tested.[13]



FIG. C-352 Impeller showing nodal points.[13]

In the case, for instance, of an impeller with 20 blades, a design speed of 3000 rpm (or 50 Hz), and a critical frequency of 1000 Hz, the impeller may have a serious problem at these conditions (the intersection of the running speed line, 1000-Hz frequency line, and line of slope 20 N) as its critical is 20 N.

Introduction to balancing using influence coefficients

The theory of balancing using influence coefficients will not be dealt with fully here as it involves some cumbersome mathematics. However, this technique

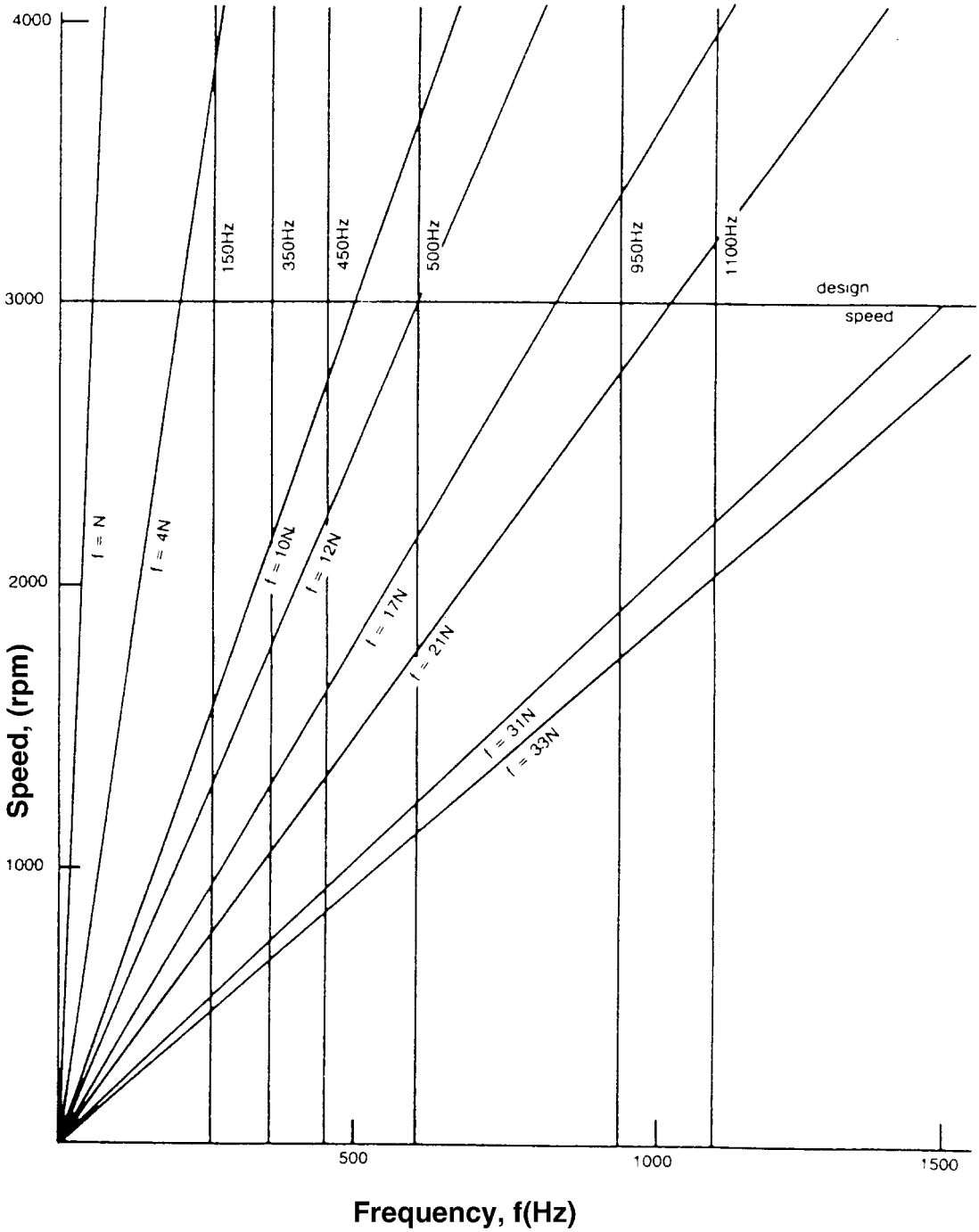


FIG. C-353 Campbell diagram of tested impeller.[13]

is incorporated into modern multiplane balancing machines today. The accuracy of these machines is possible because of advances in sensor technology and minicomputers.

Consider a rotor with n discs. Let $P, P, \ldots, P_n$ be the forces acting on the shaft. Then the deflection Z_i in the ith plane is given by:

$$Z_i = \sum_{j=1}^{n} e_{ij} P_j, i = 1, \ldots, n$$

This equation defines the compliance matrix $[e_{ij}]$ and the elements of the matrix are called *influence coefficients*. The compliance matrix is obtained by making

$$P_j = \delta_y$$

where δ_y is the Kronecker delta, and measuring deflections and Z_j. As j is varied from 1 to n, each column of the compliance matrix is developed.

Knowing the initial vibration in each plane q_i, the system of equations

$$\sum_{j=1}^{n} e_{ij} F_j = q_1, i = 1, \ldots, n$$

is solved for the correction forces. The correction weights can be computed from the correction forces.

In balancing with influence coefficients:

- Initial unbalance amplitudes and phases are recorded
- Trial weights are inserted sequentially at selected locations along the rotor
- Resultant amplitudes and phases are measured at convenient locations
- Required corrective weights are computed and added to the system

This method requires no foreknowledge of the system dynamic response characteristics. If it were available it would help in selecting more effective readout locations and trial weights.

Influence coefficients examine relative displacement rather than absolute displacements. Damping or initially bent rotors do not affect the process.

Introduction to modal balancing

The modal balance technique will not be discussed in detail in this course. It does, however, involve equating the deflection of a rotor at any speed to the sum of various modal deflections multiplied by constants that depend on speed. So a rotor that has been balanced at all critical speeds will then be balanced at all other speeds. The basic procedure for a rotor with end bearings is to first balance the shaft as a rigid body, then balance for all critical speeds in the operating range, then balance out any noncritical modes at the running speed.

Typical principal modes (1st, 2nd, and 3rd) for a symmetric and uniform shaft are illustrated in Fig. C-354.

Modal balancing, mathematically speaking, is based on the fact that a flexible rotor may be balanced by eliminating the effect of the unbalance distribution in a mode-by-mode sequence. The deflections of a rotor at any speed may be represented by the sum of various modal deflections multiplied by constants dependent on speed.

$$\overline{Y}(x, \omega) = \sum_{r=1}^{\infty} \overline{B}_r(\omega) \times \eta_r(x)$$

where $\overline{Y}(x, \omega)$ represents the amplitude of transverse vibrations, as a function of the distance along the shaft at a rotational speed ω. $\overline{B}_r(\omega)$ and $\eta_r(x)$ express the complex coefficient at rotating speed ω and the rth principal mode, respectively.

In other words *a rotor that is balanced at all critical speeds is also balanced at any other speed*. For end bearing rotors, the usual procedure is:

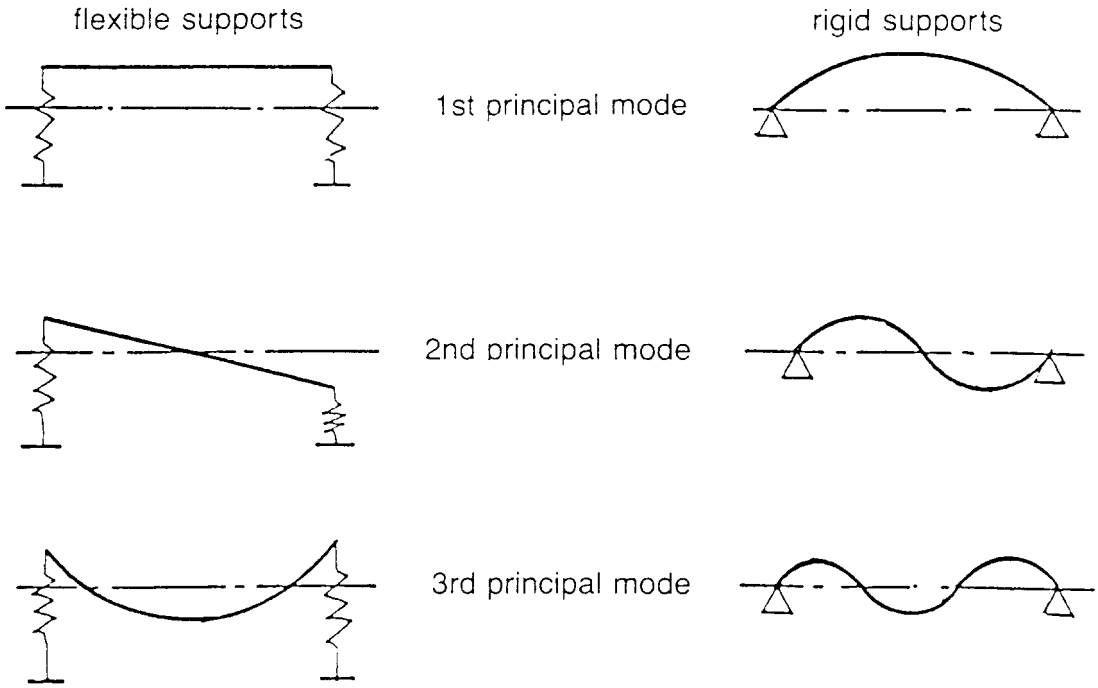


FIG. C-354 Typical principal modes for a symmetric and uniform shaft.[13]

1. Balance the shaft as a rigid body.
2. Balance for each critical speed in the operating range.
3. Balance out the remaining noncritical modes as far as possible at the running speed. Balance planes picked are the ones where maximum amplitudes of vibration occur.

Modal balancing has severe limitations. Calculated values for mode shapes and natural frequencies must be computed. Input data may limit the accuracy of these figures. Then, when damping is significant, such as in the case of fluid film bearings, problems arise. Damped modes look very different from undamped ones, so predicting modes and frequencies can be a severe problem. No one has overcome this to the extent that this method can be used as influence coefficients can in modern balancing machines.

Surge Control; Pressure and Flow Transducers

Optimum performance of driven equipment (compressors and pumps) enhances delivery of revenue product. A great deal of effort is, therefore, spent determining the parameters that affect turbomachinery performance. These include capacity, pressure ratio, power (and therefore fuel) consumed, and the surge characteristics of the machine.

Surge describes the condition when pressure at the suction drops to the extent that the flow reverses on itself. In a compressor, it first occurs on individual blades. This is called *stall* or *rotating stall*. Fully developed rotating stall is surge. In a pump, the surge phenomenon is called *cavitation*. Both surge and cavitation destroy equipment in what may be disastrous failures.

Performance analysis of driven equipment, whether done for test or for normal operation, generally requires that the following readings be taken:

1. Inlet temperature
2. Inlet pressure
3. Discharge temperature
4. Discharge pressure
5. Compressor speed
6. Atmospheric pressure
7. Differential pressure across the flowmeter
8. Temperature at the flowmeter
9. Pressure at the flowmeter
10. Fluid (gas, liquid) properties

The preceding properties determine the mass flow through the machine, based on volume flow, pressure and temperature readings, and fluid properties. The gas properties help work out the revenue base for the delivered fluid. For instance, gas may be sold to customers based on a certain heat value per unit time.

$$\text{Mass flow} \times \text{calorific value of fuel per unit mass} = \text{heat delivered per unit time}$$

As field composition changes, molecular weight frequently changes so gas composition analysis should be done regularly. If done frequently, changes in gas composition may provide early warning of a declining field.

All flow and pressure measurement devices require certain minimum lengths of straight pipe prior to measurement stations. A length of 10 pipe diameters is considered good practice. If piping is protracted and tortuous, accurate measurement may be difficult. Valves in the system must have the right closure speed characteristics and leak-tight properties.

Pressure measurement

Pressure measuring devices include the following:

1. Pitot static tubes
2. Pressure transmitters and transducers
3. Barometers
4. Liquid manometers

During commissioning, startup screens are used to protect the machine against welding slag and other inclusions left in the piping during construction. They cause a sizable differential drop, however, and should be removed.

Manometers are frequently used to measure differential pressures and atmospheric pressure. They should be calibrated regularly. It is important to watch for trapped bubbles within the liquid because they can make a reading inaccurate.

Inlet and discharge pressures are the sum of the static pressure and the velocity pressure at the respective points.

Flow transducers

Fluid (gas and liquid) flow through compressors and pumps (driven equipment) is measured by appropriate measuring transducers. These transducers include:

1. *Venturi tubes.* These consist of a convergent section at the entrance, a constant diameter throat, and a divergent section. They are accurate but hard to retrofit. They should be planned for in advance.
2. *American Society of Mechanical Engineers (ASME) flow nozzle.* This is essentially little used in the field as it is hard to fit in a process plant. It works well in test rigs.

Items 1 and 2 handle about 60 percent more flow than orifice plates.

3. *Orifice plates.* Orifice plates are mostly the concentric orifice type, although an eccentric orifice might be used, depending on the quantity of fluid to be measured.
4. *Elbow flowmeters.* Centrifugal force at the elbow is used to find the difference between the inside and the outside of the elbow. This is then related to discharge pressure.

There are many other techniques that are more theoretical, better suited for lab work, or require complex data to be supplied by the OEM.

Nonintrusive Wear Monitoring

Nonintrusive wear monitoring is a technique where neutron bombardment is used to determine the state of wear of a bearing in a nonintrusive wear. The technique was pioneered on lower-temperature machinery such as diesels. Although it is dropping in price, it is still expensive. The wear monitors are relatively bulky, which has not helped their popularity.

Since its introduction, vibration techniques and performance analysis techniques have developed well and dropped in price. It is unlikely that nonintrusive wear monitoring will ever rival contemporary vibration and temperature monitoring for detecting bearing failures. It may make a comeback when development budgets are not tight.

References and Additional Reading

1. Soares, C. M., "Aspects of Aircraft Engine Monitoring Systems Experience as Applicable to Ground Based Gas Turbine Engines," TMC, 1988.
2. Soares, C. M., "Design, Installation and Operation of Turbomachinery in Western Canadian Gas and Oil Production," ASME IGTI, 1981.
3. Lifshits, Simon, and Smalley, "More Comprehensive Limits for Rotating Machinery," ASME *Journal for Gas Turbines*, vol. 108, October 1986.
4. Lifson et al., "Assessment of Gas Turbine Vibration Monitoring," ASME-GT-204.
5. Floyd, "Key Issues and Technology for Future Programs," *Proceedings Fifteenth Annual SFTE Conference*.
6. "An Overview of Airborne Vibration Monitoring (AVM) Systems," Society of Automotive Engineers, SAE 871731.
7. Vibration seminar notes, Mechanical Technology Incorporated.
8. *Aspects of Flexible Rotor Balancing*, 2d ed., Schenk Treble, January 1976.
9. Simmons and Smalley, "Effective Tools for Diagnosing Elusive Turbomachinery Problems in the Field," ASME 89-GT-71.
10. *Balancing of Jet Engines*, Schenk Treble, A1511.
11. Soares, C. M., "Vibration Analysis: Separating the Elements of Machinery, Process and Personnel," TMC, 1992.
12. Soares, C. M., "Latest Techniques in Repair Technology," TMC, 1990.
13. Boyce, turbomachinery course notes, 1979.
14. Soares, C. M., "Condition Monitoring," *Asian Electricity*, 1997.
15. Bloch, H., machinery failure analysis and troubleshooting course notes.
16. Gunter, "Rotor Bearing Stability," *Proceedings First Turbomachinery Symposium*, 1972.
17. Thomson, *Mechanical Vibrations*, 2d ed., Prentice Hall, Englewood Cliffs, NJ, 1961.
18. Buscarello, R., vibration course notes, 1980.
19. Handbook on vibration, McGraw-Hill, New York.
20. Bloch, H., and Soares, C. M., *Turboexpanders and Process Application*, Gulf Professional Pub., 2001.

Control Regulators (*see* Actuators)

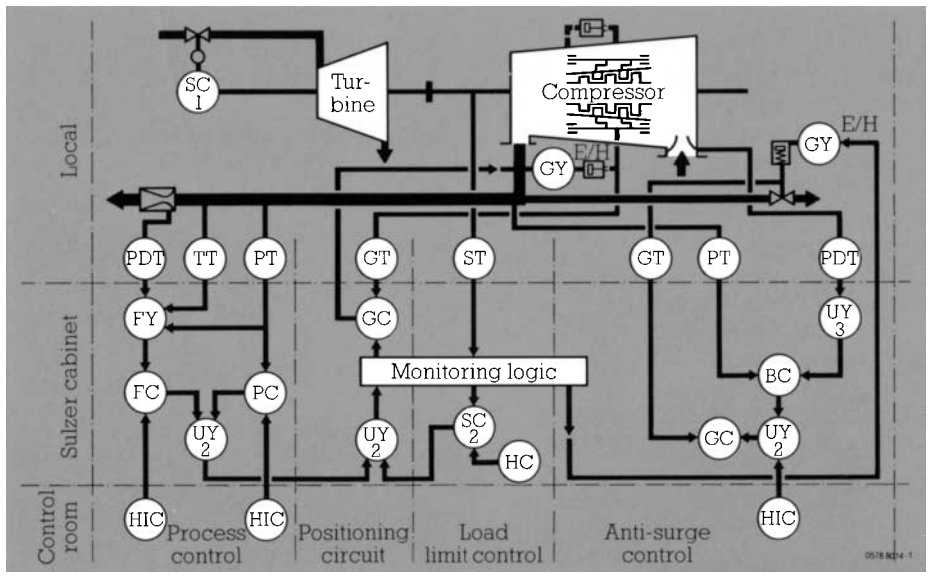


FIG. C-355 Antisurge control system and combined discharge pressure/power limit control system. SC 1, speed controller; E/HGY, electrohydraulic converter; PDT, differential pressure transmitter; TT, temperature transmitter; PT, pressure transmitter; GT, position feedback transmitter; ST, speed transmitter; FY, computer; GC, positioning controller; SC 2, load limit controller; UY 3, function generator; FC, volume controller; PC, pressure controller; BC, antisurge controller; UY 2, minimum-selection relay; HC, reference value station; HIC, manual station. (Source: Sulzer-Burckhardt.)

Control Systems; Controls

A *control system* is what controls and governs an operating system or part of that system. The field of controls and control systems is wide enough to fill several handbooks on its own. However, in this more general book, we shall attempt to highlight certain common aspects and systems of process controls with examples. We have selected systems that have, when malfunctioning, missing, or wrongly applied, caused the bulk of control-related problems in operating plants in the major process industries. This selection of features and systems will also illustrate where other problems with control system elements might be found.

A control system can be represented by a schematic pressure and instrument (P&I) line diagram. Its scope may vary. For instance we have a control system that runs an overall plant, a control system that runs a gas turbine package (driver, driven compressor, and all auxiliaries), and a control system that governs part of that overall package (say compressor) or one (say stator blade movement) or more of its functions. Several control systems for compressors are looked at in some detail.

Control Systems of Turbocompressors*

Two main requirements are to be met by the control system of turbocompressors. See Figs. C-355 through C-357.

- Safety: to prevent the compressor from operating in an unstable range or at other hazardous conditions
- Process: to adapt the compressor performance to the demands of the process

* Source: Sulzer-Burckhardt, Switzerland. Adapted with permission.

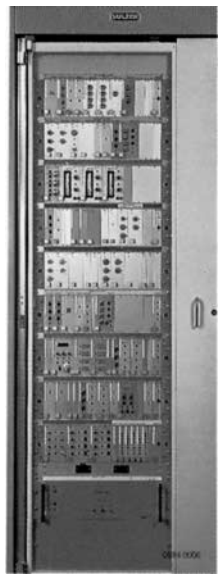


FIG. C-356 Electronic control cabinet. (Source: Sulzer-Burckhardt.)

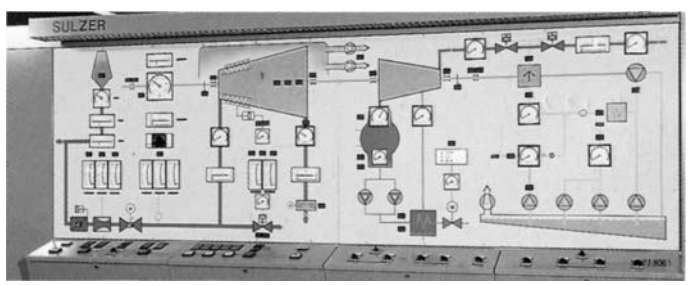


FIG. C-357 Example of a monitoring panel. (Source: Sulzer-Burckhardt.)

The control systems may be pneumatic or electronic with hydraulic or electromechanical servomotor. In cases where the compressor control system is engineered or furnished by others, it is customary to review and approve the system in order to ensure the compatibility of all equipment and functions. With employment of standard signals, the control system can be integrated into other systems, including safety systems, without difficulty: it allows remote control, automation of starting and stopping, and integration with process computers.

Safety systems

Antisurge control. Turbocompressors have a limited stable operating range, regardless of the type of blading and other influencing factors. This range is given by the characteristic curves limited by the surge lines. Surge conditions, occurring on the left side of this surge line, are avoided by an antisurge control system. It measures flow and pressure and can be designed to closely follow the actually measured surge line at a given safety margin. See Fig. C-358.

As soon as the operating point approaches the surge line, the controller starts opening the antisurge valve according to the preset values (L).

For air compressors, the excess capacity is blown off to the atmosphere.

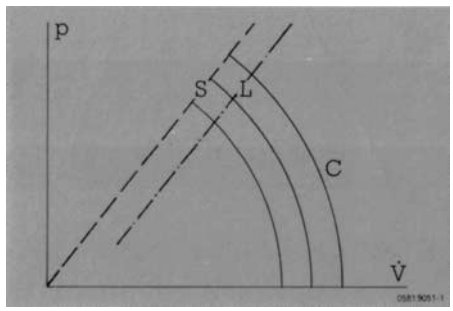


FIG. C-358 Characteristics of a turbocompressor. *p* = discharge pressure, *V* = flow rate, *C* = compressor characteristic curves, *S* = surge line, *L* = limit flow. (Source: Sulzer-Burckhardt.)

For a gas that cannot be wasted to the atmosphere, antisurge control is a bypass control, the unwanted flow being returned to the suction side. A bypass cooler may then be required.

The antisurge control system is not a flow or pressure control device, but a safety device that has to act independently of any other control.

Vibration, temperature, pressure, power limitation. Under certain circumstances external influences may lead to undesired changes of the normal level of vibration, gas, and bearing temperatures, pressure, and power. A reliable interlock, alarm, and shutdown system must protect the compressor and driver from possible damage under such conditions.

Auxiliary component control. Auxiliary component control ensures a safe supply of lube, control, and seal oil.

Process parameters

Suction pressure. Constant suction pressure to adapt the compressor flow to an upstream production unit or to maintain a constant evaporation temperature for refrigerating units.

Discharge pressure. Constant discharge pressure in cases where chemical reactions have to take place at a clearly defined pressure, or where the compressor flow has to be adapted to a fluctuating downstream demand.

Flow. Constant mass flow control corresponding to a constant plant output.

Typical process control system A (see Fig. C-355)

System for either capacity control and pressure limitation or pressure control with capacity limitation.

The capacity is measured by means of a Venturi tube. The computer FY receives input signals from the differential pressure transmitter (PDT), pressure transmitter (PT), and temperature transmitter (TT) and calculates the actual capacity value for the flow controller (FC).

The FC and the pressure controller (PC) are each equipped with one manual station HIC for the reference value. By way of the minimal-selection relay UY2, the volume or pressure controller acts upon the positioning controller GC of the stator blade adjusting mechanism. The positioning circuit for the stator blade adjusting mechanism comprises the positioning controller GC, the electrohydraulic converter GY, and the position feedback transmitter GT.

Load limit control

Malfunctioning in the steam or condensate system of the driving turbine may lead in certain circumstances to an undesired drop in speed. In such cases, the stator blades will be closed to such an extent by the load limit controller SC 2 that the speed will remain nearly constant. The speed controller SC 2, which is equipped with the reference value station HC, receives its input signal from the speed transmitter ST and has priority over the process controller with minimal selection relay UY 2, when necessary.

Antisurge control system

The system will maintain stable operation of the compressor, even when the process operating point moves into the unstable range of the compressor performance characteristic.

The PT is used to determine the actual value. A PDT and a function generator UY 3 are employed for setting the reference value. This computer enables the response line (blow-off line) to be well adapted to the surge limit. The output signal of the antisurge controller BC acts by way of the minimum selection relay UY 2 on the positioning controller GC of the blow-off valve. The valve can also be opened by the manual station HIC and override the antisurge controller. The positioning circuit for the blow-off valve comprises the positioning controller GC, the electrohydraulic converter GY, and the position feedback transmitter.

Typical process control system B

- Suction pressure: constant suction pressure to adapt the compressor flow to an upstream production unit or to maintain a constant evaporation temperature for refrigerating units
- Discharge pressure: constant discharge pressure in cases where chemical reactions have to take place at a clearly defined pressure, or where the compressor flow has to be adapted to a fluctuating downstream demand
- Flow: constant mass flow control corresponding to a constant plant output

Figure C-359 shows an antisurge control system for a centrifugal compressor having practically constant suction conditions. The pressure differential transmitter (PDT) is used to determine the actual volume. A pressure transmitter (PT) and a function generator UY1 are used to set the reference value for the antisurge controller. This computer enables the response line (blow-off line) to be well adapted to the actual surge limit. The output signal of the antisurge controller BC acts upon the actuator of the blow-off valve by way of the minimum selection relay UY2 via positioner GY. The valve can also be opened by the manual station HIC independently of the antisurge controller.

The discharge pressure control system with power limit control regulates the adjustable inlet guide vanes. The pressure controller PC receives its actual value from pressure transmitter PT and the set value from the manual station HIC. The power limit controller EC receives the corresponding values from the power converter ET and manual station HC. By way of the minimal selection relay UY2, the pressure or power controller acts upon the actuator of the inlet guide vanes through the electric/pneumatic converter GY.

Features of the system

Frequently, the OEM supplies the entire control, safety, and instrument system. Their experience comprises various suppliers of equipment for all types of drivers and most diverse process applications. OEMs commonly offer:

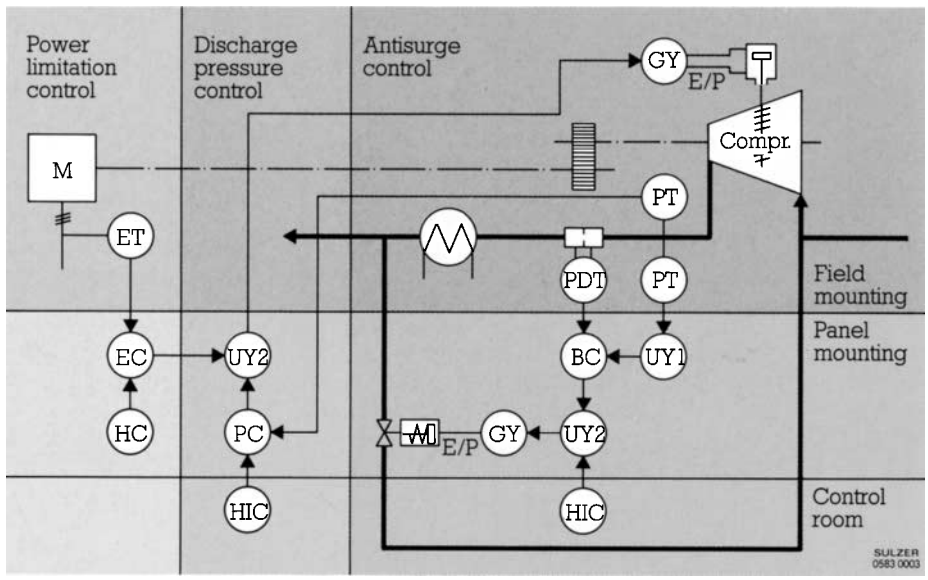


FIG. C-359 Antisurge control system and combined discharge pressure/power limit control system. (Source: Sulzer-Burckhardt.)

- Responsibility of compressor and driver control/safety system
- Design and construction of instrumentation and panels in accordance with all kind of standards
- Engineering and supply of instrumentation and panels for the entire compressor train, including driver, in line with clients specifications and incorporating all considerations resulting from actual operational experience with compressor trains of similar duty
- Experience with various suppliers of instruments and control/safety equipment
- Full functional test of instrument panel in the works
- Simple interface in case of use of a process computer

Monitoring logic

The monitoring logic supervises various controller reference values, the speed of the set, time-dependent operations such as startup and the positioning of the final control elements. The system facilitates the determination and processing of any disturbances in the final positioning elements and transmitter signals.

General system layout

The control system of these axial compressors is designed for all types of drive and diverse process applications.

- Transmitter according to the 2-wire system; standard signals 4–20 mA
- Isolating amplifier in the case of external measurement data
- Plausibility supervision for the determination of measurement data faults
- Output tracking for automatic changeover to manual control made in case of measurement failure
- Impulse technique of the control units

- Positioning elements driven by hydraulic servomotors and controlled by electrohydraulic converters
- Two changeover contacts for each output for alarm and control purposes

Controls, of Airfoil Movement

For example, Figs. C-360 and C-361 show an axial compressor where the intake flow may be governed by automatic control of variable stator vanes. In the event of a control oil pump failure, the compressor stator blades fail closed. See Fig. C-362.

Controls, of Lube and Seal Oil Systems*

Any item of turbomachinery or a turbomachinery train must have a lube and seal oil system or an oil system where the lube and seal oil have separate circuits of their own (see Fig. C-357). Figure C-363 is a P&I diagram of an intercooled centrifugal compressor. An example of a standard lube oil supply system is shown in Fig. C-364. A seal oil system for floating ring seals (standard for API packages) is depicted in Fig. C-365.

Lube oil unit

The standardized lube oil units are generally available in two versions: according to the OEM's standard and the API standard 614. See Figs. C-366 and C-367. The major components of a unit are the oil tank, auxiliary oil pump, double filter, and, selectively, one or two oil coolers. All components of the smaller units are mounted on a common bedplate, while the oil tank of a big unit is mounted on its own bedplate, separated from the other components. The oil can be heated up by an electrical or steam-heating unit. The necessary instrumentation is a standard supply item and, if requested, the switches and motors can be wired up. The main oil pump is driven preferably by the gear unit of the driving machine. A separate drive arrangement is also possible.

Seal oil unit

When floating ring seals are used, seal oil (or seal water) is required. A seal oil unit according to API standards, either separate or combined with the lube oil unit, is available. Besides the oil tank, two oil pumps, double filter, and cooler, which are the basic components of any oil supply system, the seal oil unit comprises also an overhead tank that creates the static seal oil overpressure and a degasifier drum with nitrogen injection.

Control of Overload and Overspeed (of Gas Turbines)

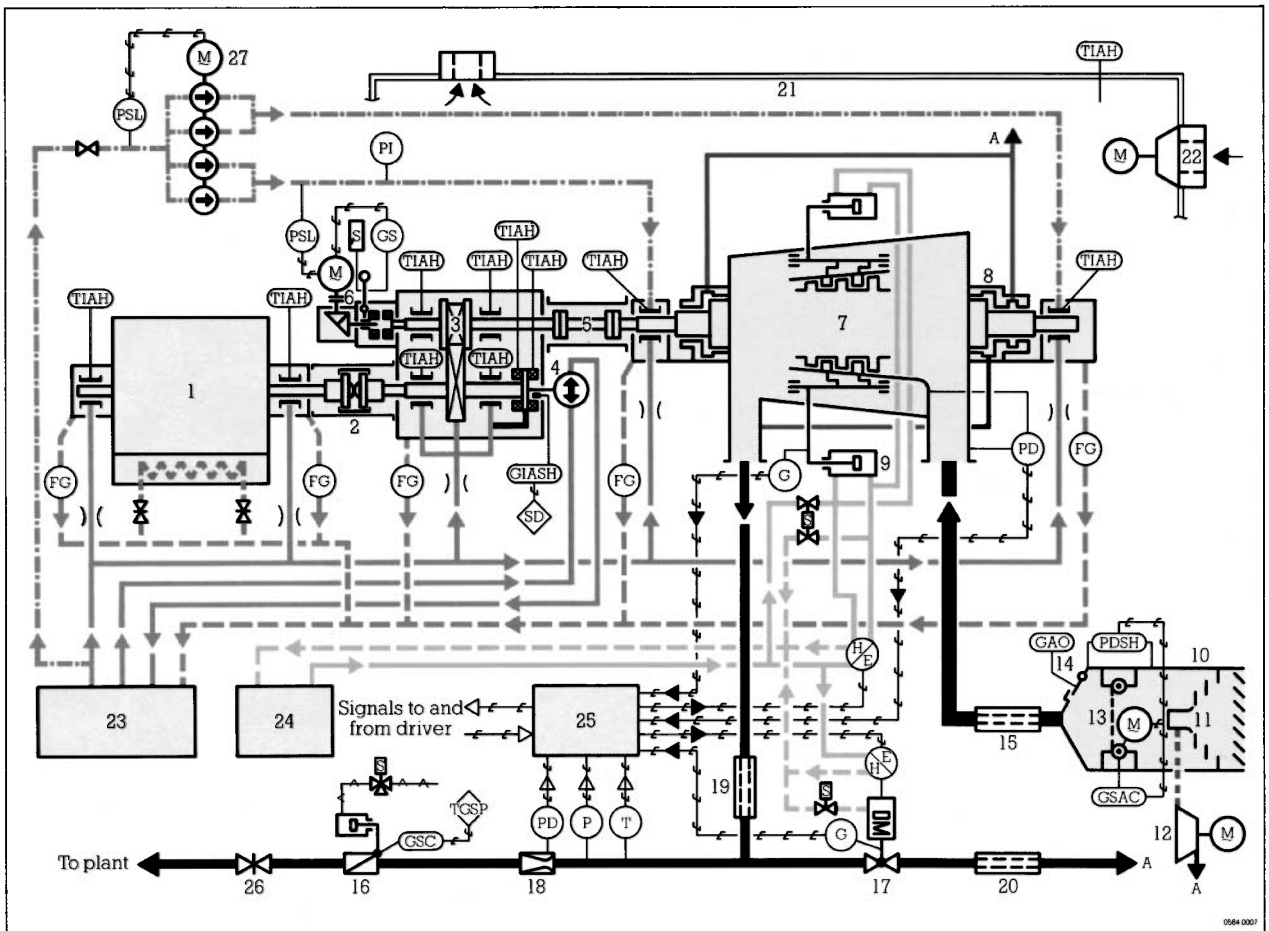
Protection of gas turbines against overload and overspeed†

Gas turbine generator sets (see Fig. C-368), particularly aeroderivative gas turbines, are sensitive to electrical fault events that can damage the gas turbine. Mechanical protection—i.e., with a torque-limiting coupling—is in many cases not

---

*Source: Sulzer-Burckhardt, Switzerland. Adapted with permission.

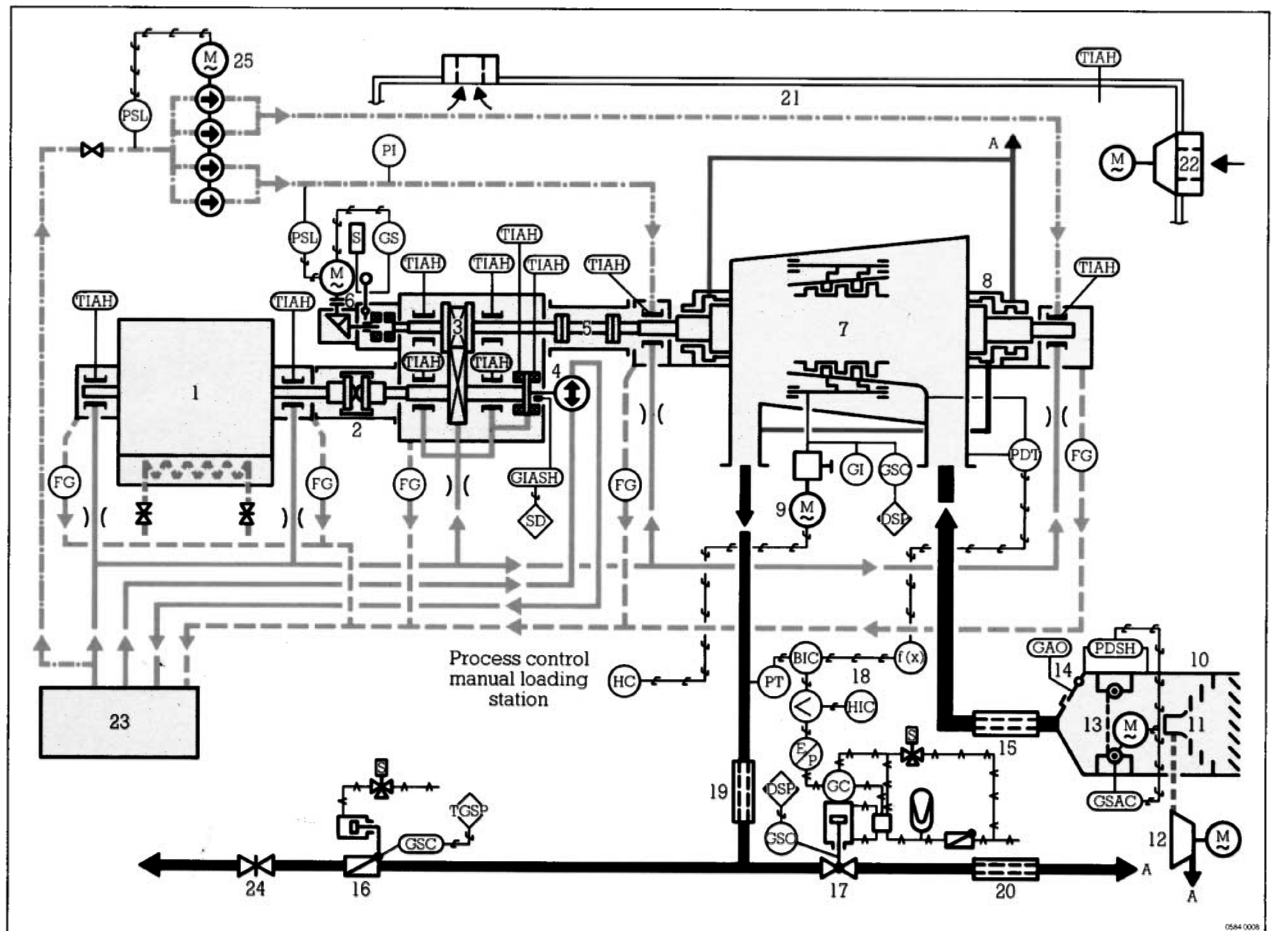
†Source: J.M. Voith GmbH, Germany.



List of major components

1 Driving motor
2 Gear coupling, oil-filled
3 Speed-increasing gear with thrust collars, thrust bearing on the slow-running shaft
4 Main oil pump, laterally flanged to the gear-box, driven by the slow-running gear shaft
5 Solid coupling
6 Turning gear
7 Axial compressor with adjustable stator blades
8 Labyrinth glands
9 Servomotor of the stator blade adjusting mechanism
10 Two-stage air intake filter consisting of:
  11 Inertia dust separator (1st stage)
  12 Dust evacuation fan
  13 Roll-band filter (2nd stage) with driving motor and automatic forward feed control
  14 Bypass flap with open alarm switch GAO
15 Suction silencer
16 Non-return valve with air-operated closing cylinder and solenoid valve
17 Antisurge valve, hydraulically operated
18 Venturi tube for measuring the air flow
19 Discharge silencer
20 Blow-off silencer
21 Noise-attenuating hood
22 Ventilation fan with silencer
23 Lube oil supply unit (see Fig. C-362)
24 Control oil supply unit (see Fig. C-362)
25 Electronic control system and monitoring logic
26 Discharge isolating valve
27 Jacking oil pump

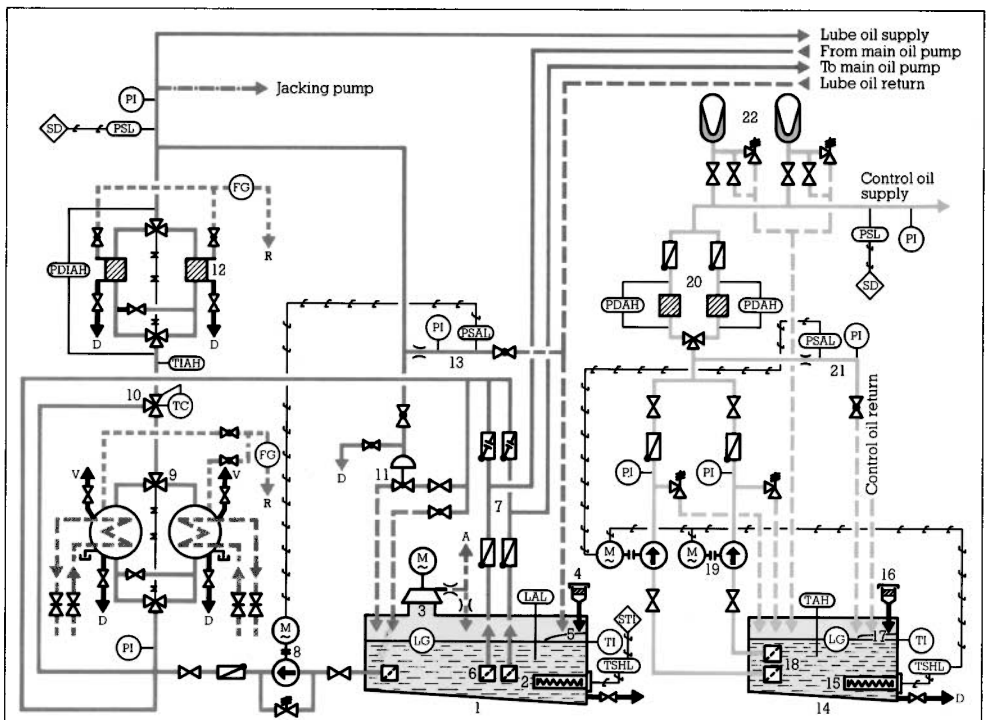
FIG. C-360 Automatic process flow control by means of adjustable stator blades. (Source: Sulzer-Burckhardt.)



List of major components

1 Driving motor
2 Gear coupling, oil-filled
3 Speed-increasing gear with thrust collars, thrust bearing on slow-running shaft
4 Main oil pump, laterally flanged to the gear-box, driven by the slow-running gear shaft
5 Solid coupling
6 Turning gear
7 Axial compressor with adjustable stator blades
8 Labyrinth glands
9 Electric servomotor and gear for remote setting of stator blade adjusting mechanism by means of controller HIC; gear with local manual intervention facility
10 Two-stage air intake filter consisting of:
11 Inertia dust separator (1st stage)
12 Dust evacuation fan
13 Roll-band filter (2nd stage) with driving motor and automatic forward feed control
14 Bypass flap with open alarm switch GAO
15 Suction silencer
16 Non-return valve with air-operated closing cylinder and solenoid valve
17 Antisurge valve, pneumatically operated, with air failure to open safety device (pressure accumulator) and solenoid valve. The antisurge valve is controlled via an electric/pneumatic converter E/P by the antisurge controller BIC. Manual control of the valve is possible in the sense "Valve to open" (the antisurge controller thereby being overridden) by operating the manual loader HIC
18 Function generator, computing the set points of the antisurge controller BIC in function of the aspired air flow rate
19 Discharge silencer
20 Blow-off silencer
21 Noise-attenuating hood
22 Ventilation fan with silencer
23 Lube oil supply unit (see Fig. C-362)
24 Discharge isolating valve
25 Jacking oil pump

FIG. C-361 Remote setting of adjustable stator blades used to control process flow. (Source: Sulzer-Burckhardt.)



Scope of supply and functional description

1 *Lube oil tank* with auxiliary equipment
2 Electric oil heaters
3 Oil mist fan
4 Filling sieve and breather
5 Degasifier plates
6 Suction strainers
7 *Safety arrangement* (4 non-return valves) in the main oil pump suction and discharge lines to prevent backflow of oil if the compressor and hence the main oil pump should accidentally turn in reverse direction
8 *Auxiliary oil pump*, electric-motor-driven; automatic start up of the pump in the event of oil pressure dropping
9 *Twin oil cooler*, each of the coolers being sized for full flow, with transfer valve for cooler changeover during operation
10 Oil temperature control valve, maintaining constant the oil cooler oil outlet temperature by bypassing more or less the oil around the cooler
11 Oil pressure control valve, maintaining constant the lube oil supply pressure
12 *Twin oil filter*, each of the filter screens being sized for full flow, with transfer valve for filter changeover during operation and pressure loss indicator as well as alarm PDIAH
13 Testing device for checking the automatic start up of the auxiliary oil pump
14 *Control oil tank* with auxiliary equipment
15 Electric oil heaters
16 Filling sieve and breather
17 Degasifier plates
18 Suction strainers
19 *Control oil pumps*, motor-driven; one pump in operation, the other one acting as standby pump, starting automatically if control oil pressure drops
20 *Control oil filters*, two in parallel, each one sized for full flow, with transfer valve for filter changeover during operation and pressure loss alarm PDAH
21 *Testing device* for checking the automatic start up of the standby pump
22 *Bladder-type pressure accumulators* assuring that the compressor stator blades are closed properly in the event of control oil pump failure

Line symbols

- Process main
- Gland balancing line / Gland leakage line
- Lube oil supply
- Jacking oil
- Lube oil return line
- Oil mist
- Control oil supply
- Control oil return line
- Cooling water
- Vent line
- Dust evacuation line
- Instrument air
- Electric/electronic signal
- Mechanical connection

Valve and apparatus symbols

- Globe/ball/disc valve
- Gate valve
- Valve with control trim
- Three-way valve
- Non-return valve
- Non-return valve with bore
- Suction strainer
- Spring-loaded safety valve
- Flow orifice
- Drain plug
- Double acting piston actuator
- Piston actuator with counter spring
- Diaphragm actuator
- Solenoid
- Electric motor
- Pressure accumulator
- Electric/pneumatic converter
- Electric/hydraulic converter
- Low signal selecting relay

Mechanical symbols

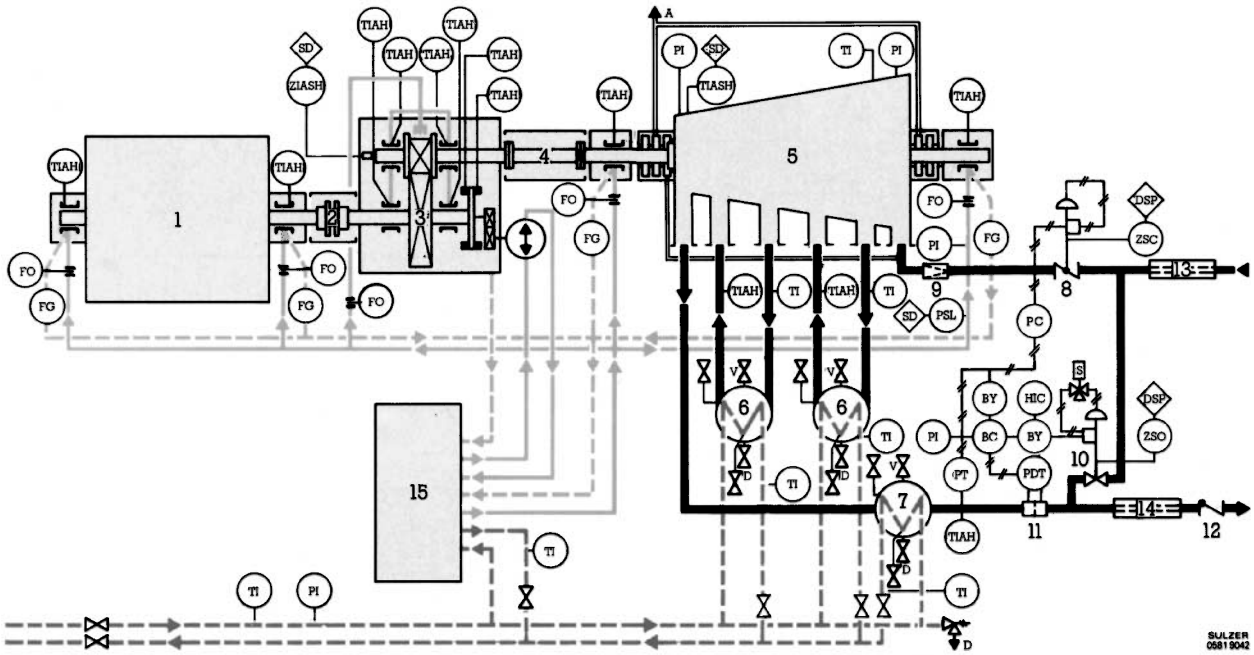
- Flexible coupling
- Solid coupling
- Journal bearing
- Thrust bearing, tilting-pad type
- Gear with thrust collar
- Labyrinth gland chamber

Instrument symbols

Measured variables (first letter)
B Antisurge
F Flow
G Position
H Hand
L Level
P Pressure
PD Pressure difference
T Temperature
Output function (following letter)
A Alarm
G Sight-glass without measurement
I Indication
S Switch function
Extreme values (last letter)
C Closed
O Open
H High
L Low
Signal processing
- Compressor driver start permission
- Turning gear start permission
- Compressor driver shutdown
- Compressor driver start up interlock

Abbreviations
A Exhaust to atmosphere
D Drain
V Vent

FIG. C-362 Oil supply schematic for an axial compressor. (Source: Sulzer-Burckhardt.)



Scope of supply and functional description

1 *Driving motor*
2 *Flexible coupling*
3 *Speed increasing gear* with thrust collars, thrust bearing and laterally flanged main oil pump
4 *Solid coupling* with flexible intermediate shaft
5 *Compressor*, RZ-type, with labyrinth glands
6 *Intercoolers*
7 *Aftercooler*
8 *Suction throttle valve* with pneumatic actuator. The valve serves as final control element to control the discharge pressure via pressure transmitter PT and pressure controller PC
9 *Suction strainer*
10 *Antisurge valve* with pneumatic actuator and solenoid valve, the former controlled by the antisurge controller BC. Manual control of the antisurge valve is possible, in the sense "valve to open" (the antisurge controller thereby being overridden) by operating the manual loader HIC
11 *Metering device* (orifice) for measuring the flow. The orifice delivers a differential pressure signal PD which is transmitted to the antisurge controller BC as actual value for flow. The controller set value is shaped as a function of the compressor discharge pressure
12 *Non-return valve*
13 *Suction silencer*
14 *Discharge silencer*
15 *Lube oil supply unit*

Line symbols

- Process main
- Lube oil
- Oil return
- Cooling water
- Leakage
- Pneumatic control signal

Mechanical symbols

- Flexible coupling
- Solid coupling
- Journal bearing
- Thrust bearing, tilting-pad type
- Gear with thrust collar
- Labyrinth gland chamber

Instrument and controller symbols

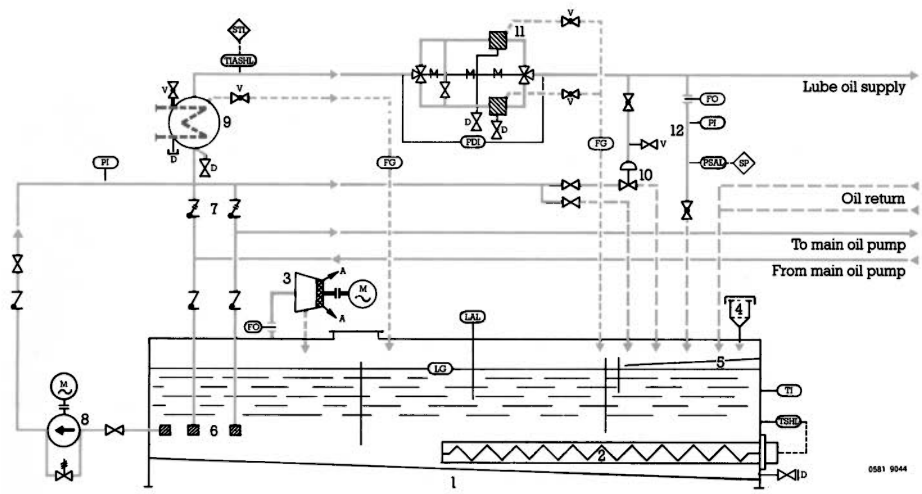
The first letter indicates the measured or initiating variable:
P Pressure
T Temperature
PD Pressure difference
F Flow
Z Position
H Hand
B Antisurge
The second and the following letters indicate the readout or passive function:
I Indication
A Alarm
S Switch function
C Control
G Sight glass without measurement
Y Relay
The last letter indicates the extreme value at which the alarm/switch function is operated:
H High
L Low
C Closed
O Open

Signal processing:
- Driver start permission
- Driver shut-down

Valve and apparatus symbols

- Globe/gate/disc valve
- Three-way valve
- Spring-loaded safety valve
- Butterfly valve
- Non-return valve
- Measuring orifice
- Filter (temporarily mounted)
- Pneumatic valve actuator
- Solenoid valve actuator
- A Atmospheric outlet
- Oil pump
- Cooler
- Orifice

FIG. C-363 Typical P&I diagram of an intercooled compressor. (Source: Sulzer-Burckhardt.)



Scope of supply and functional description

1 *Lube oil tank* with auxiliary equipment:
2 Electric oil heaters
3 Oil mist fan
4 Filling sieve and breather
5 Degasifier plates
6 Suction strainers
7 *Safety arrangement (4 non-return valves)* in the main oil pump suction and discharge lines to prevent back-flow of oil if the compressor and hence the main oil pump should accidentally turn in reverse direction
8 *Auxiliary oil pump*, electric motor driven. Automatic start-up of the pump in the event of oil pressure dropping
9 *Oil cooler*
10 *Oil pressure control valve*, maintaining the bearing oil pressure at 1.5 bar
11 *Twin oil filter*, each of the filter screens being sized for full flow, with transfer valve for filter change-over during operation and pressure loss indicator PDI
12 *Testing device* for checking the automatic start-up of the auxiliary oil pump

Line symbols

Oil supply
Oil return
Vent
Cooling water
Electrical connection
Mechanical connection

Valve and apparatus symbols

Globe valve
Three-way valve
Needle valve
Non-return valve
Non-return valve with bore
Suction strainer
Diaphragm valve actuator
Spring-loaded safety valve
Orifice
Drain plug
A Atmospheric outlet
D Drain
V Vent

Instrument symbols

Measured variables (first letter)
P Pressure
T Temperature
PD Pressure difference
F Flow
L Level
Passive function (following letter)
I Indication
A Alarm
S Switch function
G Sight glass without measurement
Extreme values (last letter)
H High
L Low
Signal processing
Compressor driver start-up interlock
Start-up of auxiliary oil pump

Line symbols

Process main
Oil supply
Oil return
Vent
Cooling water
Gas
Pneumatic control signal
Mechanical connection
Electrical connection

Valve and apparatus symbols

Gate valve
Valve with control trim
Three-way valve
Non-return valve
Line strainer
Diaphragme valve actuator
Valve with control trim
Spring-loaded safety relief valve
Float trap
Restriction orifice
Filter, strainer

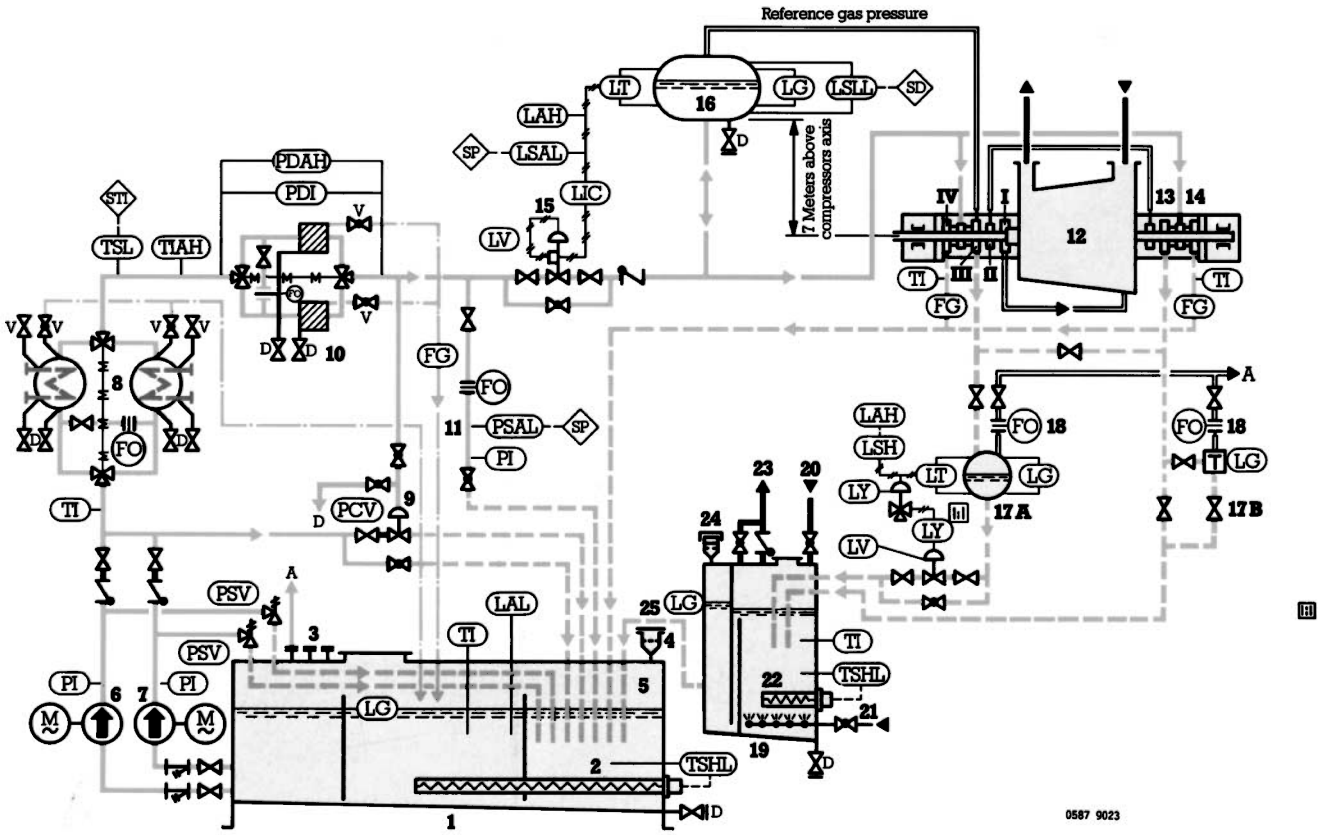
Abbreviations

A Atmospheric outlet
D Drain
V Vent

Instrument symbols

Measured variables (first letter)
P Pressure
PD Pressure difference
T Temperature
F Flow
L Level
Passive function (following letter)
I Indication
A Alarm
C Control
G Sight glass without measurement
V Valve
Y Relay, compute
Extreme values (last letter)
H High
L Low
Compressor driver start-up interlock
Compressor driver shut-down
Start-up of stand-by oil pump
Power amplifier

FIG. C-364 Lube oil supply unit, P&I schematic. (Source: Sulzer-Burckhardt.)



Scope of supply and functional description

1 *Seal oil tank* with auxiliary equipment:
2 Electric oil heaters
3 Vent- and purge connections
4 *Filler* with strainer
5 *Stilling tubes* for oil returns
6 *Seal oil pump I,* electric motor driven
7 *Seal oil pump II,* electric motor driven. During normal compressor operation, only one of the pumps is in operation. The other one acts as stand-by pump and is started automatically in the event of oil pressure dropping.
8 *Twin oil cooler,* each of the coolers being sized for full flow, with transfer valve for cooler change-over during operation
9 *Oil pressure control valve*
10 *Twin oil filter,* each of the filters being sized for full flow, with transfer valve for filter change-over during operation
11 *Testing device* for checking the automatic start-up of the stand-by oil pump
12 *Compressor,* including:
13 Gas-sealed labyrinth glands
14 Oil-sealed floating rings
Function of the labyrinth gland chambers:
I = Balancing piston pressure equalizing chamber, connected to the compressor suction side
II = Pressure equalizing chambers, interconnected, for obtaining symmetrical pressure conditions on the suction and on the discharge side labyrinth gland
III = Mixing (drain) chambers, collecting the seal gas escaping from the chambers II and the seal oil flowing from the floating rings towards the gas side
IV = Drain chambers, collecting the seal oil flowing from the floating rings towards the atmospheric side
15 *Level control valve*
16 *Overhead tank,* located at seven meters above the compressor axis, creating by the static height the necessary seal oil overpressure with reference to the momentary gas pressure. The oil level in the tank is kept constant by the level control valve (15). The space above the oil level is subjected to the gas pressure prevailing in the discharge side gland chamber III (reference pressure).
When varying the compressor suction pressure, the seal oil pressure ahead of the floating rings follows the pressure fluctuations, the seal oil-reference gas differential pressure thereby remaining constant at about 0.5 bar. The tank capacity is sized so that in the event of pump failure the oil supply to the seals is safeguarded until complete compressor standstill.
17A *Gas side seal oil drain pot,* with snap-action level control. Application for gas pressures above 800 psig (55 bar effective)
17B *Gas side seal oil float-type* drain trap. Application for gas pressures up to 800 psig (55 bar effective)
18 *Seal gas venting orifices.* The seal gas flow through the orifices prevents the seal oil from penetrating into the compressor.
19 *Degassing drum,* optionally equipped according to purchaser's specifications with:
20 Purge gas connection
21 Nitrogen bubbling facility
22 Electric or steam heater
23 Gas vent with overpressure relief valve
24 Breather/filler
25 Degasified oil overflow to the tank

FIG. C-365 Seal oil system for floating ring seals; API equivalent system. (Source: Sulzer-Burckhardt.)

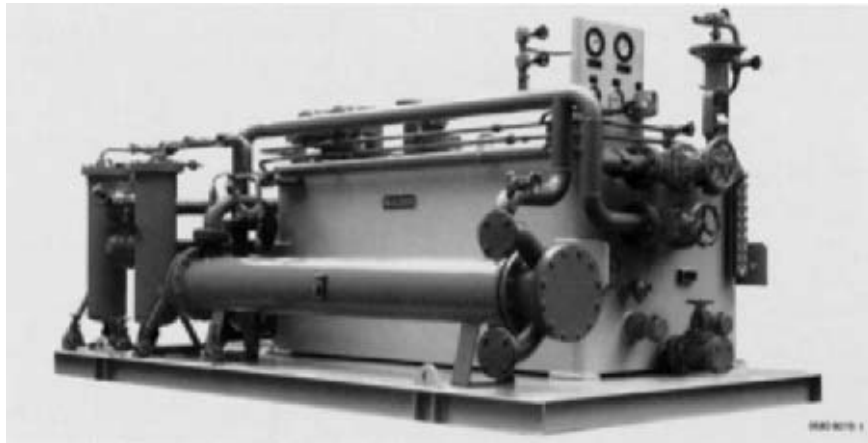


FIG. C-366 Lube oil unit. (Source: Sulzer-Burckhardt.)



FIG. C-367 Lube oil unit according to API 614. (Source: Sulzer-Burckhardt.)

possible because of potential overspeed of the turbine after the release of the coupling.

A method for protection of gas turbines against overtorque and overspeed is described below. The overspeed limitation is achieved through the incorporation of a hydrodynamic coupling, acting as a brake.

A gas turbine generator set normally consists of three major mechanical components, a gas turbine, a gearbox, and a generator.

These components are connected with couplings that besides transmitting the torque also must be able to cope with the misalignment and the displacement caused by the temperature gradients in the system.

The generators operate at standard speeds, 1500 (1800) rpm or 3000 (3600) rpm.

The gas turbine speed differs with the individual turbine design from 3600 to 20,000 rpm, typically.

A gearbox that reduces speed is required in practically all generator set designs. The gear ratio can be as high as 12 times and different types of gearboxes are used.

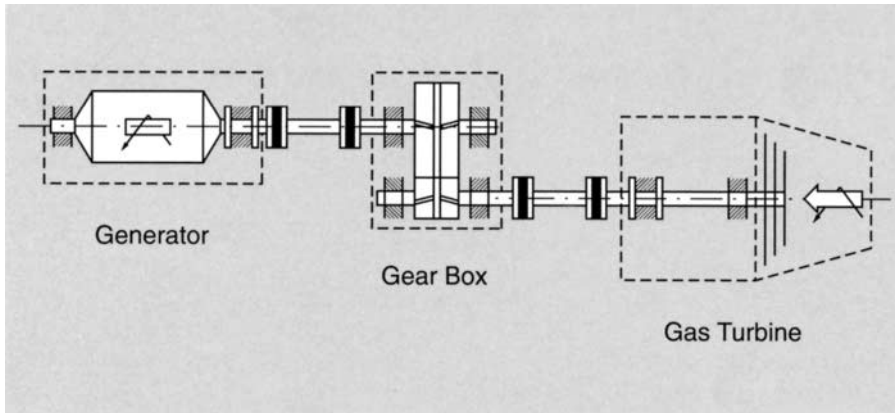


FIG. C-368 Gas turbine generator set, general layout. (Source: J.M. Voith GmbH.)

Aeroderivative gas turbines are based on aircraft engines with only minor design modifications. The lightweight design however also makes the turbines more sensitive to the overloads that can appear when there is a malfunction in the system.

Fault conditions. From a power transmission point of view the drive during normal running conditions can be considered as smooth with small variations in the torque. The overtorque that can appear, and which has to be considered in designing the system, is a rare failure event.

If we discount mechanical failures, the main potential source for overtorque is the generator.

Electrical fault conditions in the generator can produce a large overtorque that is transmitted back to the system: the turbine, the gearbox, and the power transmission components.

The electric fault possibilities are

- Malsynchronization
- Short circuit

Both events involve torque peaks at the generator output shaft of a magnitude ten times full load torque (10× FLT). The peaks are of short duration and the torque is pulsating with the frequency of the generated current. Malsynchronization only gives few torque peaks while in a short circuit situation the pulsation of the torque can go on for some seconds.

The nature and exact size of the torque peaks are well defined and normally known by the generator manufacturer.

How the torque peaks are transmitted backward through the system is governed by the inertia and the stiffness of the components involved.

The situation is complex and a dynamic analysis of the torque fault conditions is normally required for determination of the torque that reaches the gas turbine.

Torque-limiting requirements. The turbine itself, which also is the most costly item, is in many cases the weakest link that has to be protected. The requirement for

limiting the torque can in many cases be difficult. As examples, both the Allison 501-KB7 and GE's LM 6000 need protection at approximately 2–2.5 × FLT in certain configurations.

Compared to most other drives protected with torque-limiting couplings the relation between the requested torque limit and the FLT is unusually small. A shearpin coupling is inadequate for such applications.

Basic design. The basic design principle of this OEM's (Safeset®) coupling is to transmit the torque through a frictional joint in which torque capacity is controlled by hydraulic pressure. This coupling type connects a gear to a shaft in Fig. C-369.

If the coupling is exposed to a higher torque than it can transmit over the frictional joint it will slip there. The relative movement of this slippage cuts a valve (shear tube) with a shear ring so the hydraulic pressure, the contact pressure, and consequently the transmitted torque drop to zero. The drop in torque occurs in a few milliseconds.

This coupling has some basic advantages that has made it an appropriate solution in certain gas turbine generator set applications.

- The torque limit is not influenced by high fatigue and remains practically unchanged after a large number of load cycles. The coupling will thus not release unneccesarily.
- The torque limit is adjustable and can be set at low levels, i.e., 1.4–1.6× FLT and thereby protect components that have to operate close to their limits.
- The resetting of the coupling after release is quick and reliable so the downtime of the unit is minimized.

Typical applications outside of the power generation field are very highly loaded steel mill drives and pump drives in the chemical industry, where production downtime costs can be extremely high.

Overspeed and overspeed limits. When a gas turbine is mechanically disconnected from the workload and inertia of the generator it will momentarily increase speed.

The magnitude of the speed increase is controlled by the residual energy in the system, i.e., the amount of fuel that is available and how it progresses to flame out.

The overspeeding is also controlled by the inertia that is accelerated by the residual energy. Therefore there is a significant influence based on where in the drivetrain the mechanical disconnection takes place.

If the separation is made between the gearbox and the generator, the overspeeding gas turbine will have to accelerate not only its own inertia but also the inertia of the gearbox, which will result in a lower peak speed.

Speed is a critical design factor for a gas turbine and any overspeeding requires certain actions depending on how much the speed is exceeded.

Such actions could be:

- Inspection of the turbine
- Removal and complete disassembly

For the operation and for reducing the hazards it is important to reduce the overspeed, and this can be done by including a hydrodynamic coupling in the drivetrain.

The requirements on the turbo coupling are limited by letting the coupling rotate at speed and only react to the speed difference between gas turbine and generator.

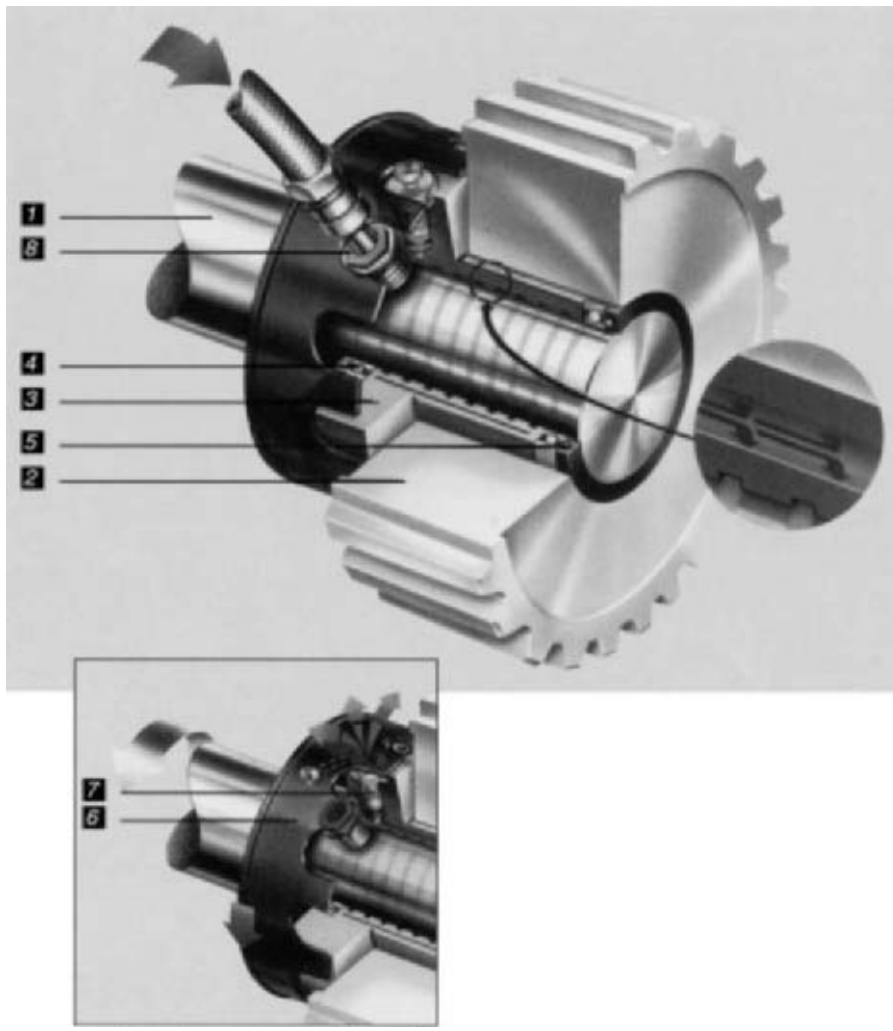


FIG. C-369 Coupling basic design principle. (Coupling is a Safeset™.) 1, shaft; 2, hub; 3, hollow steel sleeve; 4, antifriction bearings (on each side); 5, seal (on each side); 6, shear ring; 7, shear tube; 8, oil charport. (Source: J.M. Voith GmbH.)

The braking torque is thus acting toward the relatively large inertia of the generator.

The hydrodynamic principle. The torque transmission behavior of a hydrodynamic coupling (turbo coupling) is dependent mainly on the following factors.

- Geometry: profile design, diameter d_p
- Operating fluid: density ρ, fluid level, viscosity ν
- Operating conditions: input speed ω_p, speed ratio (slip) ν, acceleration

The torque transmission behavior of the turbo coupling can be described with the following formula.

$$T = \lambda \cdot \rho \cdot d_p^5 \cdot \omega_p^2$$

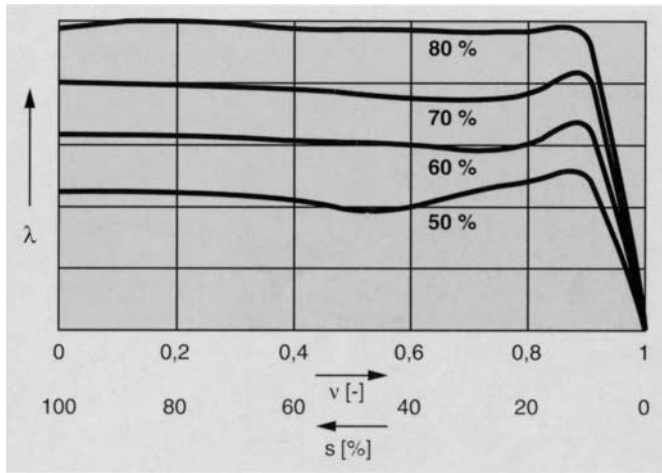


FIG. C-370 Typical slip curves for various filling levels. (Source: J.M. Voith GmbH.)

The performance coefficient, λ, is dependent on fill level, speed ratio (slip), and the profile design.

Typical λ-slip curves for a typical OEM's couplings with various fluid levels are shown in Fig. C-370.

Two main features of the hydrodynamic coupling are the torsional separation and damping.

Input and output speeds or torque fluctuations are dampened or completely separated from input to output side, depending on the frequency.

These features have a positive effect in all applications in respect to the dynamic behavior of the complete system. This will result in lower stressing of component parts and reduced stimulation.

Different applications require specific hydrodynamic coupling designs. For example:

- Constant fill coupling: soft start of electric motors, torque limitation on the driven machine
- Variable speed coupling: control of driven machine speed
- Clutch-type coupling: separating driver and driven machine

Specific coupling and profile designs have been developed to meet the various requirements.

The residual energy in a gas turbine after the release of this coupling will result in acceleration to the turbine because of its relatively low inertia. To keep the overspeed within acceptable limits, a slipping turbo coupling can be used between the gas turbine and the generator, which has a relatively high inertia (Fig. C-371).

For this application the turbo coupling must meet the following design criteria.

- Rapid torque buildup with increasing slip
- High availability

Brake properties at high speed and acceleration. Figure C-372 shows the torque transmission of a turbo coupling versus slip for generator speeds of 3000 and 3600 rpm.

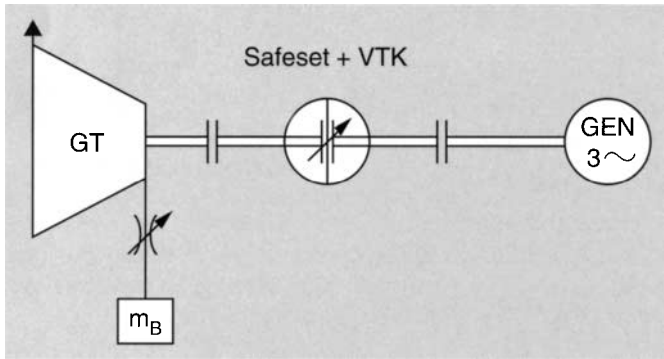


FIG. C-371 Gas turbine drive with Safeset® and coupling (without gearbox). (Source: J.M. Voith GmbH.)

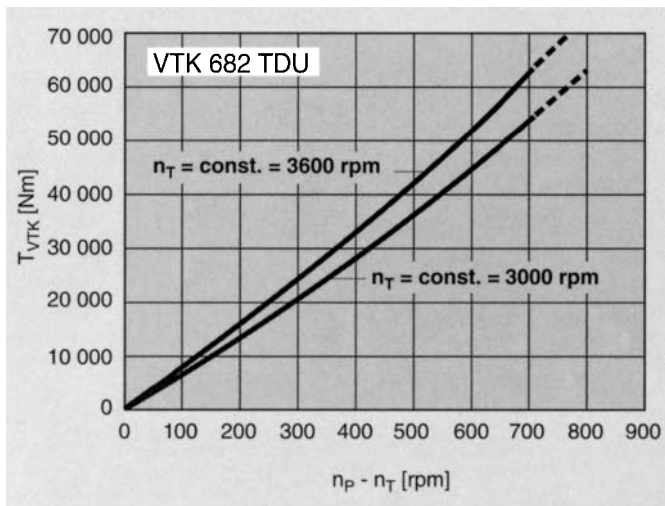


FIG. C-372 Torque transmission of a turbo coupling (Voith VTK) versus slip. (Source: J.M. Voith GmbH.)

The development of the turbo coupling was conducted on a circuit that had good torque transmission capability at very high acceleration. Tests on the circuit design were carried out up to a slip of 16 percent and a maximum acceleration of 6000 rpm.

The features of the turbo coupling unit include:

- Resetting of the system after release
- Self-contained unit, easily removed from the drive system

Figure C-373 shows a compact design for this unit with incorporated turbo coupling.

The flanged-sleeve 1 on the input side is connected via the intermediate sleeve 3 to the flanged shaft 2 on the output side. The serration connects the sleeve 3 to the output shaft. A friction joint connects the input shaft to the sleeve 3.

The friction forces are generated by pressuring the hollow sleeve 4. The slipping torque can be set by varying the oil pressure in the hollow sleeve.

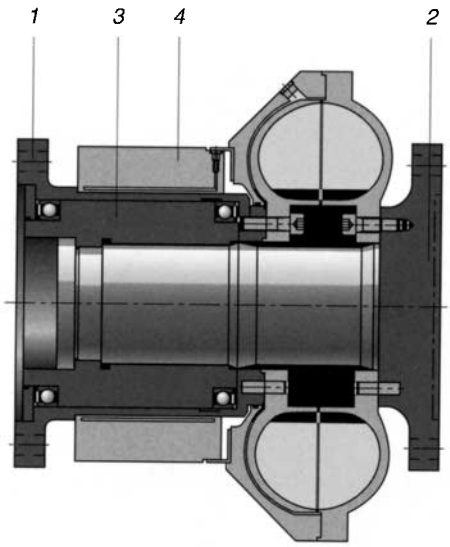


FIG. C-373 Design of safety device consisting of Safeset® coupling and turbo coupling. (Source: J.M. Voith GmbH.)

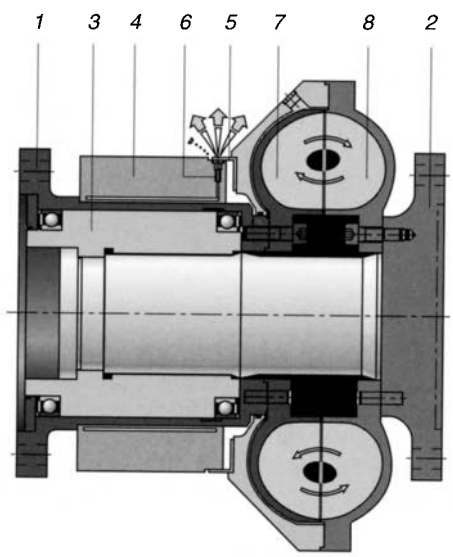


FIG. C-374 Safety device after overload occurred. (Source: J.M. Voith GmbH.)

After reaching the maximum transmittable torque the input side will rotate relative to the output side. The relative movement (slip) is used to cut open the head of valve 6 (shear tube). The oil pressure in the hollow sleeve is released and the torque transmission is interrupted. The pump-wheel 7 of the turbo coupling is connected to the flanged sleeve (input) and the turbine wheel 8 is connected to the flanged-shaft (output). The acceleration of the gas turbine results in a speed difference between the coupling wheels that generates a torque as shown in Fig. C-372. The torque is almost proportional to the slip. (See Fig. C-374.)

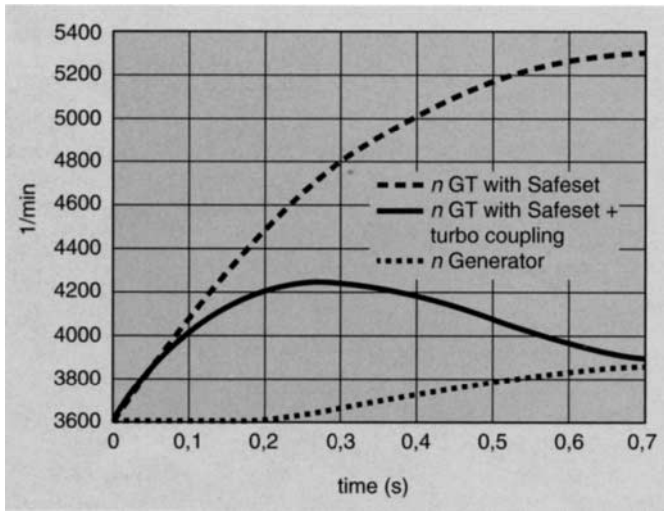


FIG. C-375 Speed response of the gas turbine and the generator with and without safety device. (Source: J.M. Voith GmbH.)

This OEM's (Safeset®) *turbo coupling unit* is designed in such a way that it can be mounted between two membrane couplings. This allows the assembly and removal of the unit without disturbing the gearbox or the gas turbine.

Simulations of LM 6000 fault events. Figure C-375 shows the speed response of a LM 6000 gas turbine and generator using the torque speed characteristic (Fig. C-372) of a turbo coupling (Voith turbo size 682).

The speed response without a turbo coupling is also shown. The significantly lower speed using a Safeset® and turbo coupling can clearly be seen. The calculation assumes the following data are known.

- Inertia of input side
- Inertia of output side
- Disconnection time of the generator
- Losses in the generator (drag torque)
- Torque/speed/time behavior of the gas turbine considering the acceleration

Controls, of Power Supply

Fluctuations and disturbances in a power supply can have expensive consequences for the process engineer. A 2-s power interruption in a semiconductor plant cost over $70,000 in 1997 dollars. The following* cases illustrate the costs associated with power fluctuations.

The power behind thunderstorms can cause problems for industrial facilities where electronic systems that control critical equipment are sensitive to the storms' slight voltage disturbances. These brief voltage sags can disrupt process electronics, resulting in losses in production and costly downtime to recalibrate and restart the

*Source: Adapted from extracts from "Compensating for Lightning," *Mechanical Engineering Power*, ASME, November 1997.

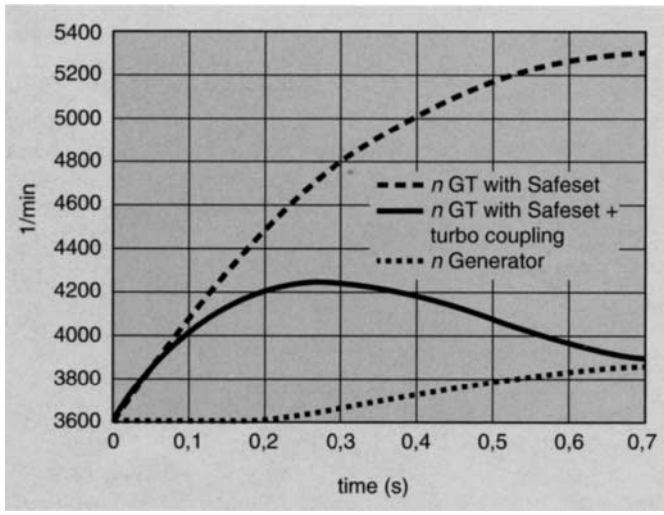


FIG. C-375 Speed response of the gas turbine and the generator with and without safety device. (Source: J.M. Voith GmbH.)

This OEM's (Safeset®) *turbo coupling unit* is designed in such a way that it can be mounted between two membrane couplings. This allows the assembly and removal of the unit without disturbing the gearbox or the gas turbine.

Simulations of LM 6000 fault events. Figure C-375 shows the speed response of a LM 6000 gas turbine and generator using the torque speed characteristic (Fig. C-372) of a turbo coupling (Voith turbo size 682).

The speed response without a turbo coupling is also shown. The significantly lower speed using a Safeset® and turbo coupling can clearly be seen. The calculation assumes the following data are known.

- Inertia of input side
- Inertia of output side
- Disconnection time of the generator
- Losses in the generator (drag torque)
- Torque/speed/time behavior of the gas turbine considering the acceleration

Controls, of Power Supply

Fluctuations and disturbances in a power supply can have expensive consequences for the process engineer. A 2-s power interruption in a semiconductor plant cost over $70,000 in 1997 dollars. The following* cases illustrate the costs associated with power fluctuations.

The power behind thunderstorms can cause problems for industrial facilities where electronic systems that control critical equipment are sensitive to the storms' slight voltage disturbances. These brief voltage sags can disrupt process electronics, resulting in losses in production and costly downtime to recalibrate and restart the

*Source: Adapted from extracts from "Compensating for Lightning," *Mechanical Engineering Power*, ASME, November 1997.

equipment. A pilot project funded by Oglethorpe Power Corp. in Tucker, Ga., and the Electric Power Research Institute (EPRI) in Palo Alto, Calif., tried to eliminate the problem by compensating voltage fluctuations with the PQ2000 energy storage system designed by AC Battery Corp. in East Troy, Wis.

Oglethorpe Power selected the Brockway Standard Lithograph plant in Homerville, Ga., as the site for the first commercial installation of the PQ2000 system. The Brockway facility is a prime location to test the power-compensation system because southeast Georgia has one of the highest rates of lightning in the United States; the flat terrain is also susceptible to high winds and hurricanes that can cause power disturbances.

The Homerville plant houses four production lines, each equipped with high-temperature drying ovens, that are used to cure printed metal for canned products such as Folger's Coffee cans in the United States as well as paint and brake-fluid cans. Fifteen adjustable-speed drives on the four lines control the printing process.

Outages number 30 to 50 times per year due to storms at the Homerville facility. Three motor burnouts per month, due to poor-quality electrical service after an outage, occur. The outages also triggered the protective devices that turned off the plant's ovens. Plant workers had to purge the oven systems of gas before relighting them, a 15-minute process for each line.

Power disturbances caused both a safety concern and a productivity issue, because workers had to climb a 20-foot ladder to purge the burners.

The PQ2000 system is designed to continuously monitor the utility voltage provided to a commercial or industrial facility. Whenever a disturbance is detected, the system switches and picks up the load, isolating itself and the load from the utility system to protect the load from the disturbance. Once the utility system returns to normal, the PQ2000 system switches the load back to the utility.

Speed is of the essence. The PQ2000 can deliver up to 2 MW in about one-quarter of a cycle (or 1/240 s) to maintain power to critical equipment. Most power disruptions typically last only a few cycles, so the AC Battery engineers designed the power-storage system to dispense power for up to 10 s, ensuring an extra margin of safety.

This system demonstrated its ability to protect plant operations from various utility disturbances ranging from a voltage sag to a complete outage up until successful reclosure. Synchronization is maintained. The PQ2000 and other improvements, such as properly grounded and improved electrical drives, trimmed the Homerville factory's annual electrical costs from a high of $110,000 to $120,000 down to $60,000 to $70,000 (see Figs. C-376 and C-377).

Using this system to correct a 2-s power outage can save a semiconductor manufacturing plant $70,000 in product that would otherwise be lost. The same 2-s interval can cause $600,000 in data processing losses for a computer center, require weeks of cleanup in a glass plant, or corrupt critical patient data at a hospital.

Other power supply improvements*

Harmonic distortion in distribution networks is a growing problem due to the increased amount of low-pulse power electronic equipment going into service.

Power supplies for computers, UPS systems, and fluorescent lamps produce harmonic current which contributes significantly to the harmonics in the network. In low-voltage networks, mainly the third and the fifth harmonics are affected.

---

*Source: This section is adapted from extracts from "Industry Needs Quality Supplies," *International Power Generation*, July 1998.

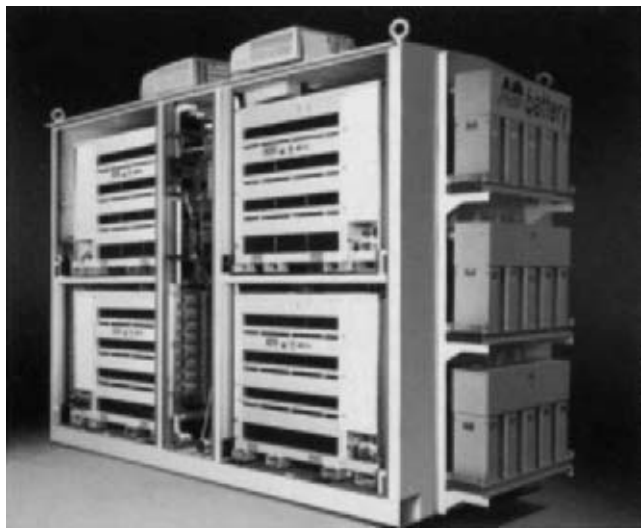


FIG. C-376 The PQ2000 system offsets voltage disturbances caused by storms, thereby preventing costly production-equipment shutdowns. (Source: *Mechanical Engineering Power*, ASME, November 1997.)

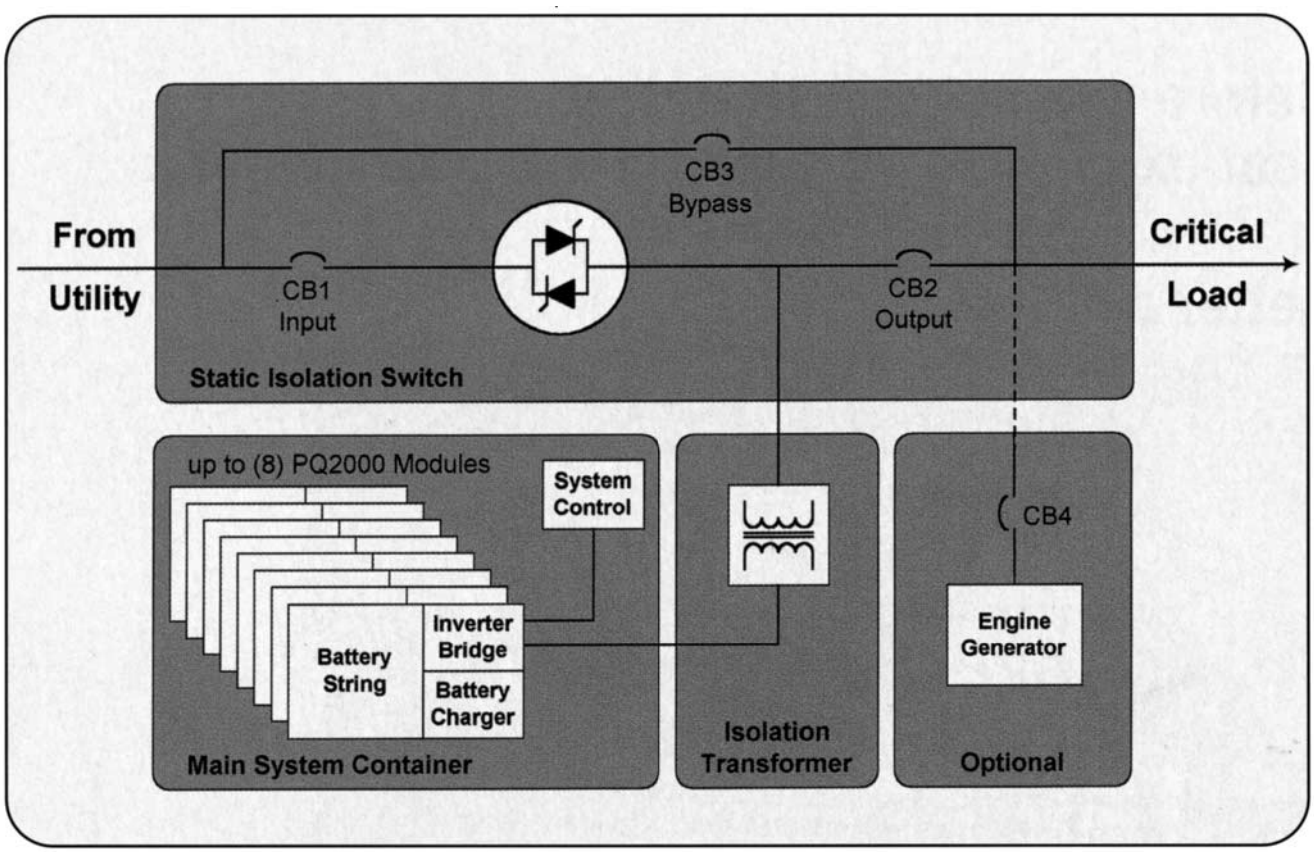


FIG. C-377 Schematic for the wearing of the PQ2000. (Source: *Mechanical Engineering Power*, ASME, November 1997.)

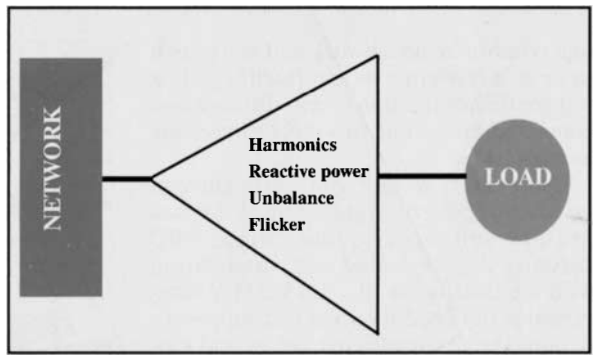


FIG. C-378 Network problems caused by the consumer. (Source: *International Power Generation*, July 1998.)

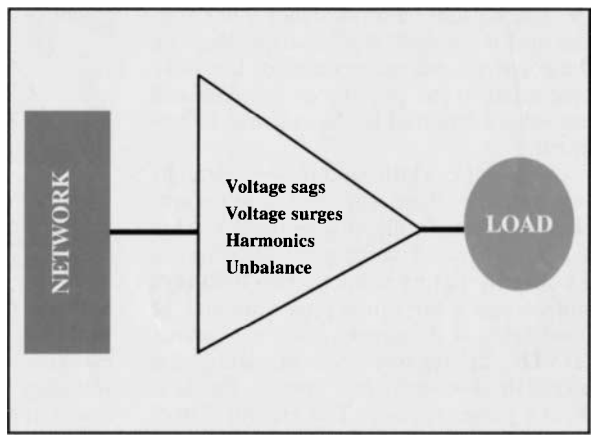


FIG. C-379 Network problems affecting the consumer. (Source: *International Power Generation*, July 1998.)

Business areas with large concentrations of office blocks can generate power quality problems.

Voltage sags are more common than complete outages. When major faults do occur on the distribution system, the customer's voltage can drop significantly (20 to 30 percent) below its nominal value on one or more phases. The embedded chips in many production processes sense disturbances in this range and can fail to perform. Both sags and complete interruptions can last from 100 ms up to a number of seconds or until the fault is cleared by the auto-recloser. Longer breaks can be put down to reliability, not quality, problems. See Figs. C-378 and C-379.

All the major manufacturers offer solutions to poor power quality. These include two technologies: FACTS (flexible AC transmission systems) and HVDC (high voltage direct current). Both use power electronics and are set to develop quickly. The use of power electronics makes it possible to design equipment that allows fast and flexible control of power flows through AC transmission systems, giving continuous control of active/reactive power and increasing network capacity, stability, and quality. HVDC is a proven technique employed by most electric power transmission organizations for a variety of reasons.

It is used on systems with long transmission lines for coupling dissimilar AC networks, and for submarine cables. There are now distinct possibilities for using DC converters to improve network power quality.

HVDC equipment takes a supply from one point in an AC network and converts it to DC in a converter station (rectifier). It is transmitted over a line of any distance and converted back to AC to supply a receiving AC network.

Using direct voltage and direct current, no reactive power is transmitted, line losses are low, and power quality is high. The OEM recently demonstrated a DC application with the installation of a 10-kV, 3-MW compensator designed for specialist supply situations such as infeeds to cities and supplies to small isolated communities.

At the heart of the system is a voltage sourced converter, which is a DC transformer of sorts, with the relationship between direct input voltage and the output voltage dependent on the relative conduction times of the valve connected to the positive DC terminal and the valve connected to the negative DC terminal.

Using pulse width modulation, most output voltage waveforms can be synthesized. Specifically, a sinusoidal voltage can be generated, which means that unlike a conventional HVDC converter, a voltage source converter can supply a passive AC load from a DC source. Such a device (HVDC Light) was installed at a Swedish steel mill to improve the network's power quality. The steel mill was the source of many power quality problems arising from the operation of its electric arc furnaces that affected surrounding users. Voltage flicker, harmonics, and current unbalance are a long standing complaint of neighbors of steel mills.

The converter stations, rated at 3 MW at ±10.5 kVdc are connected on a 10 km AC transmission line. The installation will not only reduce quality problems on the local network but improve the mill's productivity, energy consumption, and power factor. This pilot installation will provide the technology for larger applications (initially up to 50 MW).

(Note: Table C-30 is taken from a paper "Power Transmission and How It Is Changing" given by GEC Alsthom T&D Power Electronics Systems for the IEE Power Division in London.)

Controls, Retrofit

Frequently, the retrofit of an entire control system is an efficent way to optimize the performance of a plant. Many turbomachinery packages, including gas- and steam-turbine–driven ones, are in good mechanical working order but need their control systems tweaked to maximize their potential. It may be more cost effective to retrofit the entire control system. Some examples follow.*

Application case 1

The *aeroderivative gas turbine application* control package (see Fig. C-380) replaces older mechanical/hydraulic/electronic/pneumatic aeroderivative fuel regulators with a modern, reliable application control package that runs on an advanced PLC-based system. The control package for the gas turbine provides fuel control, bleed valve control, and inlet guide vane control.

Advantages

- *Hardware independent system:* Application control package's portability allows choice of platform, reducing need for additional spare parts and training expenses.

* Source: Petrotech Inc., USA.

TABLE C-30 Comparison of Conventional Equipment and Power Electronic Solutions to Network Problems

Problem	Conventional Solution	Power Electronic Solution
low voltage at heavy load	capacitor	power factor correction	—
high voltage at low load		breaker switched capacitor/reactor	—
low voltage on line outage		breaker switched capacitor	SVC
large voltage variability		tap changer	SVC
voltage variability but location unpredictable		—	relocatable SVC or Statcom
very long line		shunt reactor series capacitor	SVC or Statcom TCSC or SSC
stability limit reached		series capacitor	TCSC or SSC
subsynchronous resonance		detune; reduce series capacitor	TCSC or NGH damper
long distance instability		higher voltage, new lines	HVDC long distance
interarea swings		stabilizing signal in generator excitation	—
unstable interconnection		series capacitor, excitation damping	HVDC back-to-back link
persistent loop flow		open connections, series reactors	HVDC back-to-back link
connect unsynchronized systems		—	HVDC long distance
poor parallel line sharing		series capacitor/reactor or quad booster	—
poor post-fault sharing		breaker switched series	—
continuous need to adjust sharing		capacitor or quad booster	TCSC or SSC
voltage variable and continuous poor sharing		—	thyristor phase shifter
		—	unified power flow controller
fault level limits		series reactors	HDVC back-to-back link
more power needed, but new line impossible		cable, gas duct	convert AC line to DC

Key: SVC, static VAR compensator; Statcom, GTO thyristor-based SVC; TCSC, thyristor-controlled series capacitor; SSC, static series compensator; NGH, subsynchronous damping circuit.

- *Fault tolerant:* Control package is available on ICS Triplex fault-tolerant controllers for critical control applications. Software functionality is extended to 2 out of 3 (2oo3) voting at the CPU and I/O level.
- *Simplified interface to DCS or SCADA:* Communication tasks are handled with a separate, dedicated module in the PLC, increasing data rate and simplifying network installation.
- *Improved fuel regulation:* Fast loop sampling rate, combined with modern digital control techniques improve steady-state setpoint control and reduce overshoot during transients.
- *Improved startup reliability:* Special "lean lightoff" procedure ignites all combustors with essentially 100 percent reliability and with reduced thermal stress.
- *Improved engine temperature monitoring and control:* Advanced statistical algorithms detect turbine hot/cold spots and automatically reject failed thermocouples.
- *Fail-safe features:* Redundant overspeeds, open/short monitoring of mA and thermocouples, read-back monitoring of outputs, and special self-check features improve safety.
- *Nonproprietary interfaces:* Simple 4–20 mA, RTD, thermocouple, and dry contact I/O allow simple interface of existing sequence/protection logic unit, making low-cost partial upgrades practical and system troubleshooting easier.

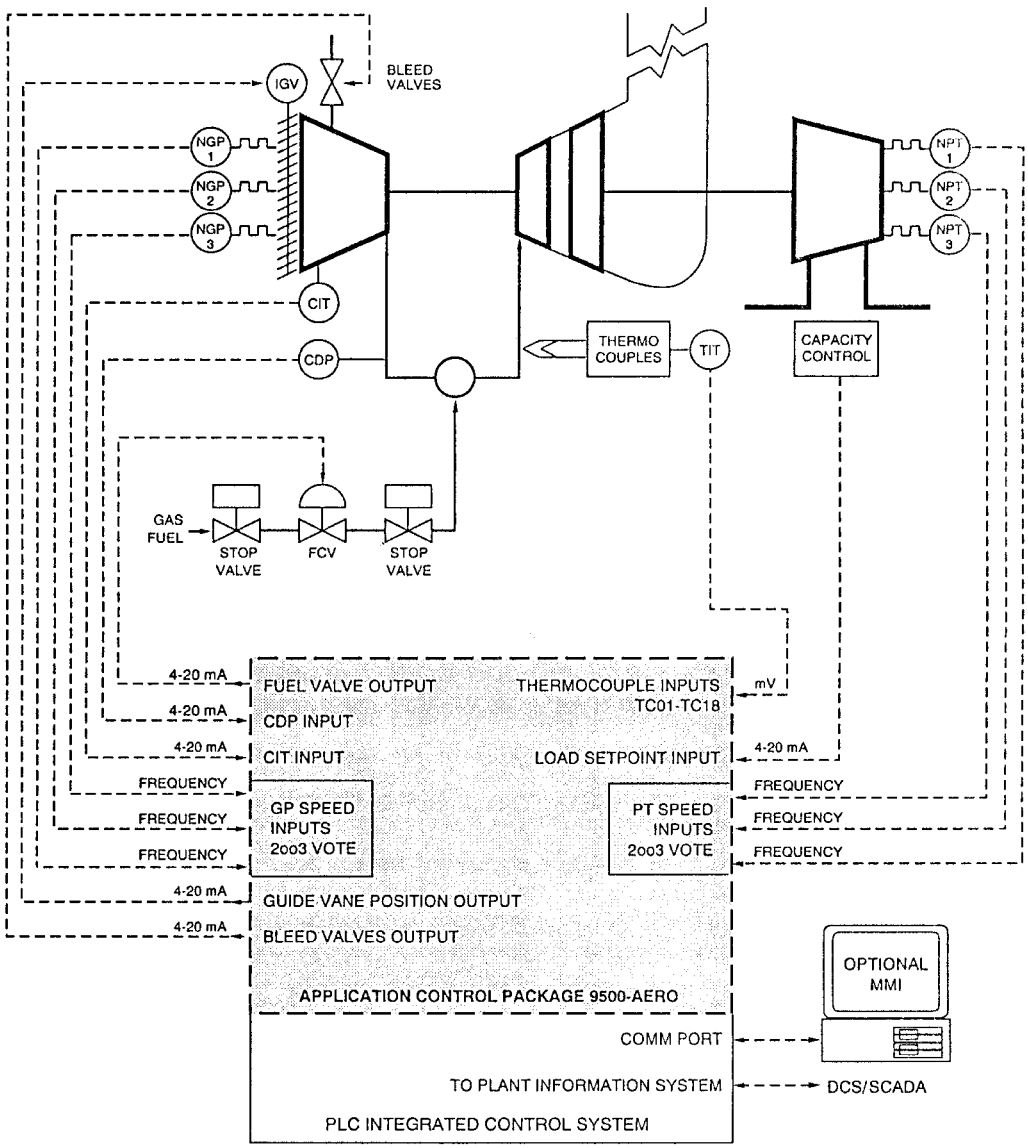


FIG. C-380 Simplified schematic showing an aeroderivative gas turbine compressor drive application control package integrated into an advanced PLC-based control system. (Source: Petrotech Inc.)

- *Improved operator information with optional MMI:* Optional Man-Machine Interface (MMI), MS Windows-based graphic operator interface, displays system status, trending, and data logging, which can be used as part of a preventative maintenance program.

Scope of supply. The application control package for aeroderivative gas turbine compressor drive system includes:

- *Analog inputs, 4–20 mA:*
  - Load setpoint (capacity control)
  - Compressor discharge pressure (CDP)
  - Ambient temperature (CIT)

- *Analog inputs, frequency:*
  - Three redundant NGP
  - Three redundant NPT
- *Analog inputs, mV:*
  - TIT (up to 18 thermocouples)
- *Analog outputs, 4–20 mA:*
  - Fuel control valve position setpoint
  - Inlet guide vane position setpoint (if applicable)
  - Bleed valve position setpoint
- *Operating states:*
  - Firing
  - Warmup
  - Accelerate
  - Load
- *Status, alarms, and shutdowns:*
  - Fault
  - GP (gas producer) overspeed alarm
  - GP underspeed shutdown
  - GP overspeed shutdown
  - Redundant GP overspeed shutdown
  - ΔGP alarm
  - NPT (power turbine) overspeed alarm
  - NPT underspeed shutdown
  - NPT overspeed shutdown
  - Redundant NPT overspeed shutdown
  - ΔNPT alarm
  - High TIT alarm
  - High TIT shutdown
  - Low TIT shutdown
  - Low TIT delayed alarm
  - Rejected thermocouple
  - Shutdown in the event of too few thermocouples
  - ΔT alarm
  - ΔT shutdown
  - Thermocouple spread alarm
  - Thermocouple spread shutdown
  - Turbine maximum limit
  - Turbine minimum limit
  - GP speed #1
  - GP speed #2
  - GP speed #3
  - GP speed #4
  - GP speed #5
  - NPT speed #1
  - NPT speed #2
  - TIT switch #1
  - Manual
  - High firing fuel pressure shutdown
  - Transmitter failure alarms
  - Transmitter failure shutdowns

  - Output failure shutdowns
  - Control mode
- *Controllers/special features:*
  - Start-up controller for fuel valve
  - NGP (gas producer speed) controller for fuel valve
  - NPT (power turbine speed) controller for fuel valve
  - TIT controller for fuel valve
  - TIT rate of rise controller
  - Fuel acceleration schedule
  - Fuel deceleration schedule
  - Deceleration rate limiter
  - Corrected speed (CNGP) override
  - Inlet guide vane controller
  - Bleed valve controller
  - Combustion monitoring system
  - Stagnation detection system
- *Ramps:*
  - Firing (lean lightoff) ramp
  - Startup ramp
  - Loading ramp
  - Cooldown ramp
- *Does not include:*
  - PLC hardware
  - Compressor application control package
  - Gas turbine sequencing and protection discrete logic
  - Compressor sequencing and protection discrete logic
  - End elements

Options for complete control system upgrade

- Compressor application control package
- Gas turbine sequencing and protection discrete logic
- Compressor sequencing and protection discrete logic
- Communication interface to DCS or SCADA
- Capacity control application control package
- PLC hardware
- Man-machine interface unit with WonderWare InTouch® licensed software package
- Complete custom-engineered control panel, factory tested and ready to install
- Fuel control valve system upgrade
- Acceleration control valve system upgrade
- Inlet guide vane actuator system upgrade or retrofit
- Bleed valve actuator system upgrade
- Thermocouple upgrade
- Vibration system upgrade
- Installation and commissioning
- Training

Application case 2

The Series 9500 integrated control system provides cost-effective complete or partial control system retrofits for *gas turbine–driven generator* packages (see Figs. C-381 and C-382). The Series 9500 system provides replacement controls for outdated electrohydraulic and analog-electronic controls. The PLC-based system can include turbine and generator sequencing, complete turbine control, load control, DCS interface, and a graphical operator interface for system status, trending, and data logging.

Main features are similar to those for the system in the preceding case.

The gas turbine generator control package includes:

- Firing (soft lightoff) ramp
- Startup controller
- NHP controller
- NHP acceleration controller
- EGT controller
- EGT rate of rise controller
- EGT controller for inlet guide vanes (if applicable)
- Combustion monitoring system
- Dual fuel capability with online transfer

Auxiliary systems for gas turbine generator packages. The following auxiliary systems and components are also typically made available for complete or partial system upgrades:

- Fuel control valve system upgrade can include replacement of fuel control valve, fuel speed ratio valve upgrade, addition of a fuel vent valve, compressor discharge pressure transmitter, and interstage fuel pressure transmitter.
- Dual fuel conversions including addition of a gas or liquid fuel valve system.
- Hydraulic servocontrols if applicable, such as second-stage nozzle controls on a GE Frame 3 gas turbine, or inlet guide vane controls on a GE Frame 5 gas turbine.
- Complete second-stage nozzle actuator and hydraulic system retrofit for GE Frame 3, with an increased capacity industrial RAM and servo with accumulator, pumps, and support components integrated into a complete system.
- Speed probe and exciter gear assemblies.
- Flame detectors for combustion chambers.
- Thermocouple retrofits.
- Skidded water or steam injection systems for NO_x reduction.

Application case 3

Retrofit controls systems are available for fully integrated, multiloop, microprocessor-based antisurge control and real-time performance monitoring for multiple-stage (tandem) turbocompressors. See Figs. C-383 and C-384. Additionally, the system can provide a variety of compressor control options (see Figs. C-385 through C-397), which makes it a completely integrated compressor control system.

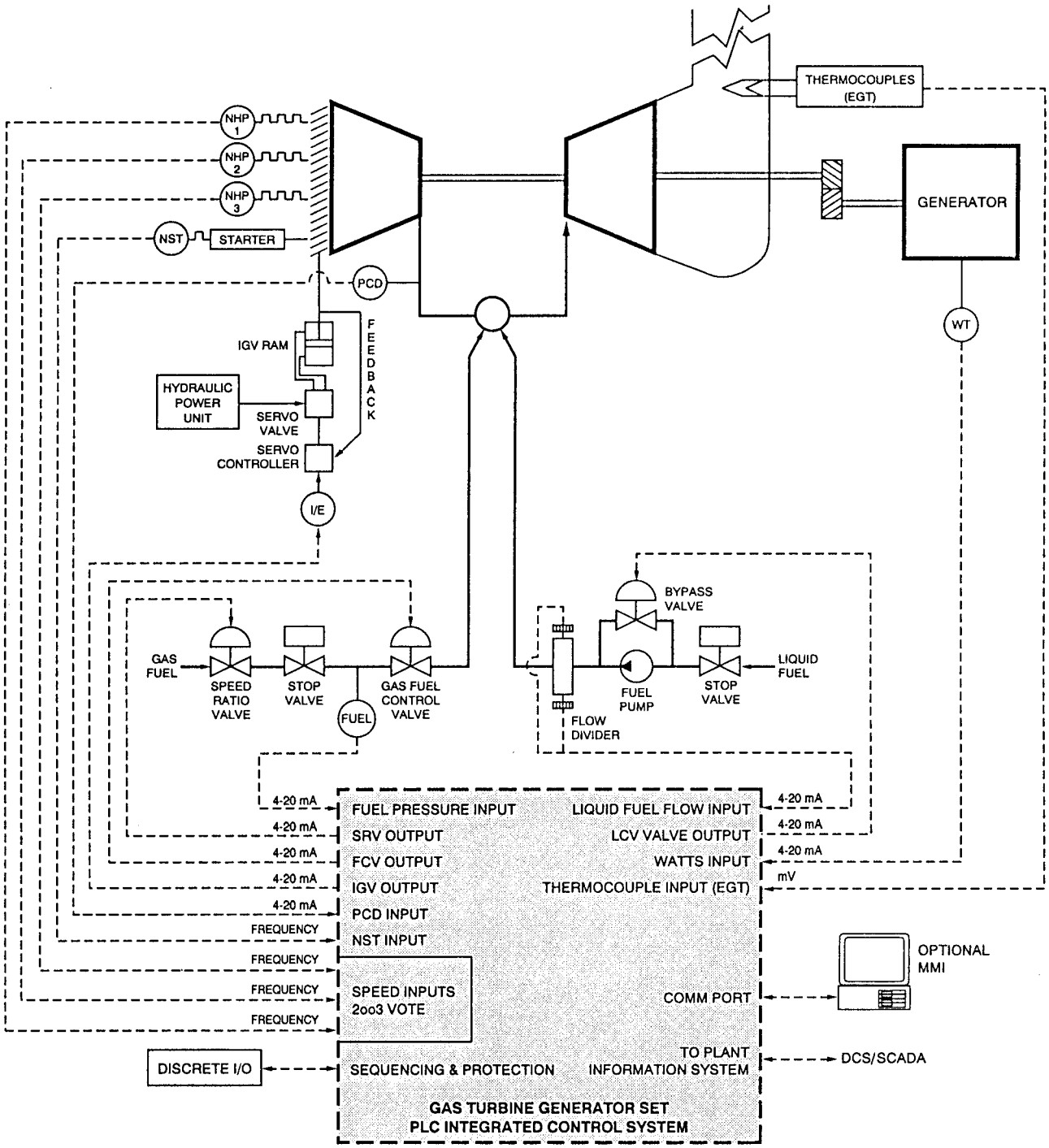


FIG. C-381 Simplified schematic showing a advanced PLC-based integrated control system for a gas turbine generator set. The system provides turbine fuel control, temperature control, sequencing/protection, and communication interfaces. (Source: Petrotech Inc.)

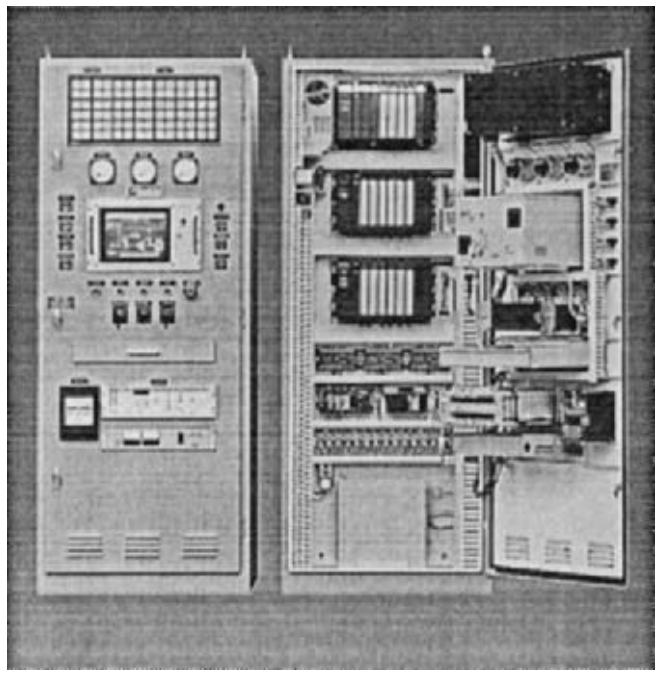


FIG. C-382 Replacement controls for two GE Frame 5 generator sets in utility power generation peaking service. (Source: Petrotech Inc.)

Retrofit system features include

- *Multiple-stage compressor control capability:* Provides integrated compressor control for up to four compressor bodies in a single hardware platform. Eliminates multiple-box control approach and simplifies controller-to-controller communication while also reducing overall system complexity and cost.
- *Built-in, proven algorithms for every application:* Seven built-in algorithms for each independent stage and ability to add customer-defined algorithms for each compressor stage.
- *Advanced control strategies enhance process stability:*
  - Each compressor stage controller is independently optimized.
  - Coordinated control action between stage controllers for runup, rundown, loading, and upsets is much smoother and faster than multiple-box systems.
  - Anticipation-based control and asymmetrically damped control provide superior response to upsets and improved compression process stability.
  - Digital curve fit surge control lines for each stage produce constant safety margins for safe operation and reduced recycle.
  - Adaptive control strategies continuously adjust control safety margins to actual compressor stage operating conditions.
  - Loop-gain linearization allows equal percentage valve trim for improved stability at lower recycle without requiring detuning for high recycle.
  - Valve actuator preload control eliminates delay in surge valve response. Typically, ASC-M3 systems have the valve full open on upsets in 3/4 s or less.
  - PURGE/RUNUP/RUNDOWN coordination feature provides optimum sequence functions without field solenoids, timers, or additional field cables.
- *Compressor efficiency increase:* Energy consumption of driver is reduced by eliminating unnecessary recycle.

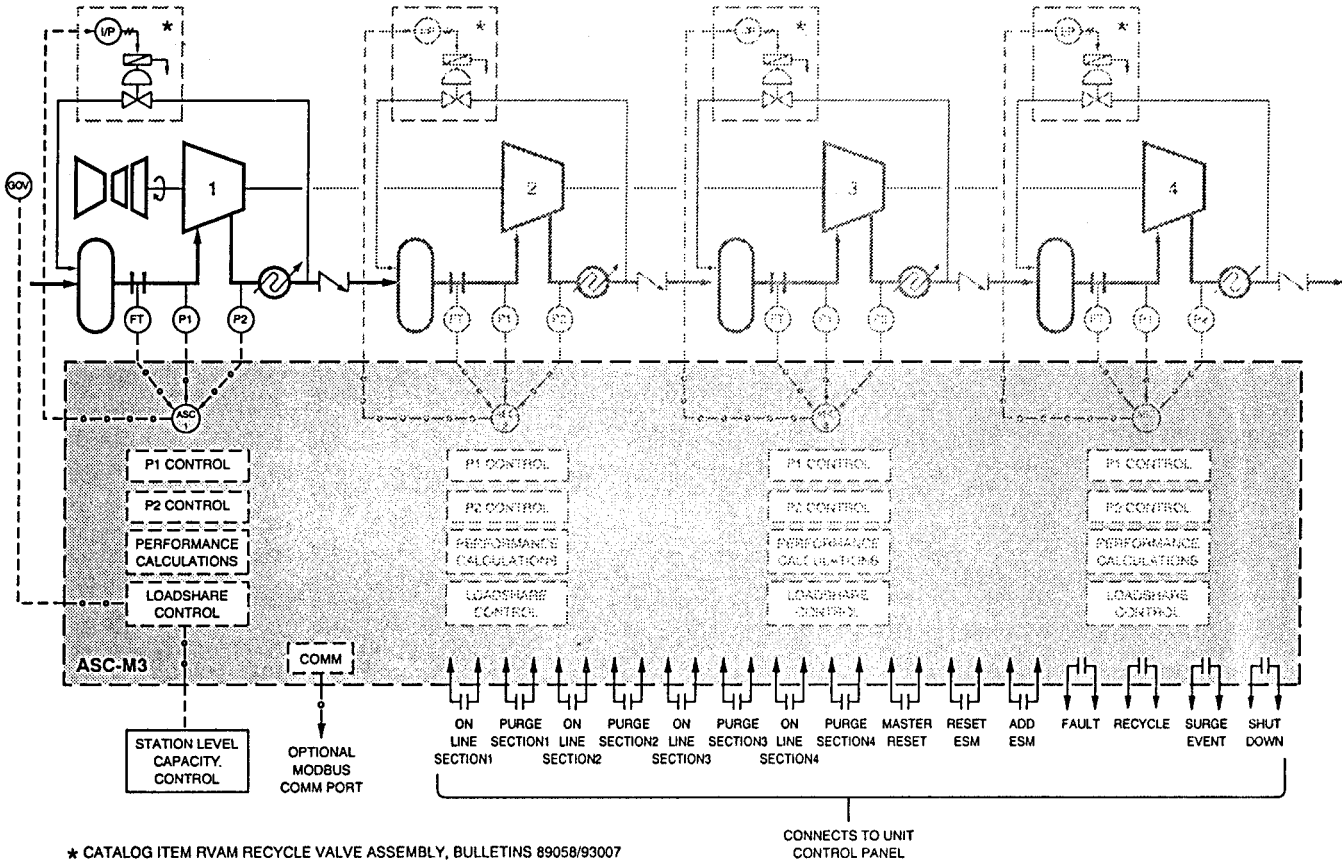


FIG. C-383 Simplified instrument diagram showing one ASC-M3 compressor controller in a four-stage compressor application with a recycle valve for each stage. Controls for each body are independently calibrated and configured per the requirements for the respective stage. A single ASC-M3 can handle compressor control applications ranging from a single-stage compressor up to four independent stages, including various integrated control options and enhancements. This flexibility eliminates a multiple-box approach and reduces overall system complexity and cost. Each compressor body can have a different control algorithm, and can have flow measurement in the suction or discharge. Runup, rundown, purge, loading, and upset control are coordinated between stages. Built-in high-select and low-select functions can combine two, three, or four "controller" outputs to a single recycle valve if required. Each ASC-M3 complete-train compressor controller is individually factory configured with exactly the inputs, outputs, and control functions appropriate for the particular compressor. Each controller requires an Application Engineering Service package, catalog item AES, which provides preliminary calibration and configuration, as well as bench test. As shipped, as configured model ASC-M3 compressor controller typically requires only verification of the field wiring and minor field tuning to be placed in service. (Source: Petrotech Inc.)

- *Integrated compressor control options:* Capability exists for integrated options such as capacity control and pressure override control. Advanced control strategies are easily accomplished at a much lower cost than typical multibox systems.
- *Command initiatives on a per-stage basis:* Individual PURGE and ON-LINE contacts for each compressor stage allow for more complex, efficient loading sequences of multiple-stage compressors.
- *Failed transmitter fallback algorithms:* Fallback algorithm allows continued, safe operation in the event of a critical transmitter failure. Critical transmitters include compressor flow, suction pressure, and discharge pressure.
- *Molecular weight correction:* Automatic surge line compensation for shifts attributable to changes in molecular weight protect against surge during changing inlet gas conditions.

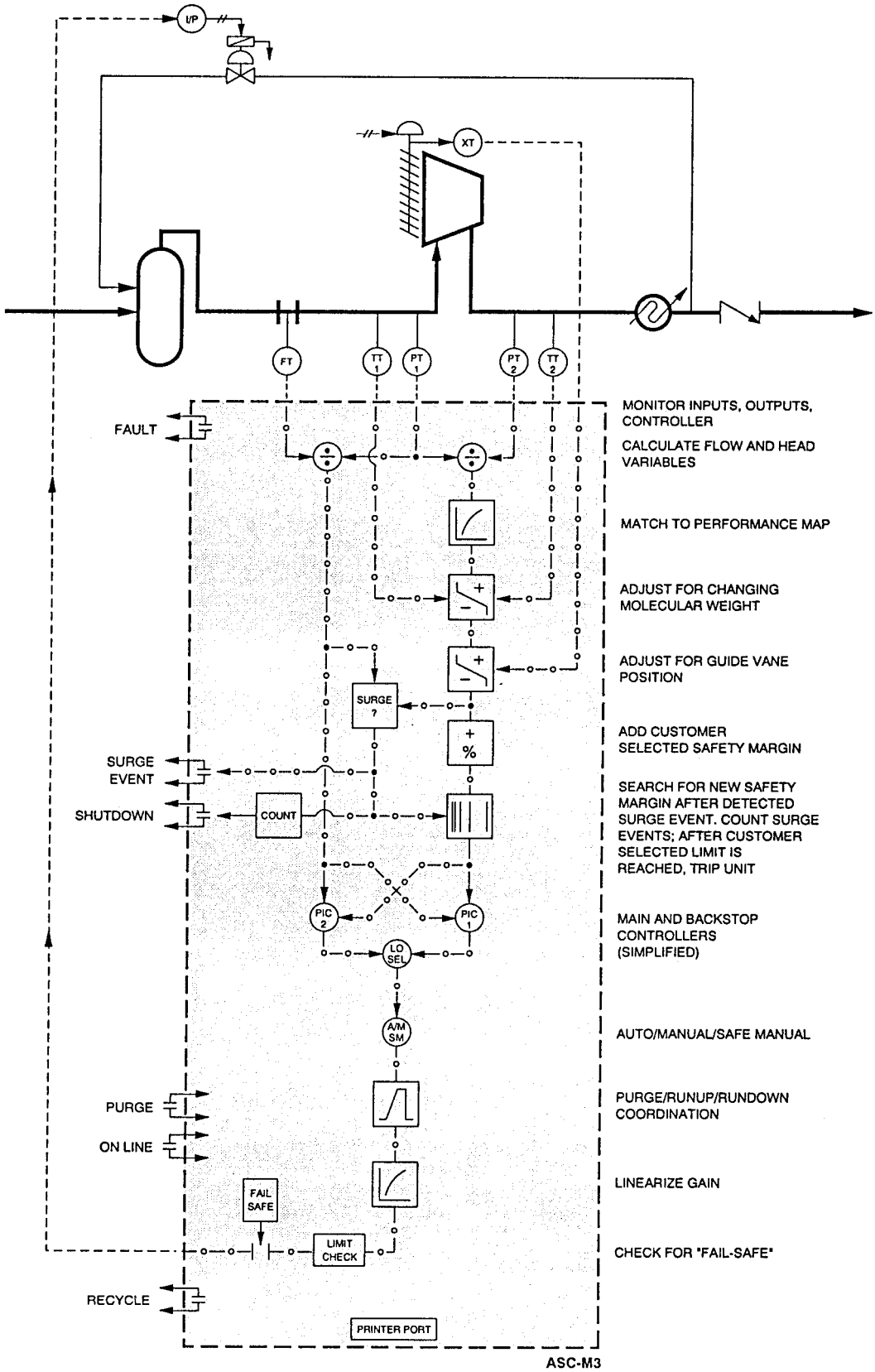
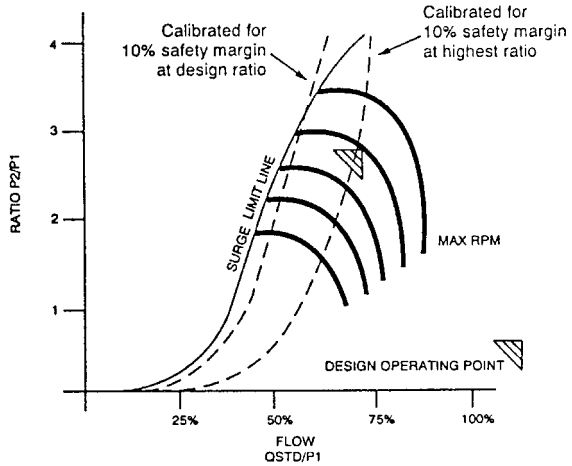
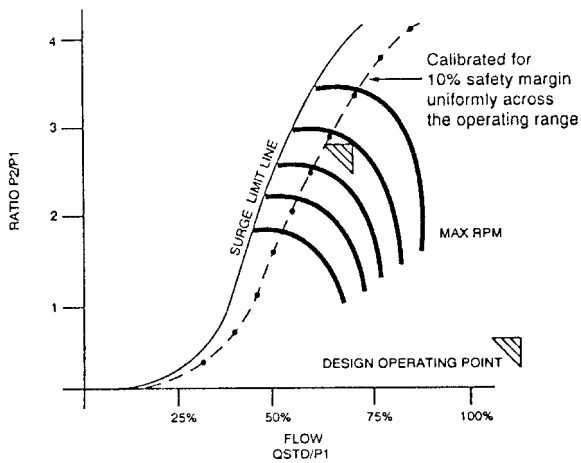


FIG. C-384 Simplified function block diagram showing the ASC-M3 control features for a single compression body. Discrete outputs and the printer port are common for all stages. (Source: Petrotech Inc.)



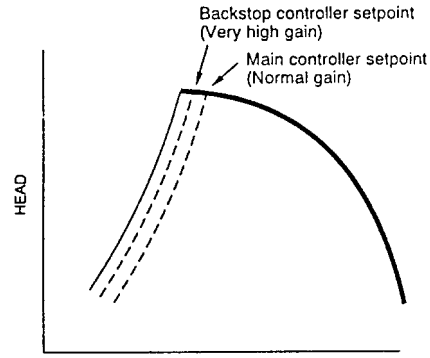
385 TRADITIONAL SURGE CONTROL STRATEGY



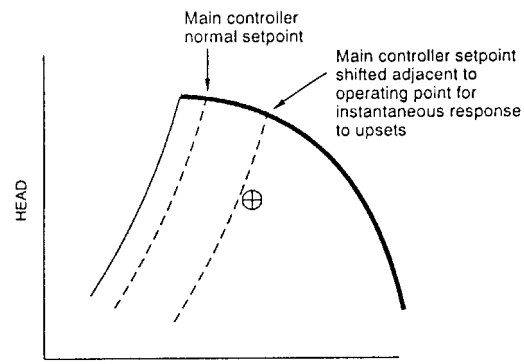
386 DIGITAL CURVE FITTING STRATEGY OF ASC-M3 USING ADVANCED ALGORITHM TO AVOID SHIFT

Compressor performance curves showing a 10 percent safety margin established at design ratio, and 10 percent safety margin at the highest ratio (FIG. C-385). Calibration of 10 percent at design ratio results in a loss of safety as ratio increases. Calibration of 10 percent at the highest ratio results in excess recycle and loss of efficiency. This information source's method (FIG. C-386) of digital curve fit results in a uniform safety margin across the entire operating range with no loss of efficiency due to excess recycle. (Source: Petrotech Inc.)

- *Incipient surge detector:* Detects mild surge and takes corrective action before a violent surge occurs. The incipient surge detection algorithm is independent of the compressor performance map and therefore is immune to inaccuracies in the compressor's respective map.
- *Increased analog input capability:* Separate transmitter inputs for control and performance monitoring allow flexibility for optimization of control while also maximizing accuracy of performance calculations.
- *Assignable AUTO/MANUAL control block with flexible operator interface:* AUTO/MANUAL station allows the manual adjustment of up to eight controllers from a single location.
- *Fault-tolerant capability:* Hot backup configuration is available for critical control applications via a transfer gate. The transfer gate monitors the health of the main



387



388

Compressor performance maps showing setpoints of the backstop controller (FIG. C-387) and main controller (FIG. C-388). As the compressor ratio approaches the surge safety margin, the low-gain main controller begins to open the recycle valve. Low-gain action maintains stability during periods of normal recycle. During upsets, or when recycle has reached a predetermined level, the high-gain backstop controller opens the recycle valve rapidly. (Source: Petrotech Inc.)

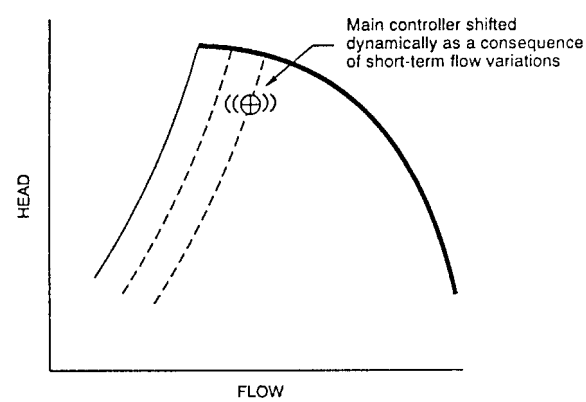


FIG. C-389 If flow is unstable near surge, as with low head rise compressors operating in a process with flow pulsations or variations, the ASC-M3 control dynamically shifts the main control line to maintain the flow envelope on the edge of the calibrated control line. (Source: Petrotech Inc.)

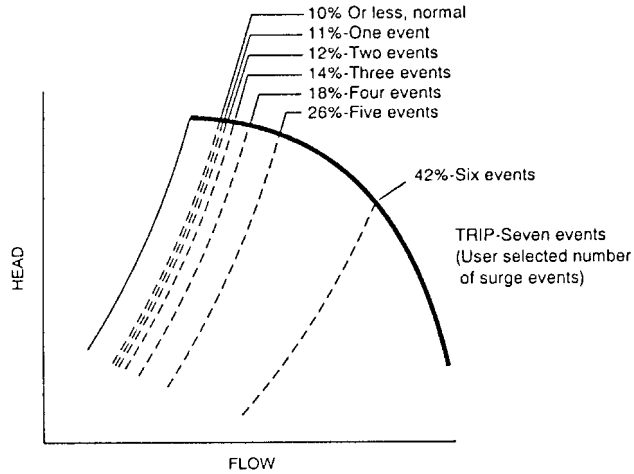


FIG. C-390 After detection of surge, the control system finds the minimum additional safety margin that will provide protection by incrementing safety margin with increasing agressiveness. (Source: Petrotech Inc.)

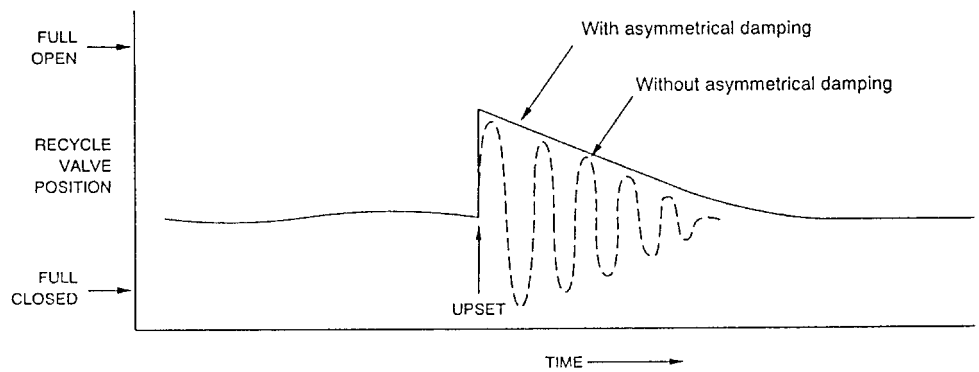


FIG. C-391 Asymmetrical damping critically damps valve response on upset recovery, restoring stability without oscillation, and eliminating unwanted interaction with other loops, without cross-coupling of controller. Asymmetrical damping allows tighter, more responsive tuning which would otherwise lead to system instability. (Source: Petrotech Inc.)

controller and automatically switches to hot backup in the event of a main controller failure.

- *Flexible operator interface display:* Configurable display allows any of the four display windows to be directed to the information on any compressor stage in order to aid troubleshooting.
- *Improved engineer/operator information:* Online compressor performance display option provides more than 33 real-time thermodynamic performance values for each compressor stage to evaluate operating strategies and help identify and track problems.
- *Simplified, accurate calibration:* All-digital calibration is retained in permanent memory, which can be changed at any time via simple keypad entry.
- *Fail-safe features:* Open/short monitoring of mA inputs, read-back monitoring of outputs, and special self-check features improve safety.

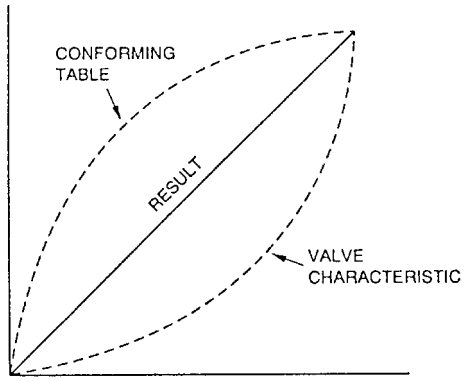


FIG. C-392 This table conforms the ASC controller gain vs. output characteristic to be the mirror image of the valve flow vs. stroke characteristic. Principally, it is used to produce linear overall loop gain while using equal percentage trim valves. Some of the advantages of equal percentage trim valves are much better (more stable) control at low recycle rates; less tendency to oscillate, and bang the valve plug against the seat at very low recycle rates; very high maximum capacity for upset control. Unfortunately, the rising gain characteristic of equal percentage trim valves usually requires that the controller be detuned (more overshoot) at low recycle rages to avoid instability at higher recycle rates. The valve gain conforming table eliminates this disadvantage. (Source: Petrotech Inc.)

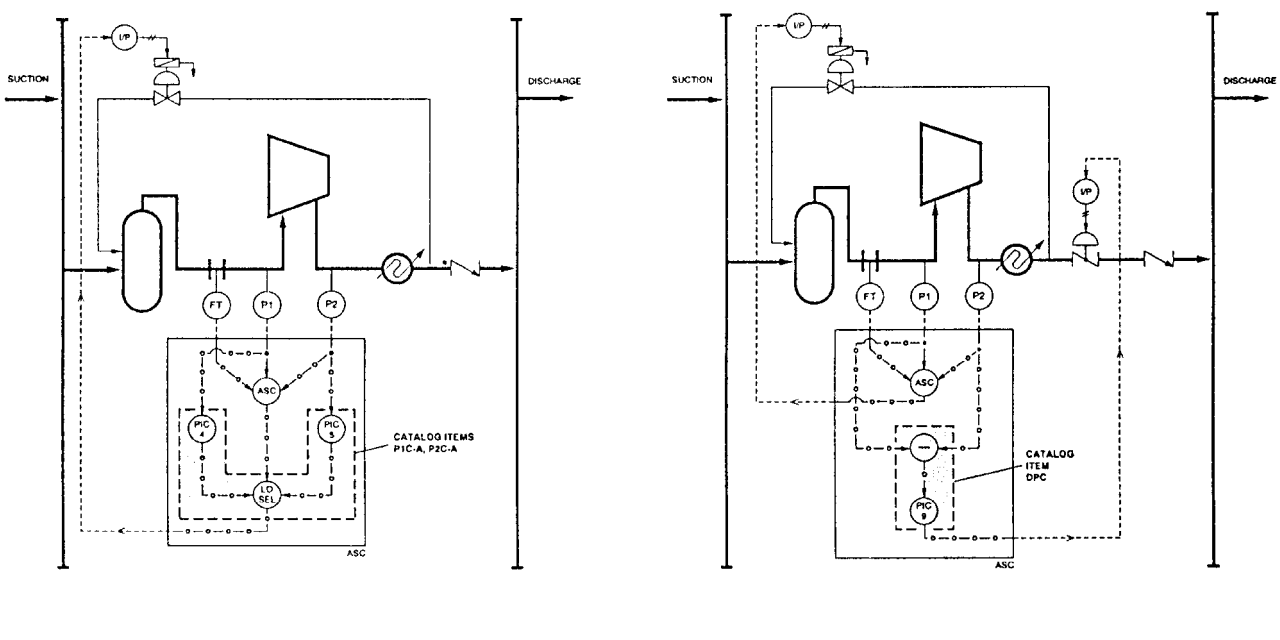


FIG. C-393 Simplified instrument diagrams for selected ASC-M3 options. (Source: Petrotech Inc.)

- *Sufficiently robust:* Independent transient absorbers on every I/O wire, massively oversized power supplies, additional analog and RF filtering, and digital rejection of spurious transients improves ride-through of lightning and other electrical transients, and ensures cool operation for long MTBF.

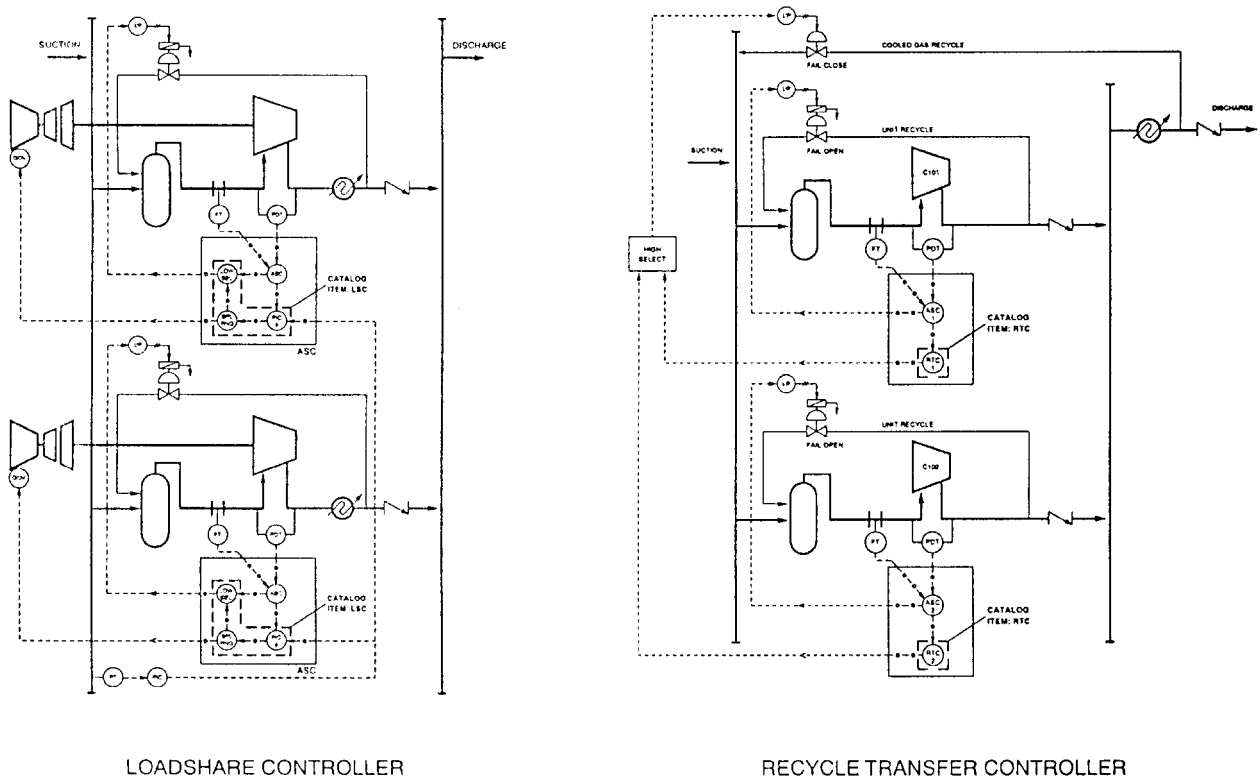


FIG. C-394 Simplified instrument diagrams for selected ASC-M3 options. (Source: Petrotech Inc.)

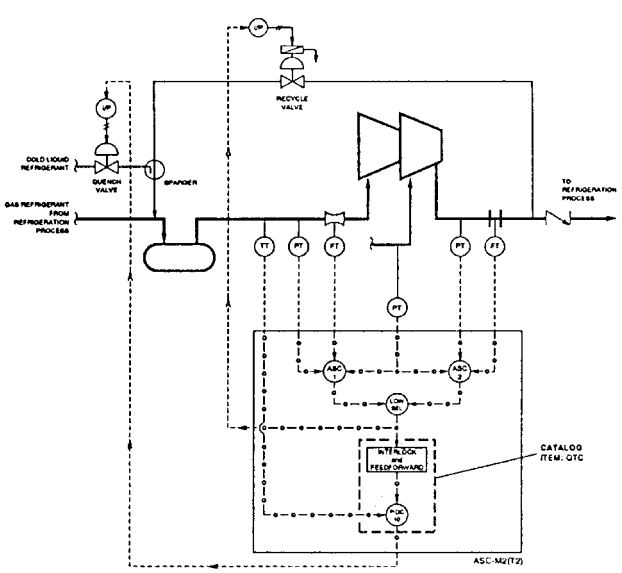
- *Nonproprietary interfaces:* Simple 4–20 mA and dry contact I/O allow simple interface to existing or customer-implemented sequence/protection logic unit, making low-cost upgrades practical, and system troubleshooting simple.

Specifications

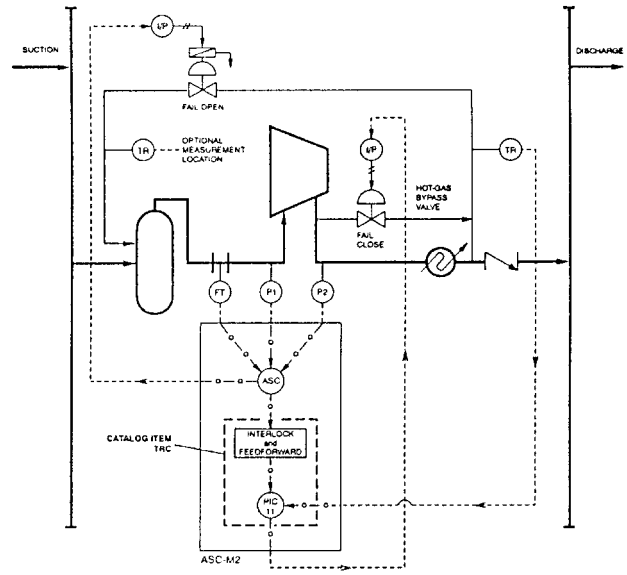
- *Inputs/outputs:*
  - Number of discrete inputs: 16
  - Number of discrete outputs: 8
  - Number of analog inputs: 24
  - Number of analog outputs: 1 standard, up to 7 total
- *Typical electrical system:*
  - Power input: 30 W maximum, basic unit with no options
  - Voltage: 24 VDC nominal (18–32 VDC)
  - Relay outputs: 5 amps at 30 VDC
  - Analog output signals: 4–20 mA, maximum load impedance 800 ohms

Application case 4

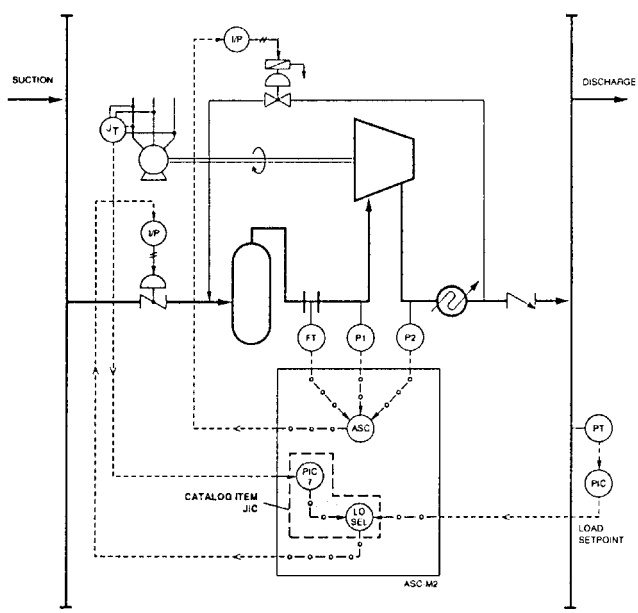
The steam turbine compressor drive application retrofit control package (see Figs. C-398 and C-399) replaces older mechanical/hydraulic/electronic/pneumatic steam turbine speed controllers with a reliable application control package that runs on an advanced PLC-based system. The control package for the steam turbine provides overspeed protection and capacity (pressure/flow) control in a single integrated platform.



QUENCH TEMPERATURE CONTROLLER



RECYCLE TEMPERATURE CONTROLLER



POWER LIMIT CONTROLLER

FIG. C-395 Simplified instrument diagrams for selected ASC-M3 options. (Source: Petrotech Inc.)

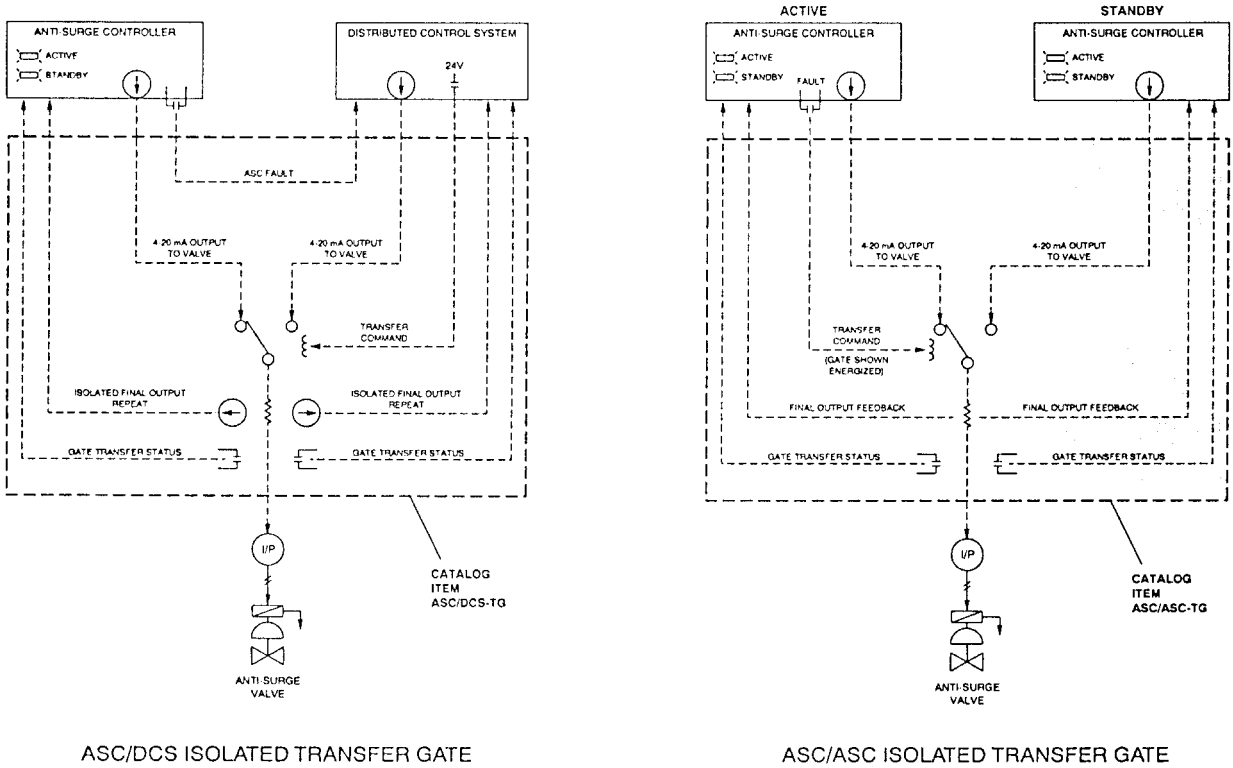


FIG. C-396 Simplified instrument diagrams for selected ASC-M3 options. (Source: Petrotech Inc.)

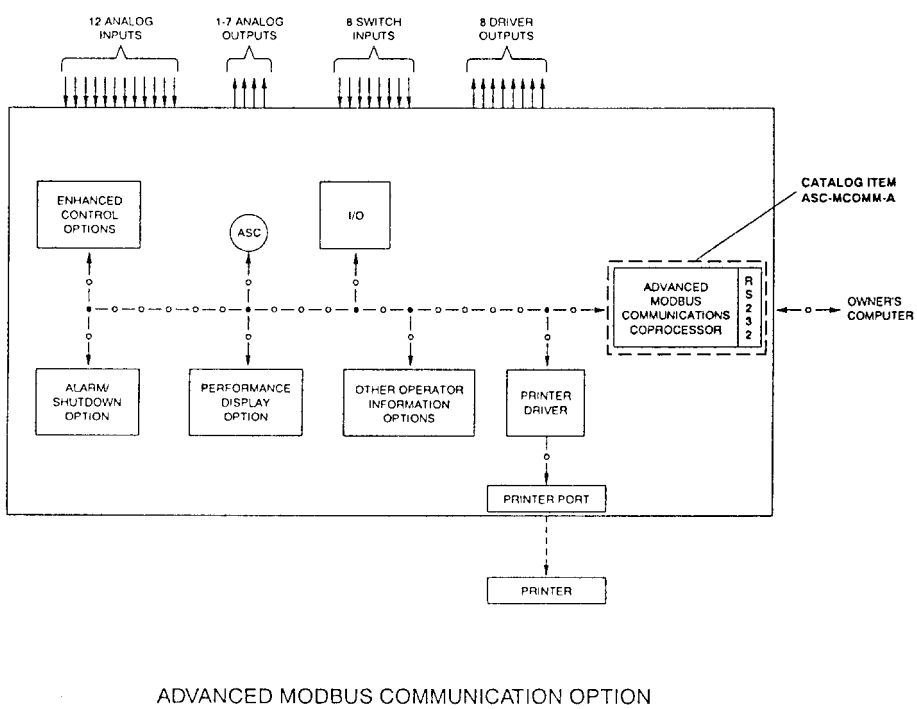


FIG. C-397 Simplified instrument diagram for selected ASC-M3 options. (Source: Petrotech Inc.)

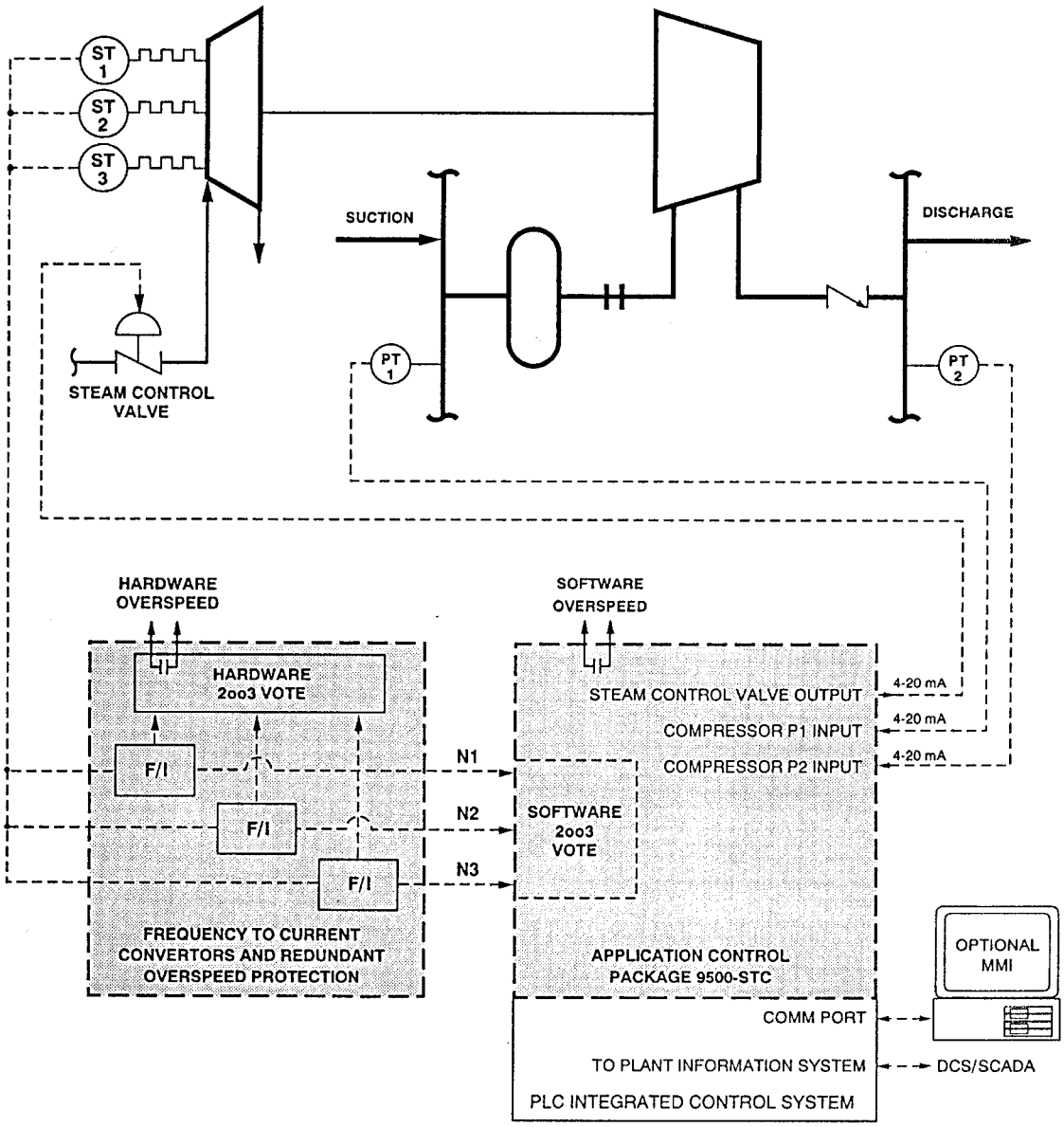


FIG. C-398 Simplified schematic showing a steam turbine compressor drive application control package integrated into an advanced PLC-based control system. The control package provides speed control and capacity control. Overspeed protection is implemented in both software and hardware. (Source: Petrotech Inc.)

System features include

- *Hardware independent system:* Application control package's portability allows customer choice of platform, reducing need for additional spare parts and training expenses.
- *Fault tolerant:* Control package is available on ICS Regent + Plus fault-tolerant controllers for critical control applications. Software functionality is extended to 2 out of 3 (2oo3) voting at the CPU and I/O level.
- *Improved regulation:* Fast loop sampling rate, combined with modern digital control techniques improve steady-state setpoint control and reduce overshoot during transients.

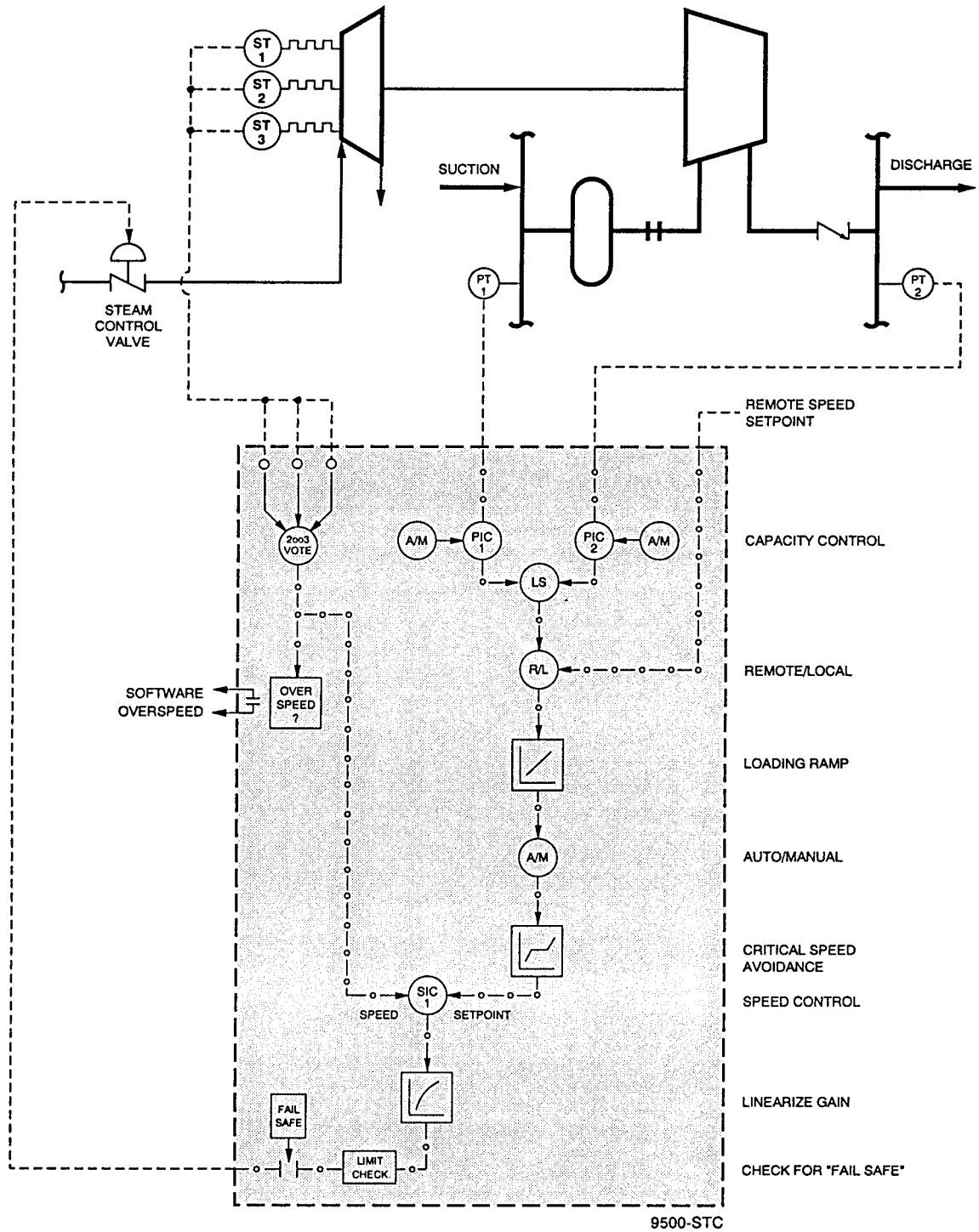


FIG. C-399 Simplified function block diagram showing the control strategy of a 9500-STC application control package. (Source: Petrotech Inc.)

- *Improved overspeed protection:* Hardware and software 2oo3 voting on overspeed shutdowns improves protection while also reducing the likelihood of nuisance trips.
- *Capacity control:* Integrated capacity (pressure/flow) control eliminates the need for additional hardware and communication links thereby providing a cleaner, cost-effective system.
- *Critical speed avoidance:* Two schedules prevent steady-state operation at critical speeds and increases the speed rate of change through criticals on startup to prevent damage due to excessive modal vibration.
- *Simplified interface to DCS or SCADA:* Communication tasks are handled with a separate, dedicated module in the PLC, increasing data rate and simplifying network installation.
- *Fail-safe features:* Redundant overspeeds, open/short monitoring of mA and thermocouples, read-back monitoring of outputs, and special self-check features improve safety.
- *Nonproprietary interfaces:* Simple 4–20 mA, RTD, thermocouple, and dry contact I/O allow simple interface to existing sequence/protection logic unit, making low-cost partial upgrades practical, and system troubleshooting simple.
- *Improved operator information with optional MMI:* Optional MMI, MS Windows-based graphic operator interface, displays system status, trending and data logging, which can be used as part of a preventative maintenance program.

Scope of supply. The retrofit control package for a steam turbine compressor drive system includes:

- Three external frequency to current converters with independent overspeed contacts
- *Analog inputs, 4–20 mA:*
  - Process compressor suction pressure
  - Process compressor discharge pressure
- *Analog inputs, frequency:*
  - Three redundant turbine speed signals
- *Analog outputs, 4–20 mA:*
  - Steam control valve
- *Operating states:*
  - Start/stop
  - Slow roll
  - Accelerate
  - Load/cooldown
  - Active/standby
  - Remote/local
  - Auto/manual
- *Status, alarms, and shutdowns:*
  - Fault
  - Overspeed alarm
  - Overspeed shutdown (software)
  - Redundant overspeed shutdown (hardware)
  - Speed switch #1

  - Speed switch #2
  - Speed switch #3
  - Speed switch #4
  - Speed switch #5
  - Bolt test
  - Manual
  - Transmitter failure alarms
  - Transmitter failure shutdowns
  - Output failure shutdowns
- *Controllers/special features:*
  - Startup control (auto or manual)
  - Slow roll
  - Speed control
  - Capacity control
  - Two out of three (2oo3) voting on speed inputs
  - Critical speed avoidance schedule
  - Remote/local capability

Application case 5

The following retrofit integrated control system (see Figs. C-400 and C-401) provides complete or partial control system retrofits for gas turbine–driven compressor packages. The system provides replacement controls for outdated electrohydraulic and analog-electronic controls. The PLC-based system can include turbine and compressor sequencing, complete turbine control, antisurge control, capacity control, DCS interface, and a graphical operator interface for system status, trending, and data logging.

System features include

- *Integrated control capability:* Fuel control, antisurge control, and sequencing/protection are integrated into a single platform. This eliminates the need for additional hardware and communication links, thereby providing a less complicated, more cost-effective solution.
- *Open architecture system:* Application control package's portability allows customer choice of platform, reducing need for additional spare parts and training expenses. Available PLCs include General Electric, Siemens/TI, and Allen-Bradley.
- *Fault tolerant:* Control package is available on ICS Regent + Plus fault-tolerant controllers for critical control applications. Software functionality is extended to 2 out of 3 (2oo3) voting at the CPU and I/O level.
- *Standard industrial components:* Nonproprietary, commonly available parts are less costly and more easily serviced by customer's on-site personnel. Much longer time to obsolescence than proprietary systems.
- *Reliability:* ALL control functions are performed by tested and proven industrial PLC equipment, not by MS-DOS–based computer equipment which is not designed to function as a controller.
- *Simplified interface to DCS or SCADA:* Communication tasks are handled with a separate, dedicated module in the PLC, increasing data rate and simplifying network installation.

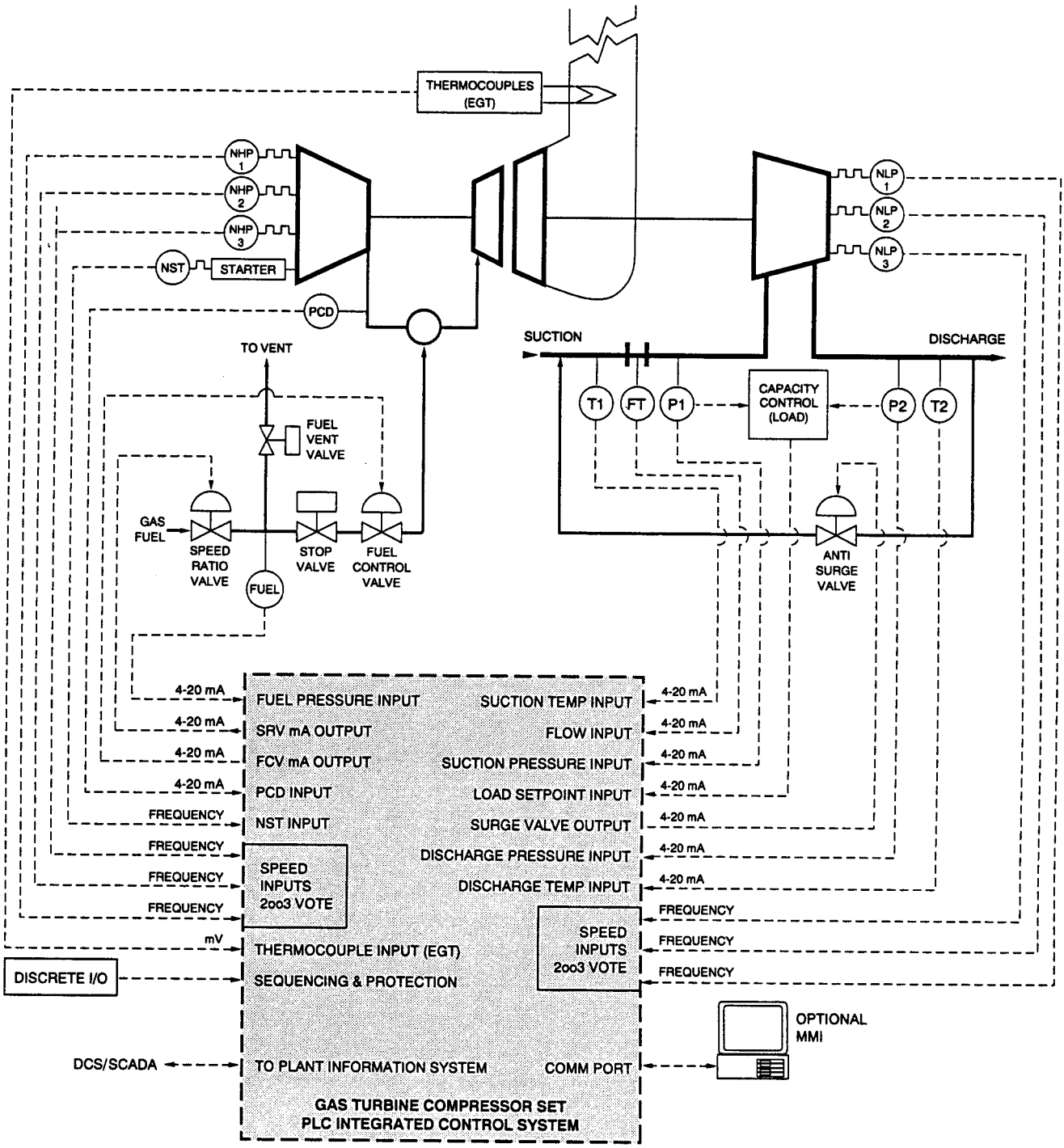


FIG. C-400 Simplified schematic showing an advanced PLC-based integrated control system for a gas turbine compressor set. The system provides turbine fuel control, temperature control, antisurge control, sequencing/protection, and communication interfaces. (Source: Petrotech Inc.)

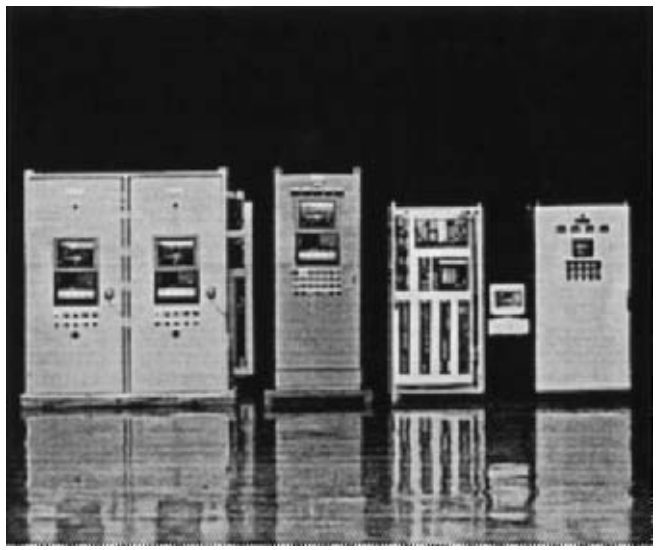


FIG. C-401 Replacement controls for two GE Frame 3, one Pratt & Whitney GG3, and two Solar Centaur gas turbine/compressors. (Source: Petrotech Inc.)

- *Nonproprietary interfaces:* Interfaces in the form of 4–20 mA, RTD, frequency, thermocouple, and dry contact I/O allow simple integration into existing sequence/protection logic controller, making very low-cost partial control upgrades simple and practical.
- *Modified fuel regulation:* Fast loop sampling rate combined with modern digital control techniques improve steady-state setpoint control and reduce overshoot during transients.
- *Modified startup reliability:* Special "lean lightoff" procedure ignites all burners with essentially 100 percent reliability and with greatly reduced thermal stress.
- *Modified exhaust temperature monitoring and control:* Advanced statistical algorithms detect turbine hot/cold spots and automatically reject failed thermocouples.
- *Modified compressor control strategies enhance process stability:*
  - Six built-in, proven algorithms for every application, plus room for a customer-defined algorithm.
  - Anticipation-based control and asymmetrically damped control provide superior response to upsets, and improved compression process stability.
  - Digital curve fit surge control line produces a constant safety margin for safe operation and reduced recycle.
  - Adaptive control strategy continuously adjusts control safety margin to actual compressor operating conditions.
  - Loop-gain linearization allows equal percentage valve trim for much improved stability at lower recycle, without requiring detuning for high recycle.
  - Valve actuator preload control eliminates delay in surge valve response. Typically, Petrotech control systems have the valve full open on upsets in $^{3}/_{4}$ s or less.
  - PURGE/RUNUP/RUNDOWN coordination feature provides optimum sequence functions without field solenoids, timers, or additional field cables.

- *Fail-safe features:* Redundant overspeeds, open/short monitoring of mA and thermocouples, read-back monitoring of outputs, and special self-check features improve safety.
- *Modified operator information with optional MMI:* Optional MMI, MS Windows-based graphic operator interface, displays system status, trending, and data logging, which can be used as part of a preventative maintenance program.

Control Switches*

Temperature switches

Temperature switches are used to monitor or control the temperature of various fluids. Their broadest use is in the range from –65 to +600°F.

A temperature switch consists primarily of a temperature sensor, an electric switch, and a means of causing the switch to be operated by the sensor's reaction to temperature changes. In the switches commonly used in industrial applications the electric switch is mechanically actuated. The sensors are of four types: bimetallic elements, gas-filled bulbs, liquid-vapor-pressure bulbs, and liquid-filled bulbs.

Proper application of temperature switches requires knowledge of the proposed service. This knowledge should include the type of fluid to be sensed, the desired set-point temperature, the extremes of temperature excursions, the need for manual reset, and the maximum pressure level of the fluid. The extremes of ambient temperature and the vibration level at the sensor location should be known. Desired service factors such as accuracy, reliability, life, and response time should also be determined.

The characteristics of the various sensors should be known (see Table C-31).

Bimetallic-type sensor. The bimetallic-type sensor operates on the differential expansion of two dissimilar metals that are bonded together. A common type of unit incorporates the bimetallic element in a helical configuration. One end is fixed so that, when heated, the element expands, causing a rotary motion of the free end. This movement actuates the switch element. One material is usually Invar because of its low coefficient of thermal expansion; the other is usually brass for low temperatures or nickel for higher temperatures. A variation of this type is a strut-and-tube thermostat composed of an outer shell of high-expanding metal and a strut assembly of low-expanding metal.

The application range is generally from –100 to +1000°F. The bimetallic sensor has the advantage of fast response time and is relatively low cost and easily adjusted in the field. It can only be local-mounted, has an accuracy of 2 to 5 percent, can be damaged by overheating, and has low repeatability.

Units based on this principle are used in wall thermostats and other space heating applications. This control generally has wider actuation values than liquid-filled systems. *Actuation value* is the difference between the temperature at which a switch actuates and the temperature at which it releases upon temperature reversal.

Vapor-pressure-type sensors. Vapor-pressure sensors contain a volatile liquid with a vapor pressure that increases with increasing temperature. The increase is independent of the volume of the liquid as long as liquid remains in the sensor to be vaporized.

---

*Source: Demag Delaval, USA.

TABLE C-31 Most Common Types of Commercially Available Temperature Switches

	Type (Range, °F)					
	Bimetallic (–100 to 1000)	Liquid-Filled (–150 to 1200)	Vapor-Pressure (–100 to 550)	Gas-Filled (–100 to 1500)	Thermocouples (–400 to 4500)	Electroresistive (–300 to 1800)
Advantages	Inexpensive, easy to adjust in field	Moderate price, low on-off differential, fast response, small bulb size	Insensitive to ambient-temperature changes	Insensitive to ambient-temperature changes	Wide temperature, rapid response	High sensitivity
Limitations	Damaged by overheating, low repeatability	Ambient compensation required	Relatively expensive, large bulb required, slow response	Relatively expensive, large bulb required, slow response	Expensive accessories required	Expensive accessories required

This type is designed in both local-mount and bulb-and-capillary models. Its general application range is from –60 to +100°F. An advantage is its insensitivity to ambient-temperature changes, and the sensor also has a fairly fast response time. But it has limited ranges, a low safety factor in high-proof temperatures, and larger bulb sizes to perform its function.

Pressure rises very rapidly in the higher portion of the temperature range. Thus, the rate of this sensor is not linear, a factor that severely limits the full range span and proof temperatures. Several types of fluids must be used to cover the full range of temperatures.

It is advisable to use a vapor-pressure sensor only when the ambient temperature is higher than the process-fluid temperature. Then the fluid in the capillary and bellows remains in the vapor state.

Gas-actuated-type sensors. The gas sensor is basically the same as the vapor-pressure type except it does not use a volatile liquid. The system is filled with inert gas, usually nitrogen. The gas pressure increases with temperature. Such systems are used in the range of –100 to +1000°F.

A great advantage of the gas sensor is the long capillary lines that can be used. Although there is an ambient-temperature effect on these lines, it can be minimized by increasing the bulb volume. Gas-filled sensors are not widely used in industry because of their inherent slow response time, large bulb size, and relatively high cost.

Liquid-filled sensors: bulb-and-capillary type. This type of sensor is shown in Fig. C-402. It comprises a bellows and a capillary and bulb assembled to produce a perfectly hermetically sealed system. The assembly is evacuated in a vacuum chamber. The filling fluid, while under vacuum, is allowed to flow into the bellows and the capillary and bulb. The bulb is heated to a specific calibration temperature for the particular sensor. The fill hole is then sealed.

The basic system requirement is linear movement of the bellows with a change of bulb temperature. To achieve this movement the fluid must have a relatively constant coefficient of thermal expansion over the range of sensing.

The relationship of bulb size, bellows size, and coefficient of thermal expansion is important and must be closely controlled to obtain the desired characteristics. The bulb size changes slightly with temperature change. With proper selection of

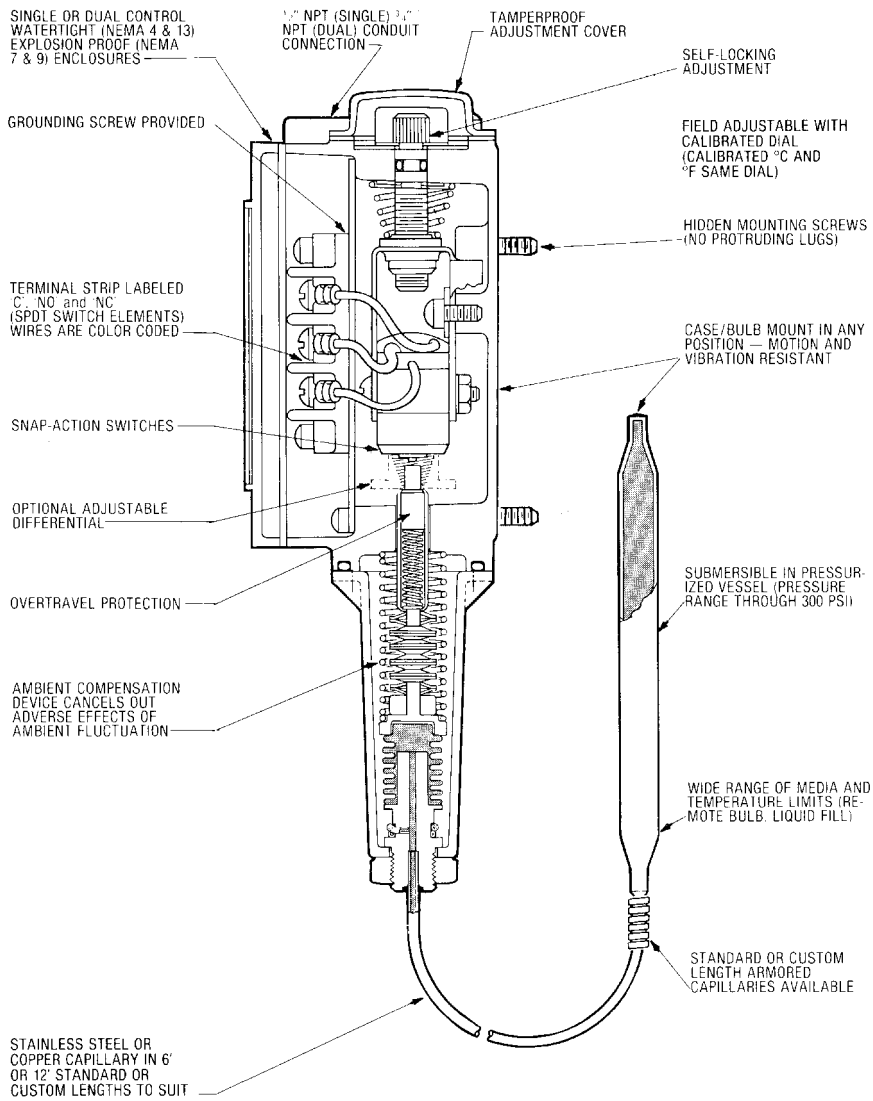


FIG. C-402 Basic components of the bulb-and-capillary type of sensor are the bellows, capillary, and bulb shown. (Source: Demag Delaval.)

the fill liquid, the coefficient of thermal expansion changes with temperature so that these two variables are compensating and the linear characteristics are maintained.

The effect of ambient-temperature changes on the fill-fluid volume in the capillary and bellows is a set-point decrease on an ambient-temperature increase and the reverse on a temperature decrease. Compensating bimetallic washers are used to offset this effect (see Fig. C-402). The total fluid change for the temperature range is absorbed by the bellows. A plunger transmits the bellows movement to the snap-action switch. The plunger spring provides overtravel to prevent damage to the switch.

Liquid-filled sensors: local-mount type. In the local-mount, outside-fill type of sensor shown in Fig. C-403 the filling surrounds the bellows. Decreases in temperature cause the fluid volume to be reduced. This causes the bellows to move toward the

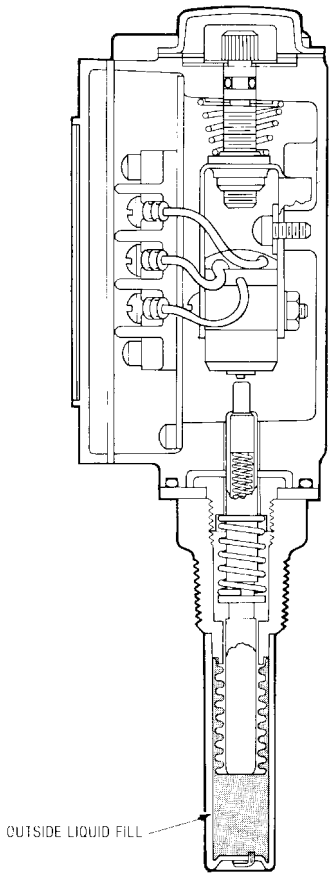


FIG. C-403 Local-mount outside-fill type. (Source: Demag Delaval.)

fill-tube end of the housing. Temperature increase causes the fluid to expand, and the bellows moves in the opposite direction. Thus, bellows action is reversed in respect to temperature change from the action of the bulb-and-capillary type.

The local-mount, inside-fill type, shown in Fig. C-404, is a lower-cost, less-responsive sensor. Construction includes an integral capsule (a bellows in a fitting) in which the inside of the bellows contains the fill fluid, rather than the fluid surrounding the bellows as in the outside-fill type.

The airspace between the shell and the bellows presents high resistance to heat transfer, resulting in a longer response time.

Local-mount switches are not affected by ambient-temperature changes because all the fill fluid is exposed to the process-fluid temperature. For this reason, no compensating washers are required (see Fig. C-405).

Response time. *Response time* is the time required for the sensor to react to a change in the process-fluid temperature. It is a function of the heat transfer between the process fluid and the sensing fluid.

A standard method of expressing response time is the step-change curve (Fig. C-406). Thirteen seconds is an excellent response time for filled systems. In general, rapid changes in temperature do not occur in process fluids.

Proof temperatures. The normal proof temperatures required generally are from 25 to 50°F above the maximum set point. Therefore, high proof temperatures are

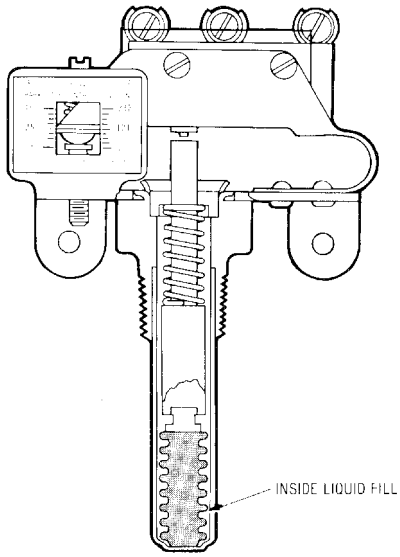


FIG. C-404 Local-mount inside-fill type. (Source: Demag Delaval.)

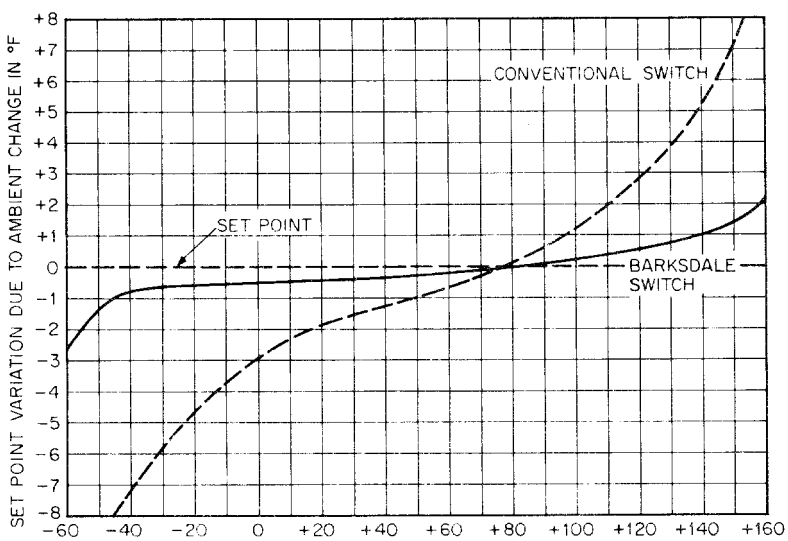
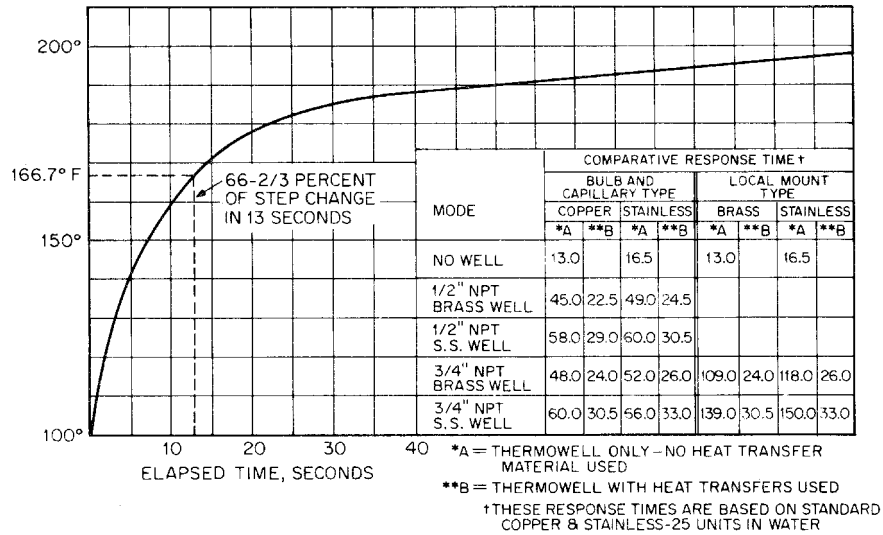


FIG. C-405 Ambient-temperature compensation. Ambient-temperature range, −65 to +165 °F for all models except −H600 series (0 to 165 °F). Curves reflect averages of a series of tests. Barksdale switches are designed for ±1/2 percent accurate response through the mid-60 percent of the adjustable range (±2 percent at minimum or maximum adjustable set point). (Source: Demag Delaval.)

usually established at 50°F above the maximum set point. It is also possible, owing to unforeseen conditions, that the actual temperature reached is higher than the proof temperature. A safety factor must therefore be available in the proof-temperature ratings.

The low proof temperature is determined by the lowest temperature that the filling fluid can tolerate with safety.

Thermowells. Both bulb-and-capillary and local-mount types of sensors may use thermowells in their installation. These are employed to allow removal of



MODE	COMPARATIVE RESPONSE TIME†							
	BULB AND CAPILLARY TYPE				LOCAL MOUNT TYPE			
	COPPER		STAINLESS		BRASS		STAINLESS	
	*A	**B	*A	**B	*A	**B	*A	**B
NO WELL	13.0		16.5		13.0		16.5	
1/2" NPT BRASS WELL	45.0	22.5	49.0	24.5				
1/2" NPT S.S. WELL	58.0	29.0	60.0	30.5				
3/4" NPT BRASS WELL	48.0	24.0	52.0	26.0	109.0	24.0	118.0	26.0
3/4" NPT S.S. WELL	60.0	30.5	66.0	33.0	139.0	30.5	150.0	33.0

FIG. C-406 Response time: 100°F step-change curve (copper bulb in water). (Source: Demag Delaval.)

TABLE C-32 Thermowell Pressure-Temperature and Velocity Limitations

Maximum fluid velocity, ft/s

	Insertion Length—U							
Material	$2^1/_2$	$4^1/_2$	$7^1/_2$	$10^1/_2$	$13^1/_2$	$16^1/_2$	$19^1/_2$	$22^1/_2$
Brass	321 (150)	129 (83.5)	46.8	23.6	14.5	9.6	6.9	5.1
Carbon steel	410 (270)	249 (150)	90.3	45.6	27.8	18.5	13.2	9.8
AISI-304 and -316	483 (350)	272 (208)	97.3	49.7	30.4	20.3	14.5	10.7
Monel	396 (300)	214 (167)	77.5	39.2	23.8	16.0	10.3	7.7

NOTE: The values in parentheses represent safe values for water flow. Unbracketed values are for steam, air, gas, and similar low-density fluids.

Pressure-temperature rating, lb/in^2

	Temperature, °F				
Material	70°	200°	400°	600°	800°
Brass	5000	4200	1000	....*	....*
Carbon steel	5200	5000	4800	4600	3500
AISI-304	7000	6200	5600	5400	5200
AISI-316	7000	7000	6400	6200	6100
Monel	6500	6000	5400	5300	5200

* Stainless steel recommended.

the temperature switch without losing process fluid, to provide extra corrosion protection, to protect the temperature sensor from system pressures, and to protect the sensor from high velocity, wear, and abrasion. Table C-32 shows velocity and pressure-temperature limitations for thermowells.

Pressure switches

A pressure switch is an electric switch designed to be actuated by a device that senses a change in pressure. All pressure switches sense the difference between two pressures, the reference pressure and the variable pressure. In most cases the reference pressure is ambient air, and the switch is then known as a *gauge-type instrument*. Some applications require a reference pressure other than ambient air. An instrument of that type is known as a *differential-pressure-actuated switch.*

In most industrial applications variations in barometric pressure are not significant. But some applications, such as aircraft with its extreme changes in altitude, require a constant reference pressure; so a reference pressure is sealed into the instrument. In an *absolute-pressure switch*, the sealed space is evacuated as closely as possible to a perfect vacuum.

When selecting a pressure switch, an analysis of the application should include expected service life, frequency and magnitude of pressure cycles, maximum pressure, the fluid to be controlled, and the desired electrical function. Consideration should be given to three basic types of pressure sensors: metal-diaphragm sensors, Bourdon-tube sensors, and piston-type sensors.

Metal-diaphragm sensors. A metal-diaphragm sensor has a diaphragm assembled into a pressure-containing capsule. The diaphragm acts as a spring that moves linearly with differential pressure applied. The design of the convolutions in the diaphragm and the thickness of the diaphragm determine the spring rate and the differential-pressure limits. Use of a base plate in the capsule designed with the same contour as the diaphragm supports and protects the diaphragm at pressures higher than its effective range. The maximum permissible pressure is called *proof pressure*.

Two materials commonly used for the diaphragm are beryllium copper and 17-7PH stainless steel. The material must have low fatigue characteristics, have a low hysteresis effect, and resist deterioration by the process fluid. (See Figs. C-407 and C-408.)

Bourdon-tube sensors. The Bourdon-tube type of sensor (see Fig. C-409) is a hollow metal tube with an elliptical cross section bent in a C-shaped arc. One end is fixed and open to pressure, and the other end is closed and free to move. When pressure is applied internally, the elliptical cross section deflects toward a more circular form. This causes the metal in the outer radius to be in tension and the material in the inner radius to be in compression. The resultant forces produce a tendency for the tube to straighten, and the free end of the tube moves in an arc. The tube acts as a spring to oppose the pressure. The amount of movement of the free end is proportional to the magnitude of the pressure.

Bourdon tubes are very accurate sensors and are capable of reproducing identical free-end movements with repeated pressure cycles. Because of their long unsupported length, however, they are sensitive to external forces on the free end. The Bourdon tube must eventually operate a switch element. It is, therefore, very important that the external forces exerted on the free end of the tube by the switch element and any related operating mechanism be constant.

Precision snap-acting switches feature constant internal spring forces, which contribute to accuracy when applied to a Bourdon tube. The direct-acting design eliminates the chance of developing unpredictable and changing frictional factors which affect accuracy.

The amount of movement of the free end is a function of the following:

1. The ratio of the major to the minor cross-sectional axis. The higher this ratio, the greater the movement.

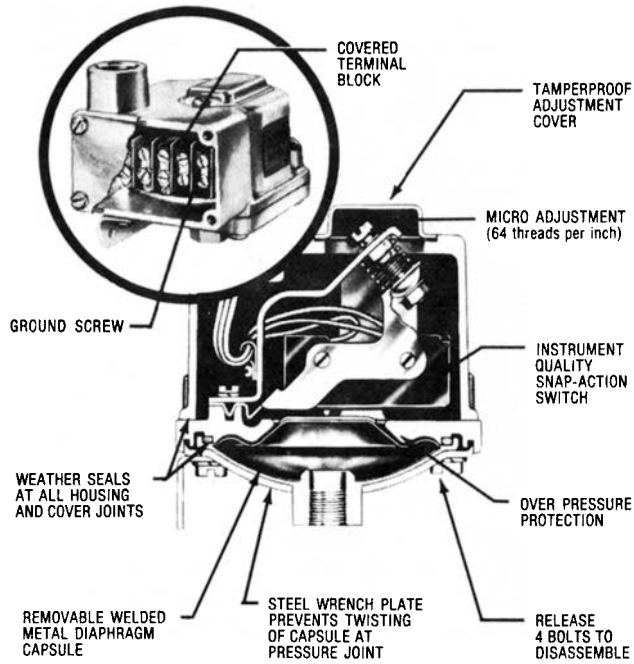


FIG. C-407 Diaphragm-model pressure-and-vacuum switch: a weld-sealed metal diaphragm direct-acting on a snap-action switch. (Source: Demag Delaval.)



FIG. C-408 Barksdale housed diaphragm pressure switch. (Source: Demag Delaval.)

2. The tube length from the fixed to the free end. A longer tube will move to a greater extent.
3. The radius of curvature. The smaller the radius, the greater the movement.
4. The total angle of curvature. The greater the angle, the greater the movement.
5. The wall thickness of the tube. The thinner the wall, the greater the movement.
6. The modulus of elasticity of the tube material. The lower the modulus, the greater the movement.

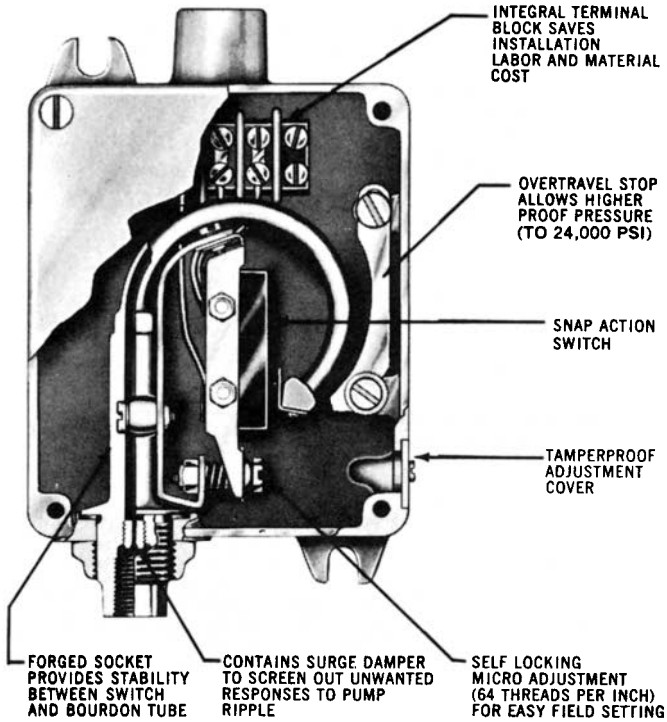


FIG. C-409 Bourdon-tube-model pressure switch: a weld-sealed Bourdon tube direct-acting on a snap-action switch. (Source: Demag Delaval.)

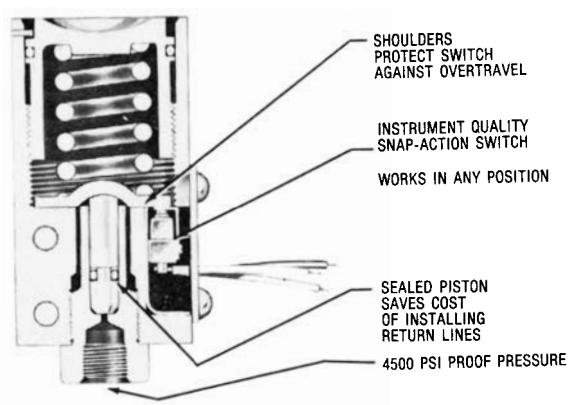


FIG. C-410 Sealed-piston pressure switch: an O-ring sealed piston acting on a snap-action switch. (Source: Demag Delaval.)

Tubes are generally formed from steel or phosphor bronze seamless tubing.

Some Bourdon-tube pressure switches incorporate a tube stop to prevent movement of the free end of the tube beyond the amount required to achieve the maximum set point of the pressure switch. This increases the proof pressure.

Piston-type pressure switches. A piston-type sensor (see Figs. C-410 and C-411) combines a rod and a piston to produce a relatively low-cost pressure switch with long life at high pressures and cycling rates. Piston switches are generally used in

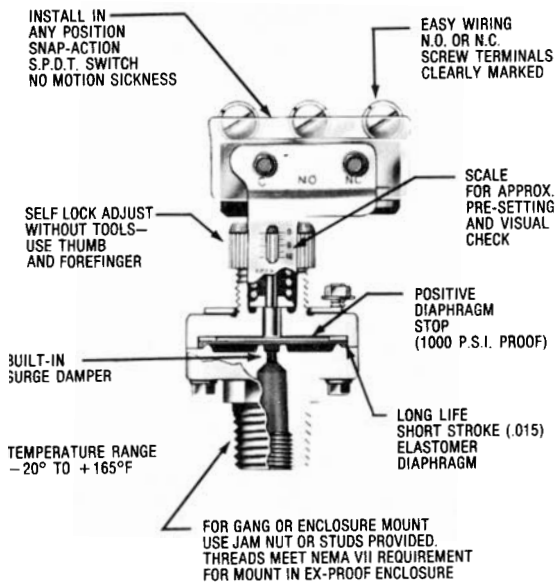


FIG. C-411 Diaphragm sealed pressure or vacuum switch: a diaphragm sealed piston direct-acting on a snap-action switch. (Source: Demag Delaval.)

rigorous service applications because of high cycling rates or system pressure surges.

The sensor is composed of a piston with an O ring and Teflon backup ring, rod, spring, and body. The amount of pressure required to produce movement is a function of the piston area, spring rate, and the extent to which the adjusting screw has compressed the spring.

Piston-type switches covering adjustable ranges from 15 to 12,000 lb/in^2 and proof pressures from 3000 to 20,000 lb/in^2 are commercially available.

Other sensor types. Other types of sensors with more limited applications include the brass bellows type used in conjunction with a loading spring inside the bellows. Pressure is introduced on the outside of the bellows, which is enclosed within a pressure-containing chamber. A plunger transmits movement of the bellows to the switch element.

But there are major weaknesses in bellows-type sensors, including relatively short life, limited proof pressure in relation to the maximum set point, sensitivity to vibration and shock, less resistance to surge pressure, and a requirement of the plunger to operate the switch element. They are generally offered for pressures up to 1500 lb/in^2 and are most often used when pressures do not exceed 500 lb/in^2.

Helical and spiral Bourdon tubes are two other types in limited use. The helical tube is similar in deflection behavior to a C-shaped tube. Since it is coiled into a multiturn helix with its total angle between 1800 and 3600°, its tip travel is proportionately greater than that of a tube with less than 360° curvature.

The spiral tube also amplifies tip travel because of its multiturn configuration (usually four to eight turns). The performance of both tubes is very similar, and choice of either tube is dictated by the limitation on the overall sensing-element diameter and length. A partly helical, partly spiral tube design has also been developed. Such tubes would be more sensitive to vibration than a C-shaped tube. Neither design is widely used in pressure switches.

Another type is the negative-spring-rate sensor called a *Belleville washer* (see Fig. C-412). When pressure is applied to the sensor, the pressure plate acts against

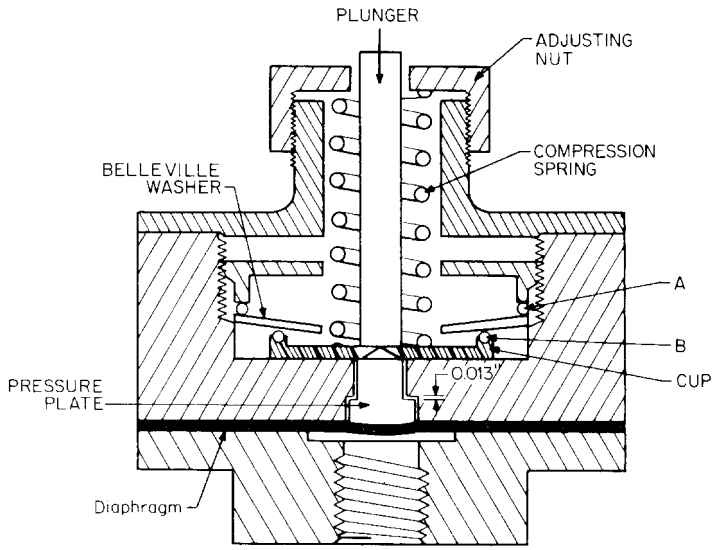


FIG. C-412 Negative-spring-rate sensor. (Source: Demag Delaval.)

the cup, which exerts force on the washer. The washer snaps upward, and the snap-over deflection operates the snap-action switch. A major advantage is that the washer makes the pressure switch less sensitive to vibration.

Another type, the unsealed-piston design, offers the advantage of not requiring an O ring. But one disadvantage is the requirement for a drain line. Also, dirt in the system can enter the grooves and destroy their ability to provide lubrication and pressure balance around the piston. When this occurs, rapid wear and high friction result.

Accuracy and life. The ability of a pressure switch to operate repetitively at its set point is its accuracy, or repeatability. It can be expressed as a percentage of either the adjustable range or the set point. The factors most strongly influencing life are:

1. Number of cycles to which the sensor and switch element respond
2. Range of pressure fluctuations through which the sensor flexes
3. The pressure surges that exist in the system
4. The electrical load on the switch element
5. Corrosion resistance to the fluid in the sensor and the environment

Anticipated service life is generally the first consideration in selecting a pressure switch regardless of the pressure or sensitivity desired. If the service life (number of cycles during which the switch is expected to operate without replacement) is 1 million or less, a Bourdon tube or a diaphragm switch is indicated. If more than 1 million cycles is desired, a piston switch should be used. An exception is made when the pressure change in a system is quite slight (20 percent or less of the adjustable range). Under such conditions, a Bourdon tube or a diaphragm switch can be used for up to 2.5 million cycles before metal fatigue.

Application and cost. There are two basic types of applications:

1. Applications that require a high degree of accuracy and in which pressure surges and cycling rates normally are relatively low. These are instrument-type applications.

2. Applications that require a moderate or low degree of accuracy and for which cycling rates and pressure surges can be relatively high. These are control-type applications.

The cost-accuracy relationship generally merits attention. Within limits, the higher the accuracy requirements, the higher the cost. In addition, the cost is generally proportional to the pressure required. Type 1 applications are generally served best by metal-diaphragm and Bourdon-tube sensors and Type 2 applications by lower-cost sealed-piston sensors.

Liquid-level and liquid-flow instrumentation

Manufacturing and process machinery today utilizes many types of liquid media to ensure correct operation and function. Liquid fuels are used as a source of energy, viscous oils lubricate moving assemblies, and water is used to remove heat from many types of electronic and process equipment. These liquids and others can be considered integral to each machine, as any subassembly or electronic circuit would be. The control and monitoring of these liquids is, therefore, a very important factor in basic design.

Many techniques that employ basic engineering principles to accomplish this control and monitoring have been developed. Liquid-level and liquid-flow instrumentation plays a major role in satisfying the requirements of indication, control, and/or protection of the medium.

Level sensing. Some of the different types of level medium-sensing techniques used today follow. This is by no means a complete list, and many slight modifications of the principles discussed also are used.

Magnetic-float principle. The liquid level is monitored by means of a magnetically equipped float that traverses an in-tank stem or probe. As the level in the tank changes, the float moves correspondingly, rising and falling along the stem. Inside the stem, a reed switch is positioned so that as the float moves by it, it actuates, signaling that the level is at that particular point.

Capacitance method. The liquid level is measured by monitoring a change in capacitive reactance between two points. A conductor probe inserted in the tank and the tank wall itself serve as two conductor plates of a capacitor. The medium between the probe and the tank wall act as the dielectric. The capacitance between the probe and the tank wall will vary as a result of liquid-level change. This varying capacitance can be related to a varying level output.

Conductance method. The liquid level is monitored by using the conduction of current through the liquid. Two conductor probes are placed within the tank. As the liquid level changes and the probes become immersed in the liquid, a voltage potential across the two probes causes current to flow from one probe through the liquid to the other probe and then out to the receiver or other sensing equipment. As current starts to conduct, it is known that the liquid level is at that point within the tank.

Ultrasonic method. The liquid level is measured through the use of ultrasonic-frequency-wave generation. This system consists of a transmitter-receiver inside the tank and a remote indicator outside. The transmitter-receiver is mounted at the tank top and generates a high-frequency wave that is directed at the medium level in the tank. Thus, the liquid level is monitored by tracking the time required for the ultrasonic-frequency wave to make contact with the liquid level and return to the receiver sensing element in the head assembly. That signal is then transferred electrically to an indicator module.



FIG. C-413 Flow switch. (Source: Demag Delaval.)

Electromechanical (displacement) method. The liquid level is measured using a weighted displacer float, which is connected to a cable length and placed inside a liquid. When the float is immersed in the liquid, it will displace a certain volume. This displaced volume exerts a force change on the connecting-cable arrangement above the displacer. That force change is detected through a sensor such as a strain gauge and converted to an indication of liquid-level change.

Differential-pressure method. The liquid level is measured by the indication of a pressure change between two points within the tank. A pressure-sensitive sensor such as a diaphragm, bellows, or strain gauge is connected to a high and a low port within the tank configuration. As liquid level changes, head pressure also changes in the tank. This pressure change can be proportional to a level change and indicated as such. For pressurized tanks, a reference pressure is established and connected to the sensor. This enables a change in pressure to be compared with the reference to indicate the level.

Flow-measurement sensing. Although not encompassing the entire range of flow-sensing techniques presently available, the techniques listed in the following paragraphs provide a guide that states the basic principles. Modifications as well as combinations of principles are used in flow measurement.

Pressure-head flow measurement. This method of monitoring flow utilizes a change in pressure drop to a corresponding change in flow. A restriction positioned inside a flow line creates a pressure drop as the flow passes across it. The volumetric flow rate past the restriction is proportional to the square root of the pressure drop. Some current devices that use the pressure-differential principle are the venturi, the flow nozzle, and the orifice plate.

In some cases, measure of a predetermined minimum or maximum flow rate is desired. One system uses the pressure drop across a restriction to cause a shuttle to move a magnet against a spring to actuate a reed switch. The reed switch closes an electric circuit to actuate a control or an alarm. Different flow-rate indications can be obtained by varying either the shuttle size or the spring rate. (See Fig. C-413.)

Positive-displacement flow measurement. This method uses the principle of volumetric displacement to determine the flow rate. It consists of a measuring chamber and a displacer rotating within that chamber. As liquid flow passes through the chamber, it causes the displacer to rotate. Each rotation displaces a given volume of liquid, which is monitored. By simply counting the revolutions, the amount of liquid displaced in a given time frame, i.e., the flow rate, can be established. Examples of this principle are the rotating vane, the sealed drum, and lobed-impeller meters.

Variable-area flow measurement. This method employs the principle of a variable-area orifice with a constant pressure differential. The construction is a vertical transparent tube with a tapered bore and the largest diameter at the top. The flow is always vertically upward through the tube. Inside the tapered bore is a float that moves vertically to an area corresponding to the flow rate. Since the weight of the float is constant, the only variable is the flow area. The outside of the tube is calibrated to show flow rate as a function of the position of the float.

Ultrasonic flow measurement. There are many modifications and versions of this method. The basic principle is that a frequency wave is established and directed at the medium. Therefore, a change in medium flow will cause a change in the established frequency. This change can be related to a change in flow rate.

The Doppler shift is used in one method. Frequency waves are transmitted in the same direction as the flow of the medium. As the fluid rate changes, the wave frequency will actually shift in output. This shift is proportional to the change in flow. By measuring this new frequency shift, the flow rate is monitored.

Vortex flow measurement. This method utilizes the principle of vortex shedding to monitor flow rates. The frequency of the vortices formed is proportional to flow rate. An element, usually of triangular shape, is placed perpendicularly to the flow. Fluid vortices are formed around the element and eventually are shed downstream. These vortices move downstream with a certain oscillatory frequency. This frequency will change as the flow is decreased or increased. Heated thermistors are placed in the path of the vortices; their cooling rate increases as a vortex passes. The frequency of the cooling cycles is an index of the flow rate.

Turbine flow measurement. This measurement principle states that a turbine wheel in a flow stream revolves at a speed proportional to the fluid-flow rate. A small turbine wheel is allowed to rotate on an axis inside the unit body. As the flow rate changes, the speed of rotation of the turbine will also change. To detect this change, a small magnet is located on a point on the wheel, thus generating a pulse for each revolution. This pulse is sensed and recorded by a pickup coil circuit on the outside of the body. The number of pulses detected can be translated into the amount of flow passing through the turbine wheel.

Reed switches. Reed switches are used in many applications as well as in level- and flow-control devices. The reason for their extensive use is their reliability and long life. These qualities are due to the hermetically sealed reed blades and the overall simplicity of the design.

Construction. A reed switch is constructed of very simple material components. Two reed blades made of a soft, thoroughly magnetic material are situated in a gas-purged glass envelope that is hermetically sealed (see Fig. C-414). The tips of the blades are plated with either gold or silver to ensure long life as well as low contact resistance. Rhodium and tungsten are also used for plating.

Contact arrangements of the reed switch are available in single-pole–single-throw (SPST) and single-pole–double-throw (SPDT) types. Reed switches are normally identified by contact configuration (see Figs. C-415 and C-416) as well as by resistive-power ratings.

For example, “SPST, 10 W” refers to a single-pole–single-throw reed switch with a 10-W resistive-power-rating factor.

Operation. When the reed switch is in proximity to a magnetic field, the blade tips will become magnetically opposed (north-south poles). At that point, the reed blades will snap together and remain in contact until the magnetic field has been removed or sufficiently decreased. The physical gap between the two reeds acts as magnetic

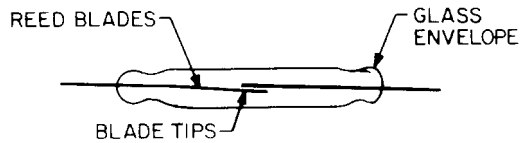


FIG. C-414 Reed switch. (Source: Demag Delaval.)

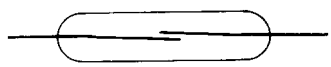


FIG. C-415 Single-pole–single-throw (SPST) reed switch. (Source: Demag Delaval.)

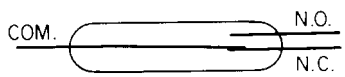


FIG. C-416 Single-pole–double-throw (SPDT) reed switch. (Source: Demag Delaval.)

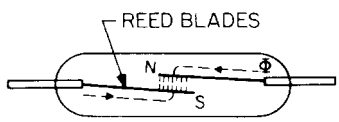


FIG. C-417 Magnetic activation of reed switch. (Source: Demag Delaval.)

resistance to the magnetic flux generated from the magnet. This gap resistance decreases as the reed blades come closer together. The magnetic force produced by permanent magnets or electromagnets is inversely proportional to the square of this distance gap. Therefore, the reed-switch-blade closure will accelerate as the tips approach each other. The larger the magnetic field, the faster the blades snap together. (See Fig. C-417.)

Control Valves

A number of process valves are simple hand-turned valves. They include:

Globe valves: Fluid flow through this valve changes direction. Fewer turns are required to move this valve than with a gate valve. It is useful for throttling service. If extremely close regulation is required, a *needle globe valve* should be used.

Ball valves require a 90° turn to shut off flow completely. They are much lighter for a given size than either a globe or a gate valve. Maintenance is simple; however, this valve type is not suitable for throttling.

Plug valves can be either lubricated or nonlubricated. They are like ball valves, except instead of the ball there is a plug, often shaped like a truncated cone. These valves do not seize or gall as might be the case with some gate valves.

Diaphragm valves have a flexible diaphragm that closes the pipe against the flow of the liquid. Isolation of the working parts from the fluid stream prevents product contamination and corrosion. Maintenance requires the occasional diaphragm change.

Pinch valves are more for laboratory-type application as they stop flow through small-diameter rubber tubing.

Some valves operate either manually or automatically.

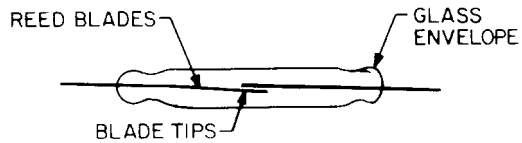


FIG. C-414 Reed switch. (Source: Demag Delaval.)

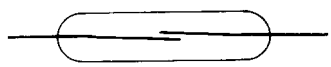


FIG. C-415 Single-pole–single-throw (SPST) reed switch. (Source: Demag Delaval.)

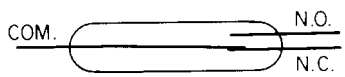


FIG. C-416 Single-pole–double-throw (SPDT) reed switch. (Source: Demag Delaval.)

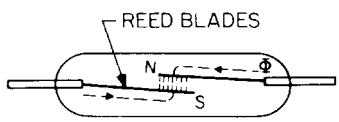


FIG. C-417 Magnetic activation of reed switch. (Source: Demag Delaval.)

resistance to the magnetic flux generated from the magnet. This gap resistance decreases as the reed blades come closer together. The magnetic force produced by permanent magnets or electromagnets is inversely proportional to the square of this distance gap. Therefore, the reed-switch-blade closure will accelerate as the tips approach each other. The larger the magnetic field, the faster the blades snap together. (See Fig. C-417.)

Control Valves

A number of process valves are simple hand-turned valves. They include:

Globe valves: Fluid flow through this valve changes direction. Fewer turns are required to move this valve than with a gate valve. It is useful for throttling service. If extremely close regulation is required, a *needle globe valve* should be used.

Ball valves require a 90° turn to shut off flow completely. They are much lighter for a given size than either a globe or a gate valve. Maintenance is simple; however, this valve type is not suitable for throttling.

Plug valves can be either lubricated or nonlubricated. They are like ball valves, except instead of the ball there is a plug, often shaped like a truncated cone. These valves do not seize or gall as might be the case with some gate valves.

Diaphragm valves have a flexible diaphragm that closes the pipe against the flow of the liquid. Isolation of the working parts from the fluid stream prevents product contamination and corrosion. Maintenance requires the occasional diaphragm change.

Pinch valves are more for laboratory-type application as they stop flow through small-diameter rubber tubing.

Some valves operate either manually or automatically.

Butterfly valves operate with the movement of a wing-like disk that works at right angles to the fluid flow. This valve type can be operated manually or using pneumatic, electrical, hydraulic, or electronic actuation.

Nonreturn or *check valves* prevent the reversal of flow in piping. In a *swing check* type the hinged disk is held open with the flow of liquid. When flow stops, gravity causes the disk to fall into closed position. With *lift check*–type valves, the closure disk is raised by the fluid flow. When flow stops, the disk falls back into closed position.

Current-to-Pressure Converters for Precise Steam and Fuel Valve Control*

The source for the information in this subsection is Voith Turcon who designate their current-to-pressure converters "I/P" ("I" for current and "P" for pressure).

I/P converters offer control of steam and fuel valve actuators. Although designed for turbine applications, these converters can also be effective in other process control situations.

This converter quickly and precisely changes a current input signal into a proportional fluid output pressure to regulate steam or fuel flow.

I/P converters are built to a solid, compact design. All of the control electronics are safely housed within the unit for reliable functioning—even in harsh environments. With just three moving parts, this I/P converter is reliable and durable ("low-wear"). (See Figs. C-418 through C-421 and Table C-33.)

Operating principles

The I/P converter reliably converts a 4–20 mA input signal into a proportional output hydraulic pressure and double-checks for supremely accurate valve positions and turbine speeds.

At the core of the I/P converter is an electromagnet. A 24-volt DC current energizes the magnet, which in turn creates a force on the actuating rod. A 4–20 mA input signal works with the unit's controller and amplifier to regulate this force. Any variation in the 4–20 mA input signal affects the pressure being exerted by the magnet onto the actuating rod. The force applied to the actuating rod is used to precisely control a hydraulic piston, which opens and closes the consumer and drain ports. (See Figs. C-422 through C-425.)

The sequence of operations is as follows:

1. When the 4–20 mA signal reaches the converter, its controller and amplifier adjust the magnetic force to a pressure directly proportional to the input signal.
2. This force is measured by a semiconductor that serves as the unit's magnetic force sensor/flux detector. Magnetic force lines penetrating this element produce a proportional output voltage (the Hall effect).
3. The output voltage is looped back to the converter's controller and compared to the set value, W. If the unit senses a difference between the input signal and the feedback signal, the controller and amplifier correct the magnetic force so that the difference is zero.
4. The magnetic force adjusts the actuating rod to the appropriate position with up to 90 lb (400 N) of pressure.
5. As a result of this precise control technology, the I/P converter's output line always contains the exact pressure needed to position the steam or fuel valve.

---

*Source: J.M. Voith GmbH, Germany.



FIG. C-418 "I/P" converters. (Source: J.M. Voith GmbH.)

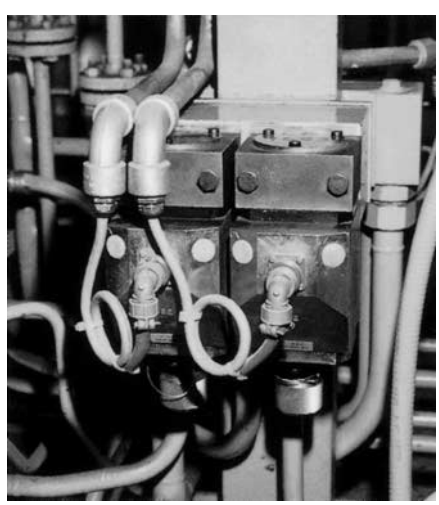


FIG. C-419 A typical installation of an I/P converter in a cogeneration plant. (Source: J.M. Voith GmbH.)



FIG. C-420 Minimum and maximum output pressures of an I/P converter can be set externally. (Source: J.M. Voith GmbH.)

TABLE C-33 Selection Table

Regulating Range	Manual Actuation		Piston Damping		I/P Converter Type		Maximum Input Pressure	Flow Rate to Consumer (Δp = 1 bar)	Flow Rate to Drain (Δp = 1 bar)
	With	Without	With	Without	Standard	EExd			
0–72.5 psi		■		■	DSG-B05102	DSG-B05202	101.5 psi	4.9 GPM	5.4 GPM
0–5 bar	■			■	DSG-B05112	DSG-B05212	7 bar	18.6 l/min	20.5 l/min
		■	■		DSG-B05103	DSG-B05203			
	■		■		DSG-B05113	DSG-B05213			
14.5–101.5 psi		■		■	DSG-B07102	DSG-B07202	101.5 psi	4.9 GPM	5.4 GPM
1–7 bar	■			■	DSG-B07112	DSG-B07212	7 bar	18.6 l/min	20.5 l/min
		■	■		DSG-B07103	DSG-B07203			
	■		■		DSG-B07113	DSG-B07213			
0–145 psi		■		■	DSG-B10102	DSG-B10202	219 psi	4.4 GPM	4.9 GPM
0–10 bar	■			■	DSG-B10112	DSG-B10212	15 bar	16.8 l/min	18.8 l/min
		■	■		DSG-B10103	DSG-B10203			
	■		■		DSG-B10113	DSG-B10213			
14.5–203 psi		■		■	DSG-B14102	DSG-B14202	219 psi	4.4 GPM	4.9 GPM
1–14 bar	■			■	DSG-B14112	DSG-B14212	15 bar	16.8 l/min	18.8 l/min
		■	■		DSG-B14103	DSG-B14203			
	■		■		DSG-B14113	DSG-B14213			
0–290 psi		■		■	DSG-B20102	DSG-B20202	655 psi	2.5 GPM	2.1 GPM
0–20 bar	■			■	DSG-B20112	DSG-B20212	45 bar	9.8 l/min	12.0 l/min
		■	■		DSG-B20103	DSG-B20203			
	■		■		DSG-B20113	DSG-B20213			
0–430 psi		■		■	DSG-B30102	DSG-B30202	430 psi	5.4 GPM	5.8 GPM
0–30 bar	■			■	DSG-B30112	DSG-B30212	30 bar	20.5 l/min	22.3 l/min
		■	■		DSG-B30103	DSG-B30203			
	■		■		DSG-B30113	DSG-B30213			

NOTES
1. Further pressure ranges available upon request.
2. Consult factory for FM-certified explosion-proof designs which meet Class I, Divisions 1 and 2, Groups B, C, and D service.
3. I/P converter weight: approximately 22 lb (10 kg) for all models.

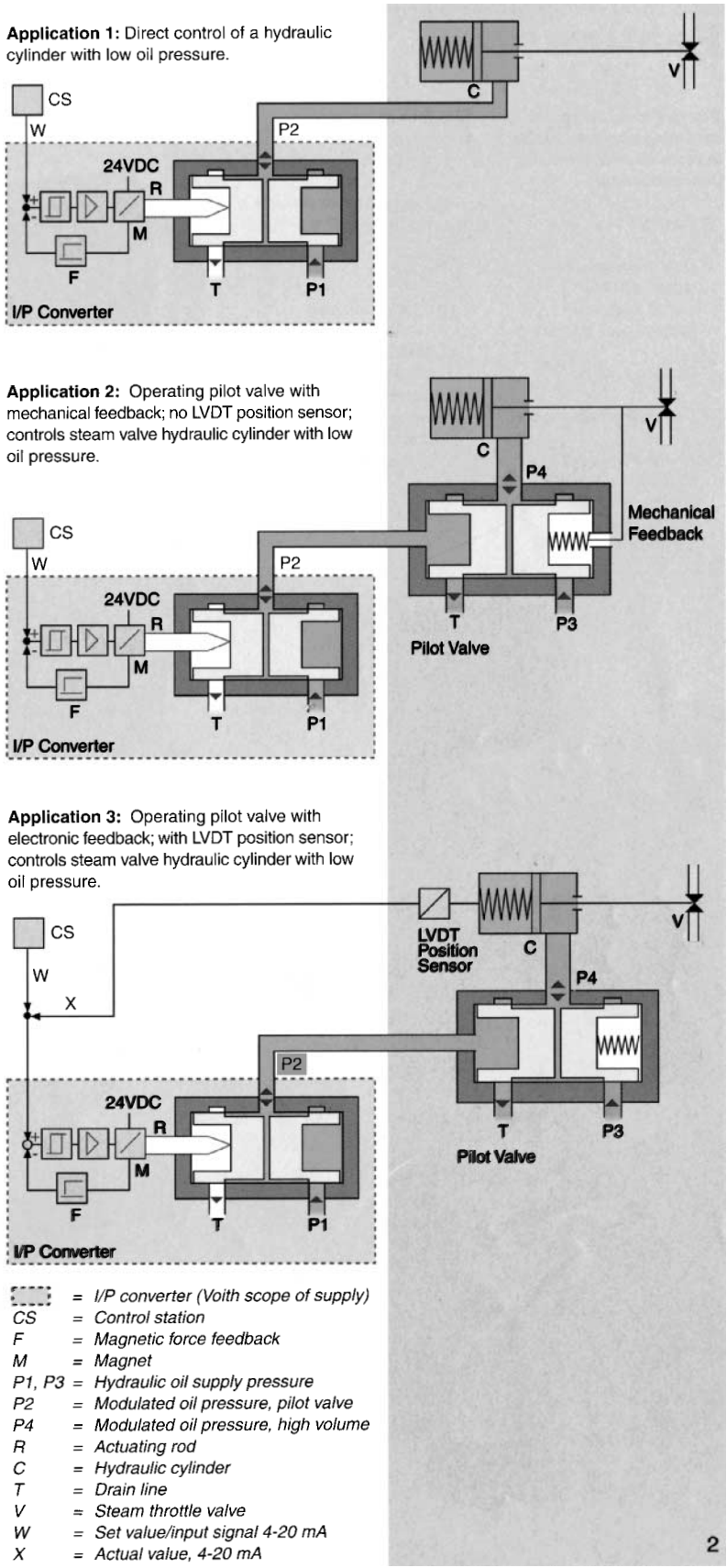


FIG. C-421 Applications of an I/P converter. (Source: J.M. Voith GmbH.)

Unit is shown in typical mounting position, which is vertical, with hydraulic connections up.

1 Control Piston

2 Dual drain lines for higher maximum output drain rates (aids quicker shutdown)

3 Actuating rod

4 Connecting cable socket

5 All electronics are contained within the housing for protection against damage, radio interference, etc.

6 Flux detector

7 Armature

8 Electromagnet consists of magnet housing and coil

9 Spring

10 Locking element

11 Manual actuation for local control during commissioning or in the event of a power loss

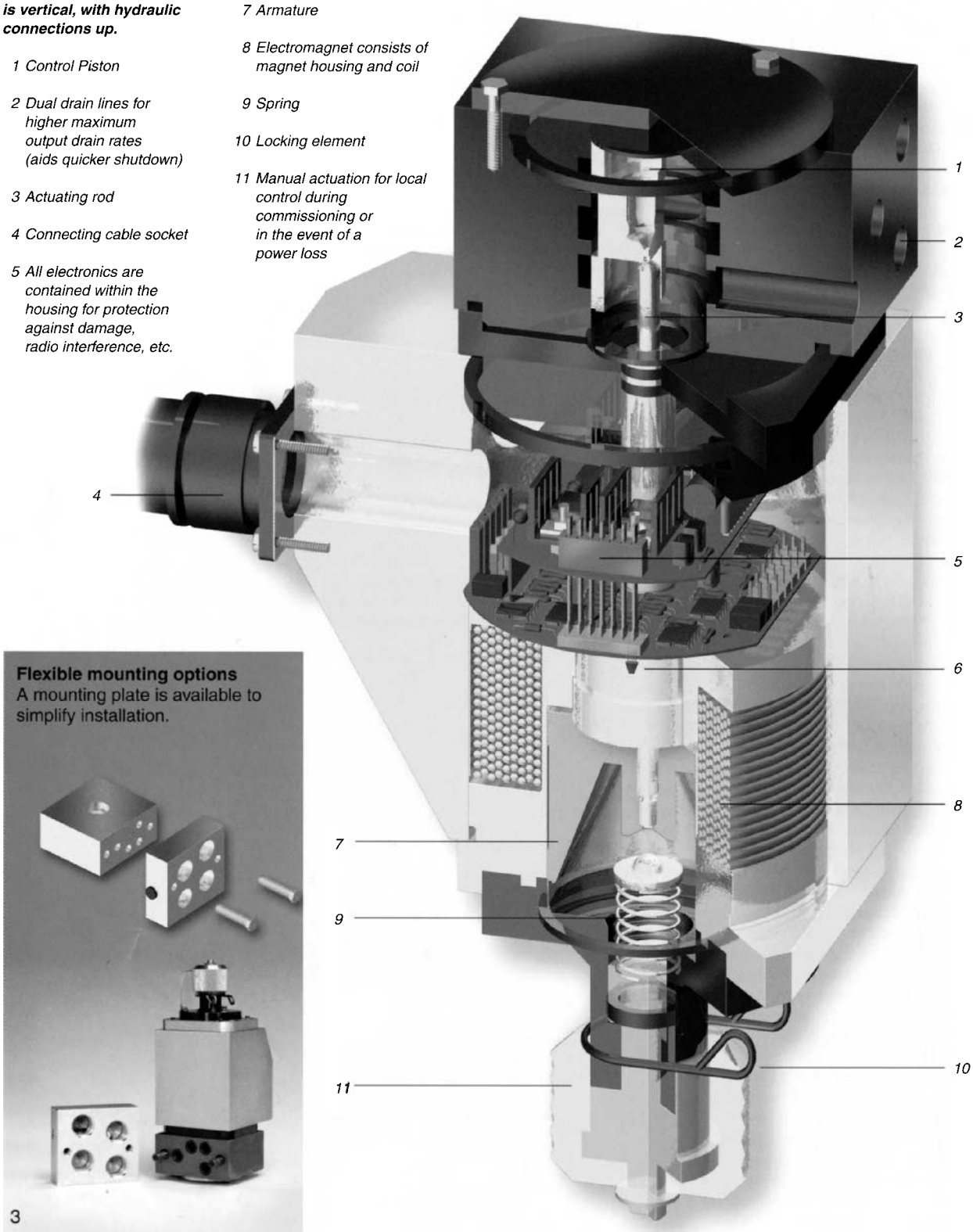


FIG. C-422 Internals of a typical I/P converter. (Source: J.M. Voith GmbH.)

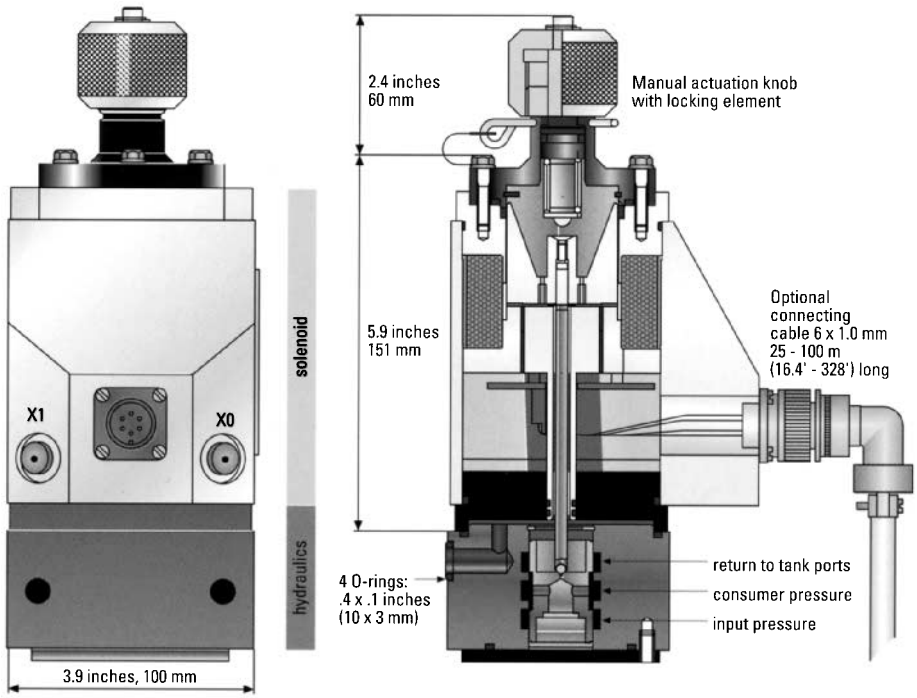


FIG. C-423 Typical dimensions of an I/P converter. (Source: J.M. Voith GmbH.)



FIG. C-424 I/P converter uses industry standard connections. (Source: J.M. Voith GmbH.)

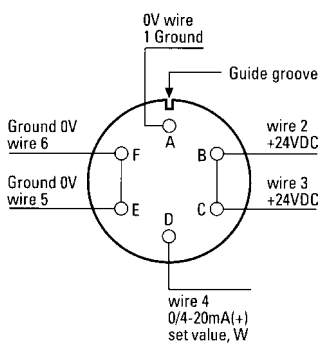


FIG. C-425 Schematic of I/P converter connections. (Source: J.M. Voith GmbH.)

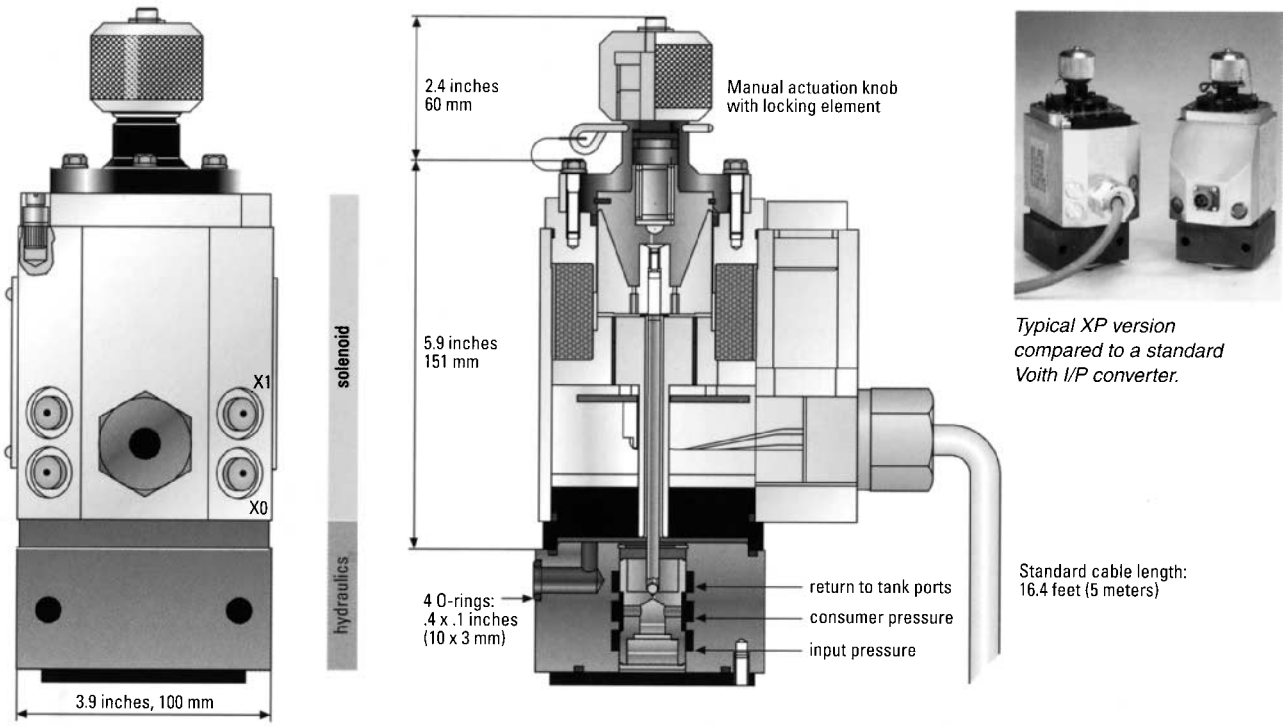


FIG. C-426 Explosion-proof design (for EExD IIC T4, PTB No Ex-90, C, 1065). (Source: J.M. Voith GmbH.)

Advantages of this basic design

- The unit's magnetic drive and the hydraulic section's pressure-reducing valve work together to function as a pressure-regulating valve.
- Dynamic and hysteresis-free
  - Resolution is better than 0.1 percent
  - Accuracy is not affected by air-gap, magnetic hysteresis, temperature, or fluctuations in supply voltage.
- Recommended oil contamination to NAS 1638 Class 7, or ISO 4406 Class 16/13.
- Short conversion time from mA input signal to proportional, stationary pressure ($t < 35$ m).
- Few electronic and mechanical parts ensure full functionality in harsh environments.
- All electronics for the I/P converter are integrated in the housing.
- Design withstands higher input pressure (pressure ranges available from 0 to 3000 psi).
- Standard and explosion-proof designs are available. (See Figs. C-426 and C-427.)
- In the version incorporating a PID controller, you can compensate for pipeline pressure losses. This optional design also allows for control of valve positions and turbine rpm.
- Uses turbine oil as hydraulic fluid with no additional filter required.

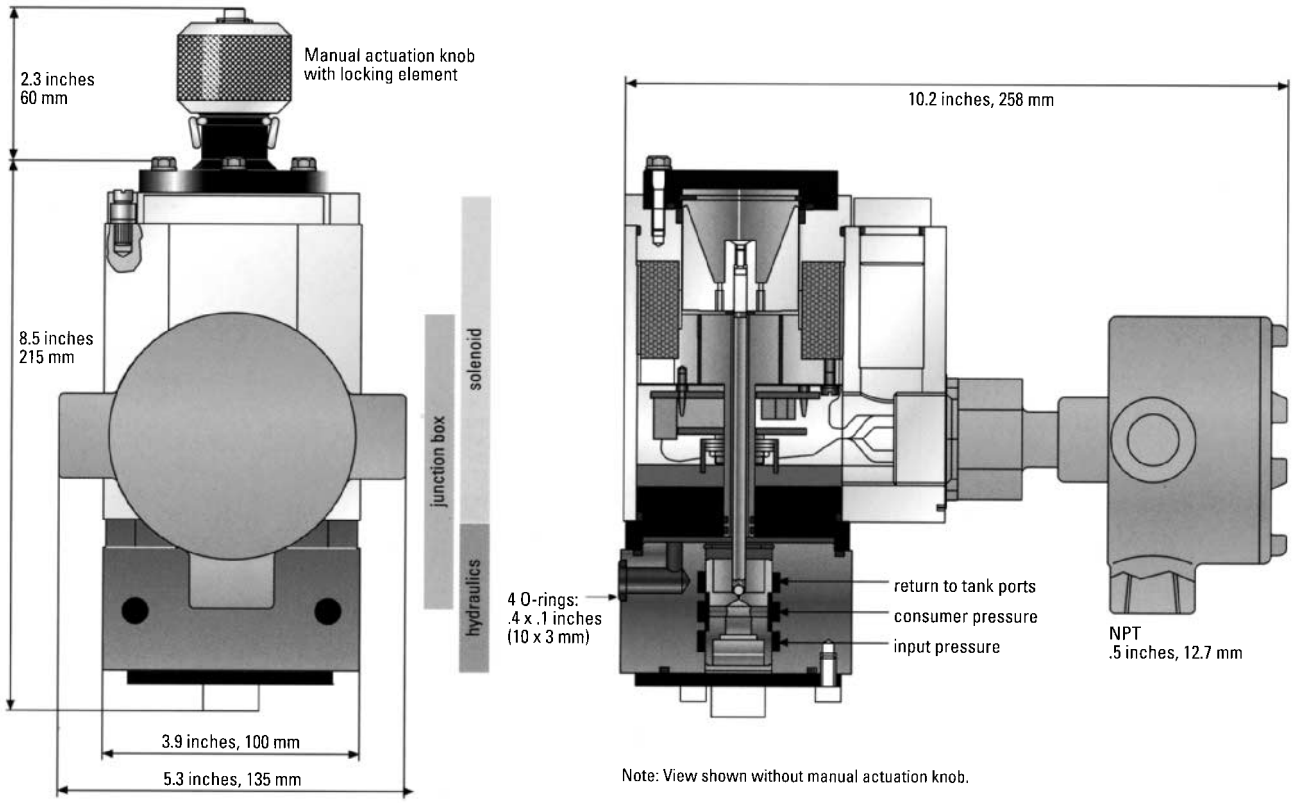


FIG. C-427 Explosion-proof design (for Class I, Divisions 1 and 2, Groups B, C, and D service). (Source: J.M. Voith GmbH.)

Speed and accuracy

- Reproducibility <0.1 percent
- Accuracy <1 percent
- No drift
- Resolution is better than 0.1 percent
- Hysteresis-free
- Does not generate static friction during operation.
- Accuracy is not affected by air-gap, magnetic hysteresis, temperature, or fluctuations in supply voltage.
- Fast response time
  - Short conversion time for proportional stationary pressure ($t < 35$ ms)
  - Rapid closedown, which includes internal oil circulation (≤0.1 s)

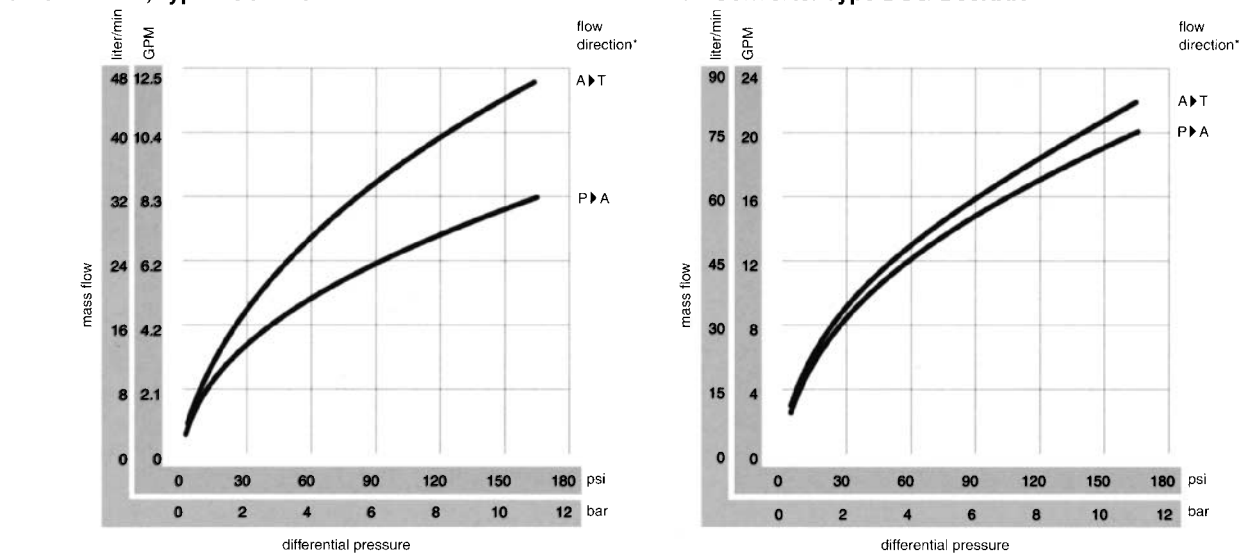
Performance and frequency response

See Figs. C-428 and C-429.

Knife-gate valve*

Depending on what flows through (process) pipe, the cause of costliest wear on one line could be heat or corrosion, while abrasiveness is the key concern in another

*Source: Adapted from extracts from "Controlling the Flow," *Mechanical Engineering*, ASME, December 1998.



*A ▶ T = Flow direction from output to drain; P ▶ A = Flow direction from oil supply to output / consumer; hydraulic oil: Teresso 46 (at 40°C)

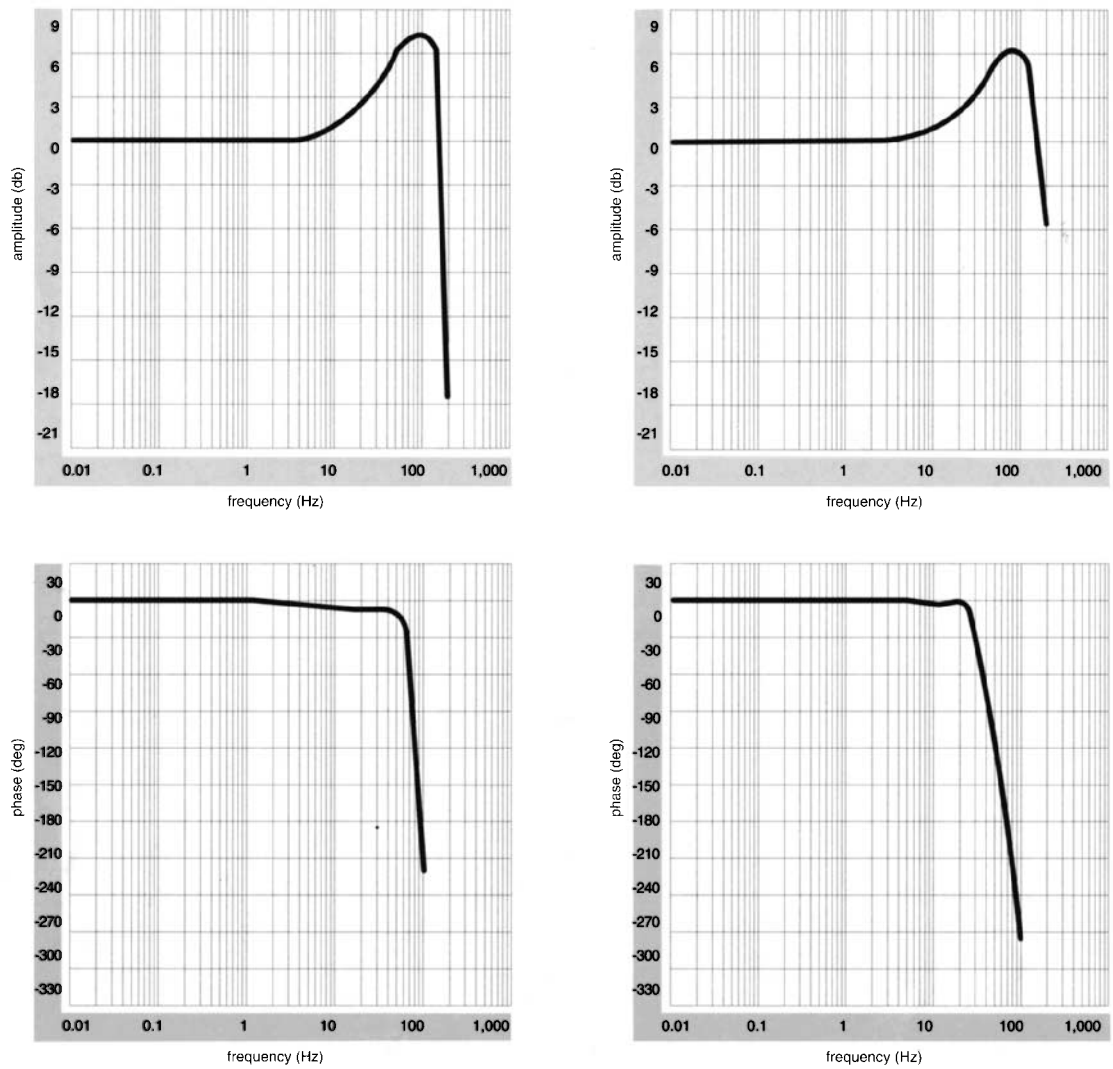
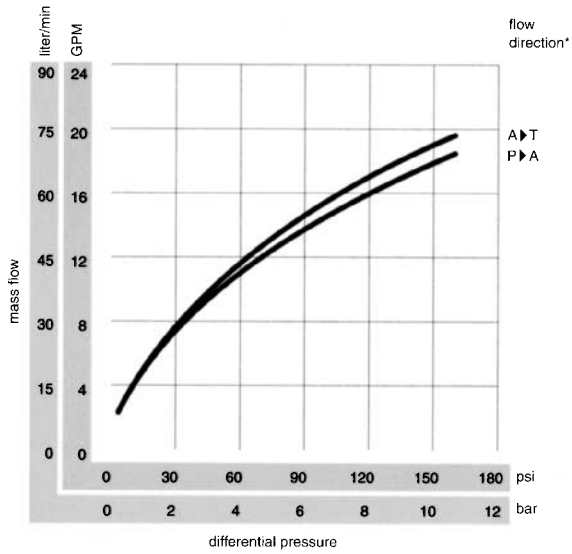


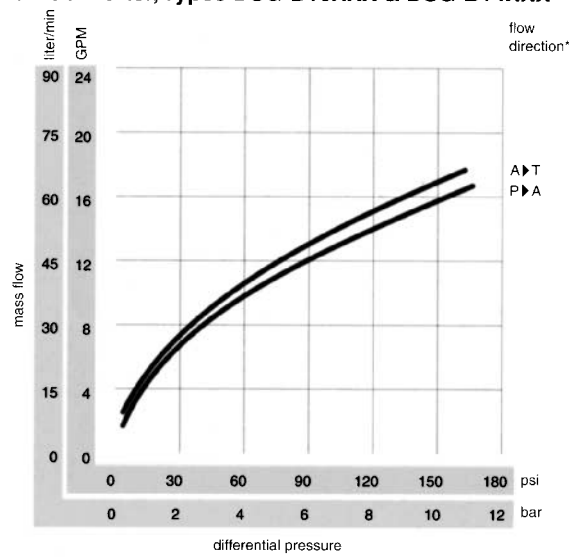
FIG. C-428 Performance and frequency response curves for different I/P converters. (Source: J.M. Voith GmbH.)

Flow Vs. Differential Pressure

I/P Converter, Types DSG-B05XXX & DSG-B07XXX

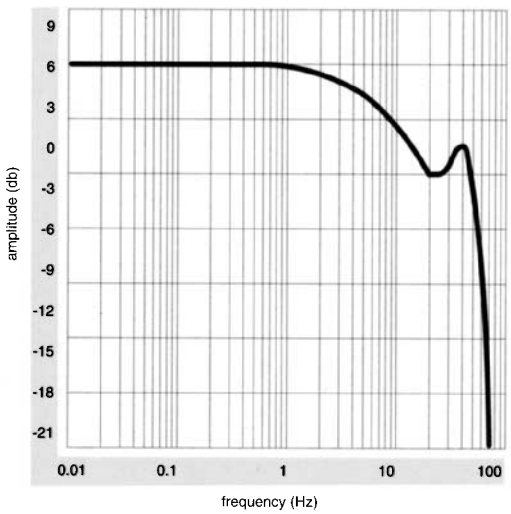


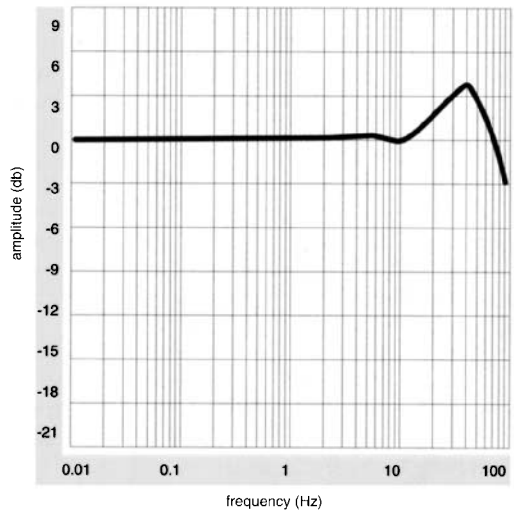
I/P Converter, Types DSG-B10XXX & DSG-B14XXX



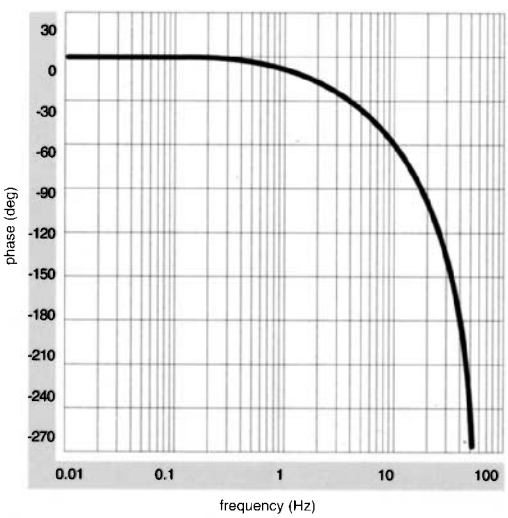
*A▶T = Flow direction from output to drain; P▶A = Flow direction from oil supply to output / consumer; hydraulic oil: Teresso 46 (at 40°C)

Amplitude





Phase



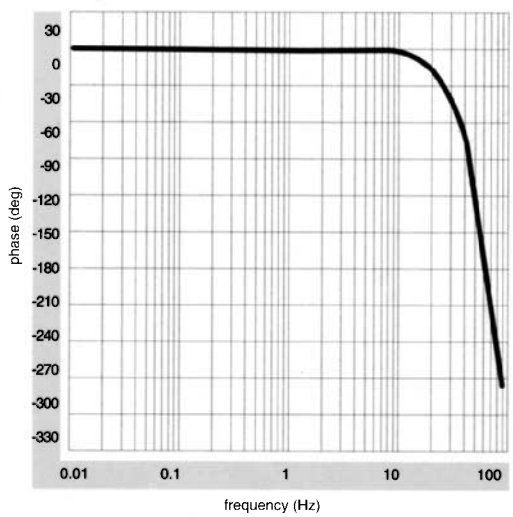


FIG. C-429 Performance and frequency response curves for different I/P converters. (Source: J.M. Voith GmbH.)

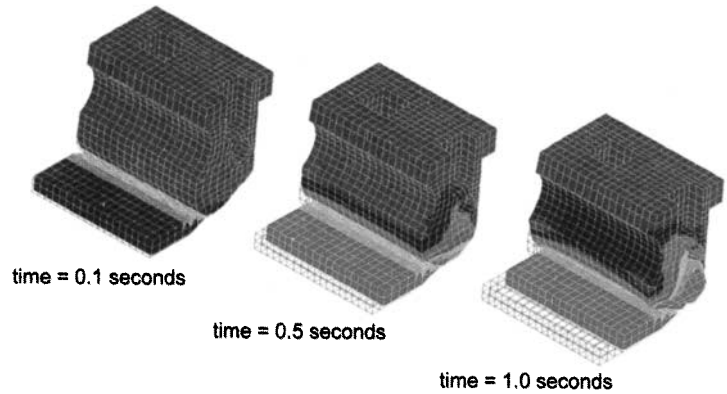


FIG. C-430 Computer simulation software was used to simulate the valve in action on a computer, reacting to the severe pressure and temperature of the slurry piping system's flowing materials. (Source: Clarkson Co.)

system. Each mining, power, or paper company has to choose parts for its pipeline that offer the best balance of performance characteristics for its particular load.

Knife-gate valves control the flow in many process piping systems.

Mining companies use piping systems to transport newly mined minerals, such as gold, ore, and coal, to processing plants. The excavated materials are crushed and suspended in a liquid slurry. An efficient slurry handling system is crucial to timely mineral processing, which is necessary for fast delivery.

The slurry flow can be very abrasive and corrosive to the hundreds of valves directing its materials. In mining, newly crushed ore has a sharp surface, can be quite hot, and flows quickly, constantly, and often at high pressure. A slurry valve must be designed for these conditions to reduce maintenance time and replacement costs. (See Fig. C-430.)

Power companies, meanwhile, transport different materials, putting their own set of demands on the line's components.

One valve OEM, Clarkson Co., USA, has simulated its valve in action to test how variations in material and design of the product will hold up under different pipeline stresses.

This OEM designs and manufactures knife-gate and control valves that can halt and isolate sections of a slurry flow. Efficient control is necessary when the slurry must be delayed, inspected, or redirected. Knife-gate valves are also used in other applications, including industrial scrubber systems, waste-water treatment systems, and industrial process water systems.

This OEM's latest knife-gate designs are referred to as wafer-type valves because they are lighter and thinner than their predecessors, although they can handle higher pressures. The narrower valve fits tighter spaces and gives pipeline designers more flexibility. Wafer-type valve dimensions meet a nationwide standard, providing greater flexibility in choice of supplier because the valves are interchangeable.

The knife-gate valve has a blade-like steel gate that lowers into the slurry flow to create a bubble-tight seal. See Fig. C-431. The valve has two matching, smooth elastomer sleeves that seal the blade when the gate is closed and seal each other when the valve is open, so the slurry can flow through unobstructed. The elastomer sleeves are designed to resist abrasion and corrosion and to cover the valve's metal parts to shield them from wear.

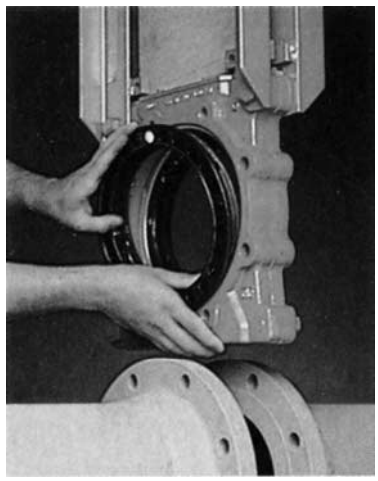


FIG. C-431 To increase its efficiency in different applications, a greater variety of elastomer seals for the wafer-type knife-gate valve were developed. (Source: Clarkson Co.)

The knife-gate valve was a new concept when it was introduced because it replaced conventional metal seats and gate guides with easily replaceable snap-in elastomer sleeves, which are more durable and versatile, and handle higher pressure and temperature. Conventional metal seats and gate guides can fill with hardened slurry and then fail to open or close.

Pipeline operators value the elastomer's long life because each seal costs between $75 and $500 to replace. More important, they lose revenue when they suspend the slurry system for maintenance repairs.

This OEM wants to develop a greater variety of elastomer seals for the wafer-type valve to increase its efficiency and reliability in different applications. For example, power companies prefer synthetic types of elastomers like neoprene, butyl or viton, which handle high temperatures and corrosive materials, while mining companies prefer elastomers like natural gum rubber for abrasive slurries.

Control, of (Fuel) Manifold Flow*

Stepper motor–driven valve

In the control of fuel flow to fuel manifolds in a gas turbine, the advent of the stepper motor–driven valve has brought about more accurate control of fuel supply to a gas turbine, with increased safety of operation, simplicity of piping design (see Fig. C-432), and reduced time between overhauls for the gas turbine.

This is by no means the only application for a stepper motor–driven valve, which is popular now in many processes and also in aviation applications.

Case study: stepper valve usage. The Petrochemical Corporation of Singapore (PCS) was initially discouraged from producing independent power in excess of its needs. Singapore Power (SP) has preferred to continue to receive the high tariffs paid by its consumers rather than administrate the buyback of small amounts of

* Source: Claire Soares, adapted from 1998 article written on "stepper" valves, PCS, and turbine fuel flexibility for *Asian Electricity*.

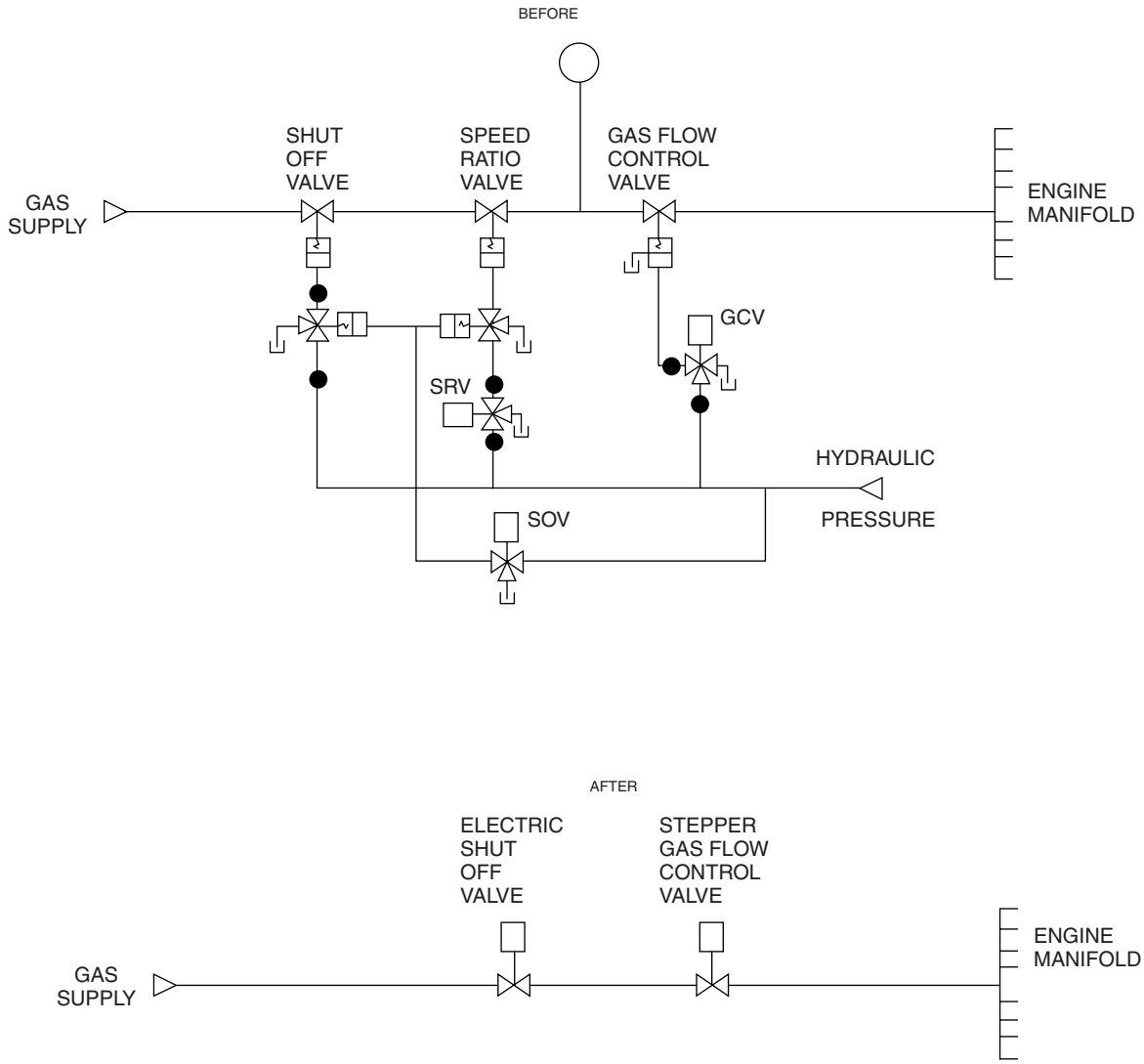


FIG. C-432 "Before" and "after" schematics showing how retrofit of a stepper value can simplify the piping and control system into a gas-turbine engine fuel manifold. (Source: HSDE.)

power from several small power producers (SPPs). However, now the move toward deregulation is changing that. "Pool rules for small generators," which covered generators of less than 10 MW and industrial in-house generators ("autogenerators"), were instituted in Singapore as of April 1, 1998.

An SPP such as PCS does not have the benefit of steady load, and the quality, type, and heating value of their fuels will vary. This is because they use process gases and fluids for fuel whenever they can, especially if that is the most cost-effective use for a process fluid. Due to the variations in the different characteristics of these fuels, which are in essence different process streams, a very fast response valve is required. Without such a valve, the exhaust gas thermocouples on the gas turbine would note larger swings in turbine exhaust temperature. The key to PCS's successful use of process fluids—which it didn't have much other use for—as fuel is valve response time and actuation characteristics. An ideal valve for this type of application is a "stepper" valve or its equivalent.

The stepper valve and functional equivalents. The stepper (short for stepper motor–driven) valve is a fast-response, electrically operated valve that was pioneered by Vosper Thornycroft (HSDE, UK) in the mid-1960s. Now this valve type is made by other well-known manufacturers too, such as Moog, in Germany. The term stepper actually refers to the motor type that drives the valve as opposed to the valve itself. The motor is a stepper motor, as opposed to a torque or AC or DC motor. Its self-integrating function ensures that the valve will proceed to a desired position and then the motor will stop. With other motors, the motor has to continue to run in order to keep the valve in that position—such valves need signals to cue them: run, stop running, then start running again, and so forth. If something were to happen causing the valve to fail, the stepper-type valve position would still lock and the system would continue running. The valve then makes the system fault tolerant, which is critical in applications such as emergency power-supply generators. It also provides the fast response required by aeroderivative and some industrial gas turbines. This is useful for both power generation and mechanical drive service. Before the stepper valve was introduced in the mid-1960s, hydraulic and pneumatic actuation valves were used to provide the required response time. This increased the overall complexity of the fuel system. As always, with instances where system complexity is heightened, system cost increased but mean time between failures (MTBF) and availability decreased.

The valve takes up very little space on the installation and service people unused to this new design spend frustrated time looking for the extensive "old" equivalent control system.

Development of valves that could compete with HSDE's original stepper arose from competition with that early design. As a result, there are now many manufacturers who produce functional equivalents on the market for use in gas-turbine fuel systems, high-resolution controls for robots, automatic machining controls, and so forth. In PCS's application, they use a Moog (German manufacturer) valve that has a DC motor. To get the same "stay in position" feature as a stepper-type valve would have, manufacturers typically use a spring to hold a position.

Design aims of fast response valves. The original design aims of the stepper-type valve and its equivalents generally include the following safety considerations:

- A fail-freeze or fail-closed option, depending on whether the operator is a power-generation facility ("freezing" at the last power setting is then required) or a pipeline (in which case turbine shutdown on valve failure is required).
- The liquid fuel version of the valve incorporates a pressure-relief valve protecting the system against overpressure and the fuel pump running on empty or "deadheading," caused by closure of valves downstream of the fuel valve during system operation.
- High-speed response of less than 60 ms required by aeroderivative gas turbines to prevent overspeed in block off-load conditions.
- Explosion-proof actuation to appropriate specification standards allows operation in hazardous methane service.
- Resistance to fuel contaminants, including tar, shale, water, sand, and so forth.
- Twenty-four volts DC is the maximum drive voltage that ensures personnel safety
- Corrosion resistance in components exposed to wet fuel and to all parts if the service is sour gas.

Operational objectives of fast-response valves. Other operational objectives that dictate design features are operator's requirements for:

- Low mean time to repair (LMTR). The target of 1 h, achieved with modular design, together with the target MTBF provided an availability of 99.998 percent for HSDE's original stepper.
- Higher MTBF. (In HSDE's case, an initial development target of 50,000 h was set and achieved.)
- Low maintenance costs, since the modular design can be repaired by an individual with relatively little experience. Service intervals are 12 months.
- Large control ratio that allows control over the ignition to full load as well as full-speed ranges to be possible with one fuel valve. Fuel pressure variation compensation is provided. The additional speed ratio–type control valve found in many other industrial gas fueled installations is not required here.
- Low power consumption since an electric motor of less than 100 W is used.

This also eliminates the need for additional hydraulic or pneumatic systems. Also, black starting is more reliable if the fuel system is powered by the same batteries as the controller.

A generic stepper valve system can consist of the following items:

- Transformer
- Power supply
- Controller
- A rack
- A DC motor

The motor, a brushless servomotor, is generally an electronically commutated three-phase motor with permanent magnet excitation on a low-inertia rotor. The stator's three-phase windings have thermistor protection. Sinusoidal back electromotor force (emf) provides improved low-speed performance and higher efficiency. The integral brushless resolver provides position feedback so no tachogenerator, mechanical commutator, or electronics within the motor are required. Bearings are preloaded and sealed with high-temperature grease. A static load holding brake is an option.

Applications experience. Power production in phase II of the Petroleum Corporation of Singapore was commissioned in June 1997. PCS is part of a massive petrochemical plastics conglomerate in Singapore. Power production was an afterthought, since when they were built, its design did not include provision for becoming an SPP. PCS chose a nominally 25-MW (23 MW in their normal ambient conditions) Alstom GT10, although their power needs are roughly 26 MW. This was because while SP were pleased to sell them their residual requirement, they would not buy any power from SPPs at the time of original power plant design.

The turbine is fueled by three different types of fuel, depending on the state of the plant. The British thermal units for each type varies, so again the fast response time for the stepper valve is critical.

As PCS operations found, the fast response valve proved as useful as the stepper valve has been for power generation on the North Sea oil and gas platforms. The fast response time of the stepper valve design helps the valve avoid the sudden burst of excess temperatures that accompany higher heating value fuel. (North Sea platform users frequently operate gas, liquid, or gas and liquid fuel mixtures.)

Not all gas turbines are tolerant of a wide range of fuel types in a single application. Some of them require a whole different fuel system—nozzles, lines, and

all components—to be able to handle a totally different heating value fuel. In this application in Singapore, the Alstom machine shows no sign of distress, which is interesting since the heating value of the fuel types varies as much as 50 percent. The exact fuel composition data are proprietary to PCS only; however, the GT10 operational data chart, figures, and curves here provide data for typical power output with light oil and natural gas fuel. The rest of the data is typical for the GT10.

PCS's GT10 heat-recovery steam generator (HRSG) provides a reliable source of steam. The plant exports steam to the nearby Seraya Chemicals plant in addition to fulfilling its own needs.

The turbine. The relatively low turbine-inlet temperature of the GT10 is one of the keys to being able to use three different fuel streams that exhibit a divergence of 50 percent in terms of heating value, without any noteworthy surges in performance or reductions in hot section component lives. As already described, valve-response speed is another critical feature for ensuring the stability of this application.

Emissions and steam supply. The Alstom EV burner—a low NO_x burner that can be fitted and retrofitted on the GT10, fuel types permitting—was not fitted in this case. The EV burner will handle clean natural gas and clean diesel fuel. It was not suitable for the high hydrogen content and variations in fuel composition that this application involves. Such fuels need a more forgiving fuel system, as well as water or steam injection to keep the NO_x down. The PCS Singapore application uses steam for NO_x reduction purposes. The steam is piped in through nozzles that are adjacent to the fuel nozzles on the fuel manifold of the GT10's annular combustor.

The source of the steam is the HRSG that is packaged as part of the GT-10 system. If and when required, the plant also can draw high-pressure steam from its process cracker.

In PCS's case, one boiler has been found to suffice. This is noteworthy as in applications such as this, a redundant "packaged boiler" (running hot and on minimum load) is often found essential. This is so that it is possible to pick up the steam load should the turbine trip or be unavailable due to maintenance. A common subject for debate is whether uninterrupted steam supply during the switch from HRSG mode to fresh-air firing is possible without flameout on the boiler supplementary burners.

The PCS plant is part Japanese owned, so the specifications the installation had to meet matched those of environmentally particular Singapore, as well as the Japanese, who are the most environmentally strict practitioners in Asia. Steam injection reduces NO_x levels from 300 to 400 mg/MJ fuel to just below 100 mg/MJ fuel.

(*Note*: As the upstream company PCS's main role is to supply high-quality ethylene, propylene, acetylene, and butadiene, as well as utility services such as water, steam, and compressed air, to downstream companies, PCS directly produces and exports benzene, toluene, and xylene for global markets.)

Future potential for power generation in Singapore. The pressure for accelerated deregulation is increasing in Singapore as well, if not as fast as in the rest of southeast Asia. Singapore Power is gradually seeing more IPP contracts let in the country. SPC's experience with the GT10 they operate has been positive in terms of availability and maintainability. Just as important is this gas turbine's ability to use three different "waste" petrochemical fluids as fuel, despite the 50

percent difference of these three fluids in terms of heating value. That it can do this while maintaining NO_x emissions below legislated limits for countries as environmentally strict as Singapore, speaks well for its continued use in similar applications.

Conversion Tables (*see* Some Commonly Used Specifications, Codes, Standards, and Texts)

Conveyors* (*see also* Drives; Power Transmission)

With numerous process plants employing conveyors of one type or another, it was felt that this text should give at least an introduction to this type of machinery by focusing on one of the more sophisticated executions: steel-belt conveyors.

The use of steel-belt conveyors has spread throughout the processing industries. Applications of steel-belt conveyors include cooling/solidification, drying, pressing, freezing, baking, and materials conveying.

The steel belt is made from flat strip steel from a rolling mill, prepared through special techniques that straighten, flatten, and make the ordinary strip suitable for welding into endless bands continuously running around two terminals. The conveyors based on this specialized technology are designed for the processing industries according to the needs of the product and the special needs of the steel belt.

Table C-34 summarizes a wide variety of steel-belt applications and the important steel-belt properties that make the applications successful. The general categories that are shown in Table C-34 are material handling, food processing, industrial processing, and presses for particle boards, plastics, and rubber. Table C-34 also indicates the four major steel-belt grades that are in common use.

The following discussions describe the applications and processes for which steel-belt conveyors have been selected as the best of competing alternatives, including the types of materials used for conveyor belts.

Reference and Additional Reading

1. Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Coolant; Engine Coolant

The number of coolants available on the market is large. For illustrative purposes, the common one chosen here is propylene glycol. The discussion that follows† indicates the characteristics sought in most coolants.

Circulation of Coolant through a Typical Engine

Coolant circulates through passages in the engine block surrounding the cylinders. Coolant also flows through the cylinder head to cool the valves and combustion chamber area. The heated coolant then flows through a thermostat to either the radiator to be cooled or to the coolant pump for circulation back to the engine. (See Fig. C-433.)

---

*Source: Sandvik Process System, Inc., USA.
†Source: ARCO Chemical, USA.

TABLE C-34 Areas of Steel-Belt Application

	Steel Belt Installation Type	Important Steel-Belt Properties: Strength—Ambient Temperature	Strength—High Temperature	Strength—Cryogenic Temperature	Stability against Temperature Difference	Corrosion Resistance	Wear Resistance	Hygienic	Flatness	Straightness	Surface Finish	Possible Steel Belt: Carbon (C)	Austenitic (SA)	Martensitic (SM)	Strength Martensitic (SM)
Material Handling	Conveyors, general	•		(•)			•					X			X
	Sorting systems	•		(•)		(•)	•		•			X			X
	Work tables, general					(•)	(•)		(•)			X	X	X	
Food Industry	Meat cutting tables					•		•	(•)				X		
	Bottle headling	•				(•)	•		•			X			X
	Bake ovens	(•)			(•)		•		•	(•)	(•)	X			
	Contact freezers			•		(•)		(•)	(•)				X		
	Belt coolers/food				(•)	(•)		(•)	(•)				X	X	
Industrial Process Applications	Belt coolers/ chemicals				(•)	(•)		(•)	(•)				X	X	
	Steel-belt dryers		(•)		(•)	(•)			•	(•)	(•)	X	X		
	Flow-through belt unit				(•)	(•)			(•)			X	X	X	
	Double belt presses	•			•	(•)			•		(•)	X	X		X
	Belt skimmers				•								X		
Particle Board Industry	Single opening presses				•	(•)	•		•	•		X			X
	Multiopening presses				•	(•)						X		X	
	Rotation presses	•			•	(•)			•	•			X		X
Rubber	Rotation presses	•							•	•	(•)		X		X
Plastics	Rotation presses	•				•			•	•	(•)		X		X

(•) = property sometimes of importance
• = property always of importance
SOURCE: Sandvik Process System, Inc.

Propylene glycol (PG) is a recent innovation in improved antifreeze formulations. Its key advantage over more traditional engine coolants made with ethylene glycol (EG) is lower toxicity to people, pets, and wildlife.

PG coolants have been extensively tested in both heavy-duty and automotive service. Laboratory, engine dynamometer, and fleet tests all prove that PG is an

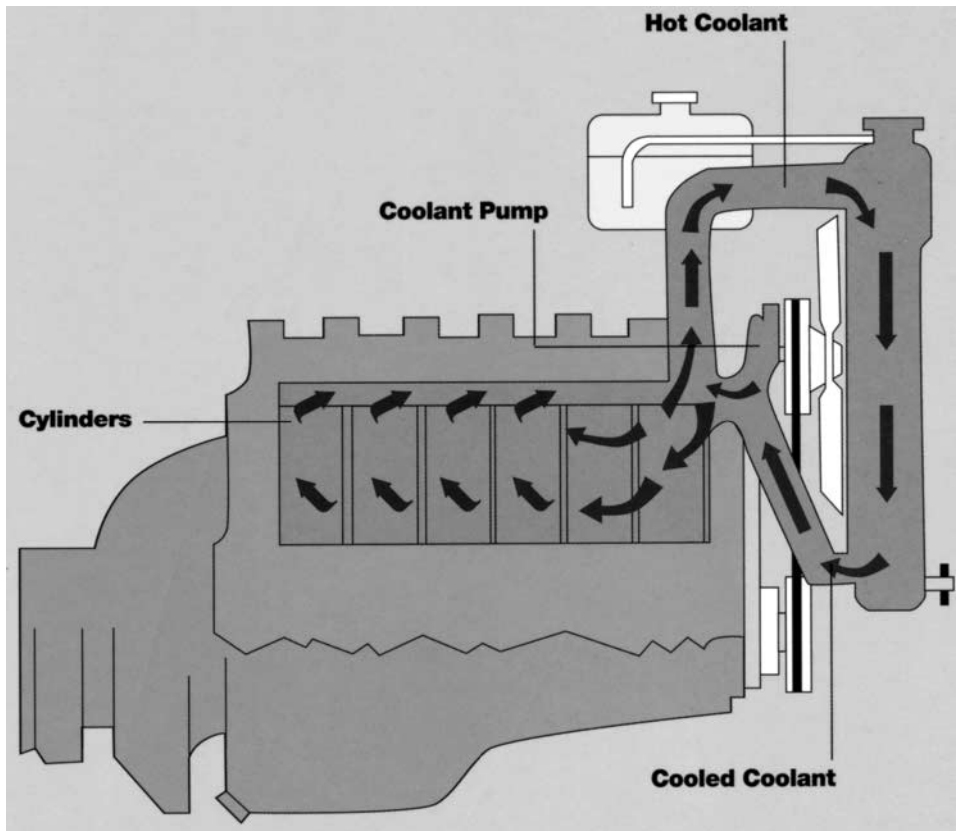


FIG. C-433 Coolant flow through an engine. (Source: ARCO Chemical.)

excellent base fluid for engine coolants, providing the necessary heat transfer characteristics, boilover prevention, freeze protection, and, when adequately inhibited, corrosion protection.

PG coolants may be recycled using the same methods used by many maintenance facilities to recycle traditional EG coolants, and similar additives can restore the recycled product to meet requirements for virgin coolants.

Commercial antifreeze formulations based on PG are readily available for both heavy-duty fleet and automotive uses. In service, PG/water or PG/EG/water antifreeze compositions and freeze protection may be easily measured using inexpensive, commercially available devices.

Results of comparison tests have demonstrated that propylene glycol is as effective an engine coolant as ethylene glycol. The testing program included laboratory bench-scale testing, engine dynamometer studies, and fleet tests on both heavy-duty and light-duty vehicles. Both high- and low-load conditions were studied at ambient temperature extremes of –43 to 49°C (–46 to 120°F).

The effectiveness of a heat transfer fluid in modern gasoline and diesel engines depends on two factors—the removal of heat from the engine as the liquid circulates through the cylinder head and engine block, and the transfer of heat to the air by the radiator. Detailed studies and analysis of the mechanisms of heat transfer have shown that although there are very slight differences between the heat transfer properties of propylene glycol and the more traditional ethylene glycol coolants, these make no difference in operating vehicles.

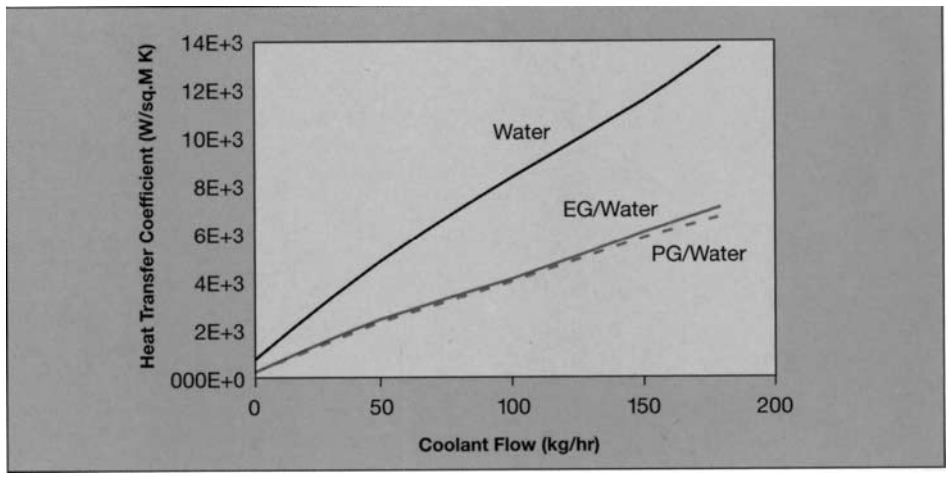


FIG. C-434 Forced convection heat transfer. (Source: ARCO Chemical.)

Engine Cooling Effectiveness

PG has proven to be as effective a coolant as EG. Removal of heat from the engine depends on heat transfer coefficients, which are dependent on the mechanism of heat transfer. At low heat flux (heat transfer rates), forced convection (heat transfer between a solid metal surface of the engine and the liquid coolant) is the predominant mechanism of heat transfer. In this regime, the Prandtl number, which is dependent upon coolant physical properties, is the controlling factor in heat transfer. At higher heat flux, the metal temperature increases and nucleate boiling (formation of some vapor bubbles at the heat transfer surface) occurs, increasing the efficiency of the heat transfer. Under nucleate boiling conditions, the Prandtl number is no longer the controlling factor in heat transfer.

At still higher heat flux, the surface temperature continues to rise, and too much vaporization occurs for an efficient transfer of heat. A vapor film may form that blocks direct contact between the liquid coolant and metal surface. At this point, a dramatic drop in the heat-transfer coefficient occurs, causing a concurrent dramatic rise in the cylinder metal temperature and possible engine failure. Ideally, a coolant will operate in the convective and nucleate boiling regions and never reach film boiling stage.

Theory predicts that, because a PG/water coolant mixture has different physical properties than an equal solution of EG/water (e.g., slightly higher viscosity and slightly lower density), convective heat transfer will be 5–10 percent lower, while heat transfer in the nucleate boiling range will be 5–10 percent higher. Figures C-434 and C-435 show the calculated heat-transfer coefficients for 50 percent solutions of PG/water and EG/water. Data for water is included for comparison. Detailed laboratory and engine dynamometer studies of heat-transfer coefficients have confirmed these predictions. These differences in the heat transfer over the combination of low and high loads seen in normal vehicle operation approximately offset each other.

With all else being comparable, the major difference between propylene and ethylene glycol is their toxicities. Two ounces of ethylene glycol antifreeze can be lethal to a dog, while a teaspoonful may kill a cat.

When ingested by humans and animals, EG is metabolized to glycolic and oxalic acids, which may cause acid-base disturbances, kidney damage, and possibly death.

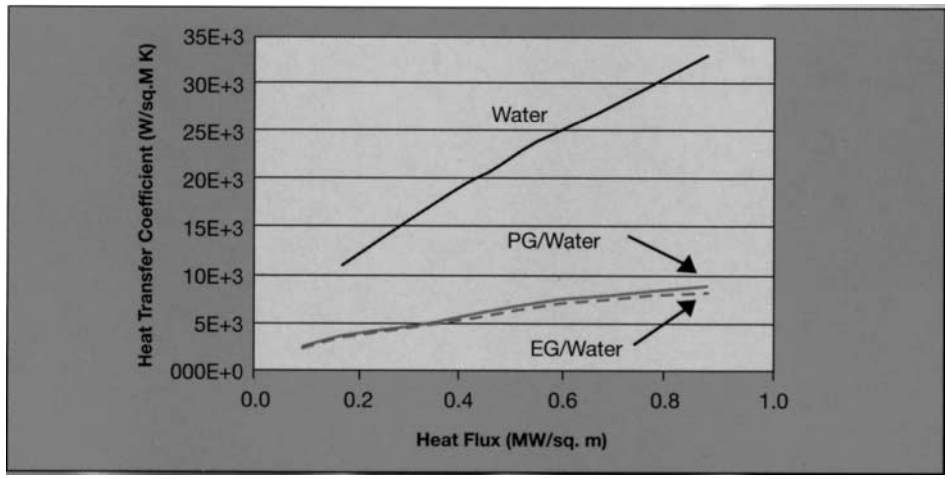


FIG. C-435 Nucleate boiling heat transfer. (Source: ARCO Chemical.)

In 1994, 4200 incidents of EG poisoning were reported by the American Association of Poison Control Centers, resulting in 29 deaths. EG is also a major source of poisonings of dogs and cats. According to an ASPCA survey of veterinarians, over 100,000 pets were poisoned in 1995 by accidentally ingesting EG antifreeze, resulting in more than 90,000 deaths.

In contrast, propylene glycol and its metabolites, lactic and pyruvic acid, account for its low toxicity in both acute and long-term exposures. Many of the harmful consequences of accidental antifreeze poisonings could be avoided by replacing EG-based coolant with PG coolant.

Because of this clear difference in toxicity, ethylene glycol is regulated by numerous federal and state health environmental acts in the United States, as summarized in Table C-35. These include the 1990 U.S. Clean Air Act Amendments and CERCLA. The U.S. Federal Health and Safety Administration also requires a specific label on ethylene glycol coolants warning of the toxicity.

In the European Union, ethylene glycol is classified as a hazardous product and must be handled according to strict regulations. In Switzerland, Austria, and the Czech Republic, it is classified as a poison and its sale to the general public is carefully regulated.

Coolers, Dairy

Glycol is an effective and fast coolant, commonly used in the food and agriculture industry. Often used in large-scale industrial systems, glycol can also be used in smaller customized designs. One example is a system designed in Hawaii's dairy sector for chilling the milk from cows on a small farm (300 to 900 cows). Milk temperature is lowered from 98 to 38°F. The standard coolers available worked well in competitive dairies in the rest of the United States, as the ambient temperatures there do not impose as much of a heat load as in Hawaii. Milk typically takes 2 h to cool in a large modern industrial dairy.

A standard air conditioner was fitted with an 800-gal glycol tank. The milk was found to cool in 30 s (the time for milk to flow through the system). A variable speed motor moves the milk through the cooler. The refrigeration unit incorporates two scroll compressors and brazed plate evaporators.

TABLE C-35 U.S. Statutes Relating to Glycols

U.S. Regulation	Propylene Glycol	Ethylene Glycol
1990 Clean Air Act Amendments (CAAA)	None	"Hazardous Air Pollutant" 42 U.S.C. § 7412 (b)
CERCLA (Superfund)	None	"Hazardous Substance" HAPs from 1990 CAAA are included into 42 U.S.C. § 9601 (14)
Superfund Amendments & Reauthorization Act (SARA)	None	"Toxic Chemical" 42 U.S.C. § 11023 42 C.F.R. § 372.65
Safe Drinking Water Act (SWDA)	None	Health Advisory Issued
Occupational Health & Safety Administration (OSHA)	None	50 ppm PEL 29 C.F.R. § 1910.1000
American Conference of Governmental Industrial Hygenists (ACGIH)	None	50 ppm TLV
Federal Health & Safety Administration (FHSA)	None	"Warning, Harmful or Fatal if Swallowed" 16 C.F.R. § 1500.132

Adjustable speed vacuum pumps replaced the constant speed milking pumps for further energy conservation. Overall about $6000 in 1997 prices was saved in energy costs with these modifications for a cooler handling output from 1000 cows. See Figs. C-436 and C-437.

Cooling; Cool, Products That (Air Conditioners); Liquid-Cooled Air Conditioners (*see also* Chillers)

Cooling (Using) Solid-State Technology

A key requirement among cooling products in the process engineers' world is that they adhere to environmental safety standards. Existing coolers may have "old" refrigerants, such as chlorofluorocarbons (CFCs), corrosive liquids, and gases, in their cooling circuitry. What follows is a summary of design environment specifications and standards, product selection charts, information on how to size an air conditioner, typical mounting configuration, theory of operation, and some example application illustrations.

Typical design environment (NEMA, Mil-Std, NED, UL/CSA) specifications* (see Figs. C-438 through C-441)

Typical NEMA	(Source: NEMA Publication No. 250, Part 1, Page 1)
NEMA-12	Type 12 enclosures are intended for indoor use primarily to provide a degree of protection against dust, falling dirt, and dripping noncorrosive liquids.
NEMA-4X	Type 4X enclosures are intended for indoor and outdoor use primarily to provide a degree of protection against corrosion, windblown dust and rain, splashing water, and hose-directed water.

*Source: Thermoelectric Cooling America Corporation (TECA), USA.

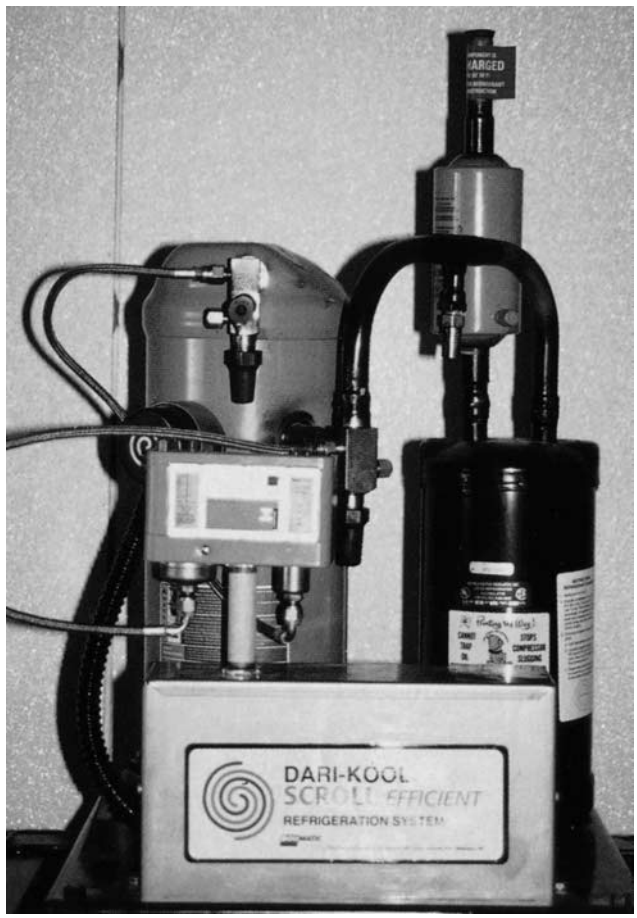


FIG. C-436 These twin high-efficiency scroll compressors are part of a small commercially available cooling system used to chill milk. (Source: *Mechanical Engineering Power*, ASME, November 1997.)

This OEM's products carrying the NEMA-4X designation use Mil-Spec fans, O-ring sealed power supplies, no exposed electronic components, stud/gasketed mounting, and Mil-Spec finishes on exterior. They are designed to maintain enclosure rating and perform in the rated environment.

Military Standards

Mil-Std 810	Corrosion	(Salt Fog Testing) Method 509.2, 168 Hours, Employed for all NEMA-4X units
	Vibration	Method 514.3, 2 hours, x, y, z axis 8.9 G's, 10–2,000 Hz with a magnitude of 0.04 G^2/Hz, Employed for all XM- Versions, Standard models are designed to withstand 2.2 G's.
	Shock	Method 516.2, with 30 G's peak amplitude, 11 ms pulse duration, half-sine waveform, and three (3) shocks in each direction along three (3) mutually orthogonal axes, Employed for all XM- Versions

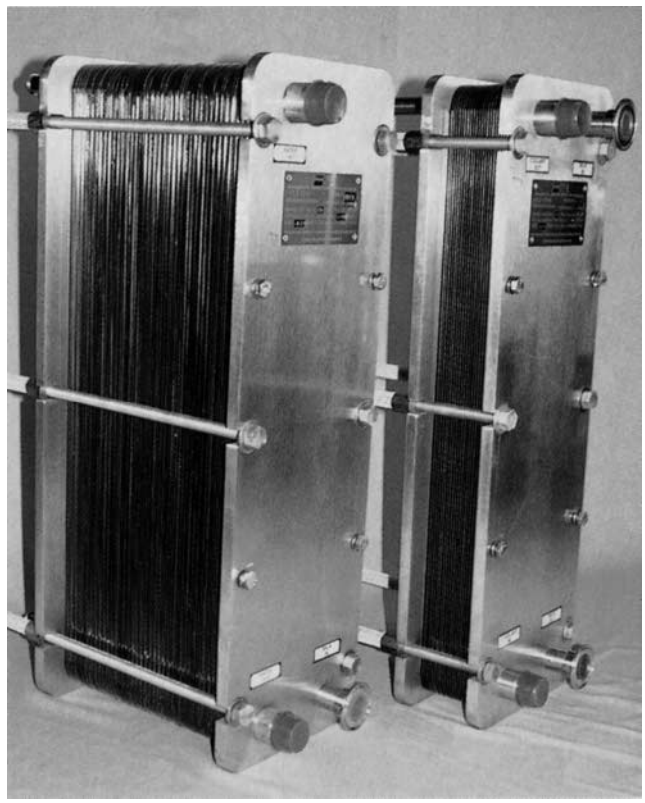


FIG. C-437 Brazed-plate evaporators are outfitted with high-efficiency scroll compressors to keep the quality of dairy products from deteriorating in Hawaii's tropical climate. (Source: *Mechanical Engineering Power*, ASME, November 1997.)

NEC (Source: NEC 1993, Article 500, 70-466 to 70-471)

CID2	Class 1, Division 2 (Hazardous Environments) A Class 1, Division 2 location is a location (1) in which volatile flammable liquids or flammable gases are handled, processed, or used, but in which the liquids, vapors, or gases will normally be confined within closed containers or closed systems from which they can escape only in case of accidential rupture or breakdown of such containers or systems, or in case of abnormal operation of equipment; or (2) in which ignitable concentrations of gases or vapors are normally prevented by positive mechanical ventilation, and which might become hazardous through failure or abnormal operation of the ventilating equipment; or (3) that is adjacent to a Class I, Division 1 location, and to which ignitible concentrations of gases or vapors might occasionally be communicated unless such communication is prevented by adequate positive-pressure ventilation from a source of clean air, and effective safeguards against ventilation failure are provided.
Groups (A–D)	Atmospheres containing the following: acetylene, hydrogen, fuel and combustible process gases containing more than 30 percent hydrogen by volume, or gases or vapors of equivalent hazard such as butadiene, ethylene oxide, propylene oxide, acrolein, ethyl ether,

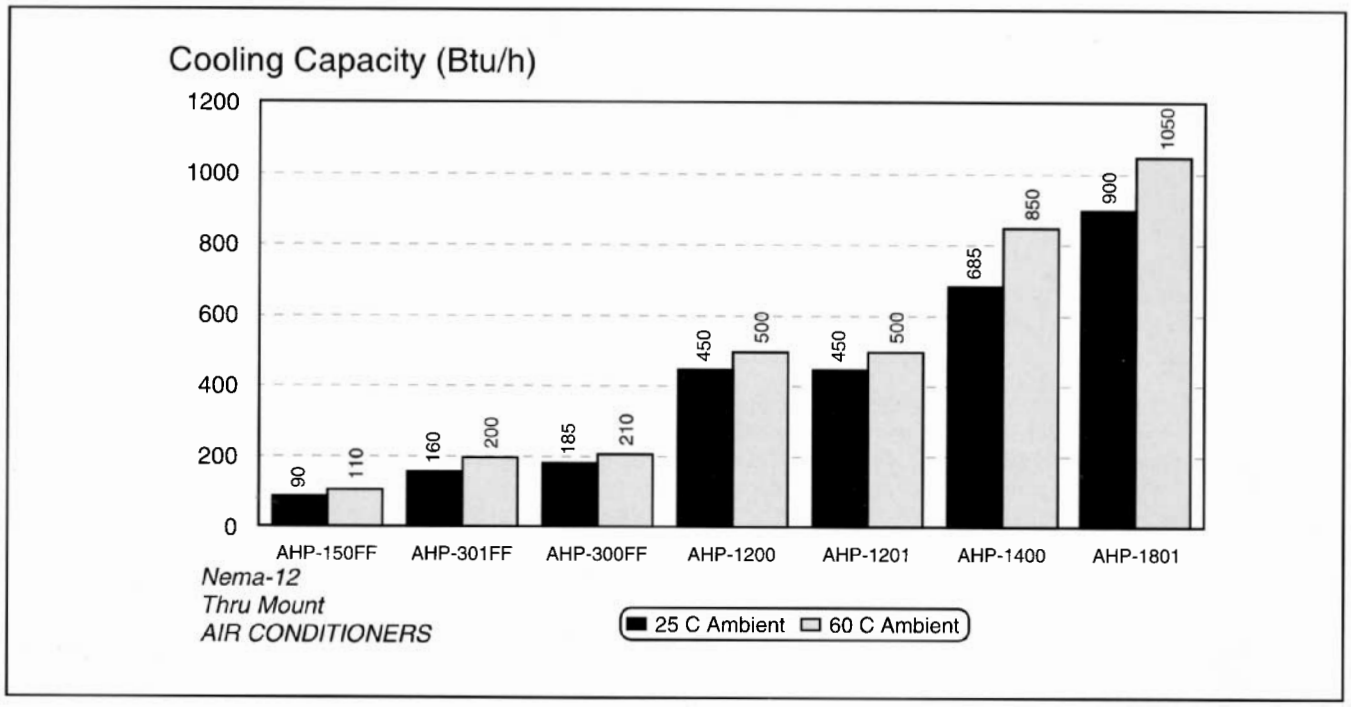


FIG. C-438 Cooling capacities of various air conditioners. (Source: TECA.)

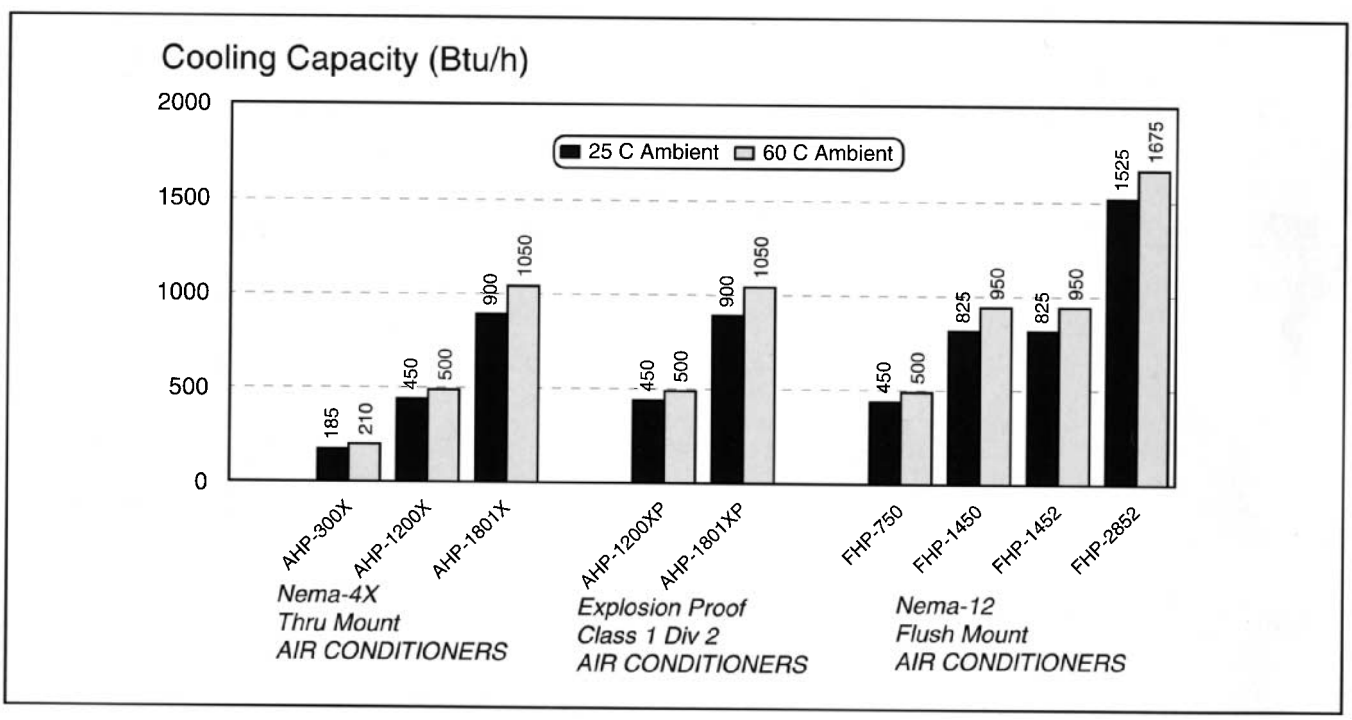


FIG. C-439 Cooling capacities of various air conditioners. (Source: TECA.)

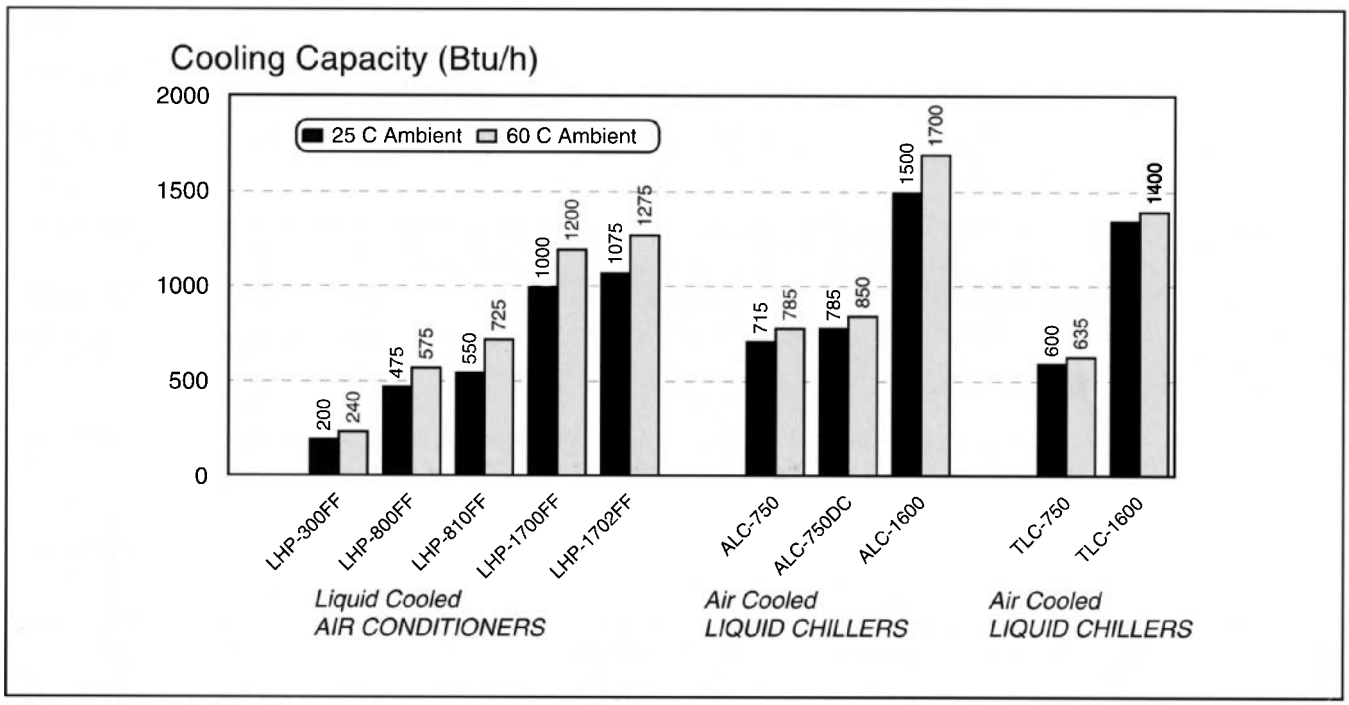


FIG. C-440 Cooling capacities of various air conditioners. (Source: TECA.)

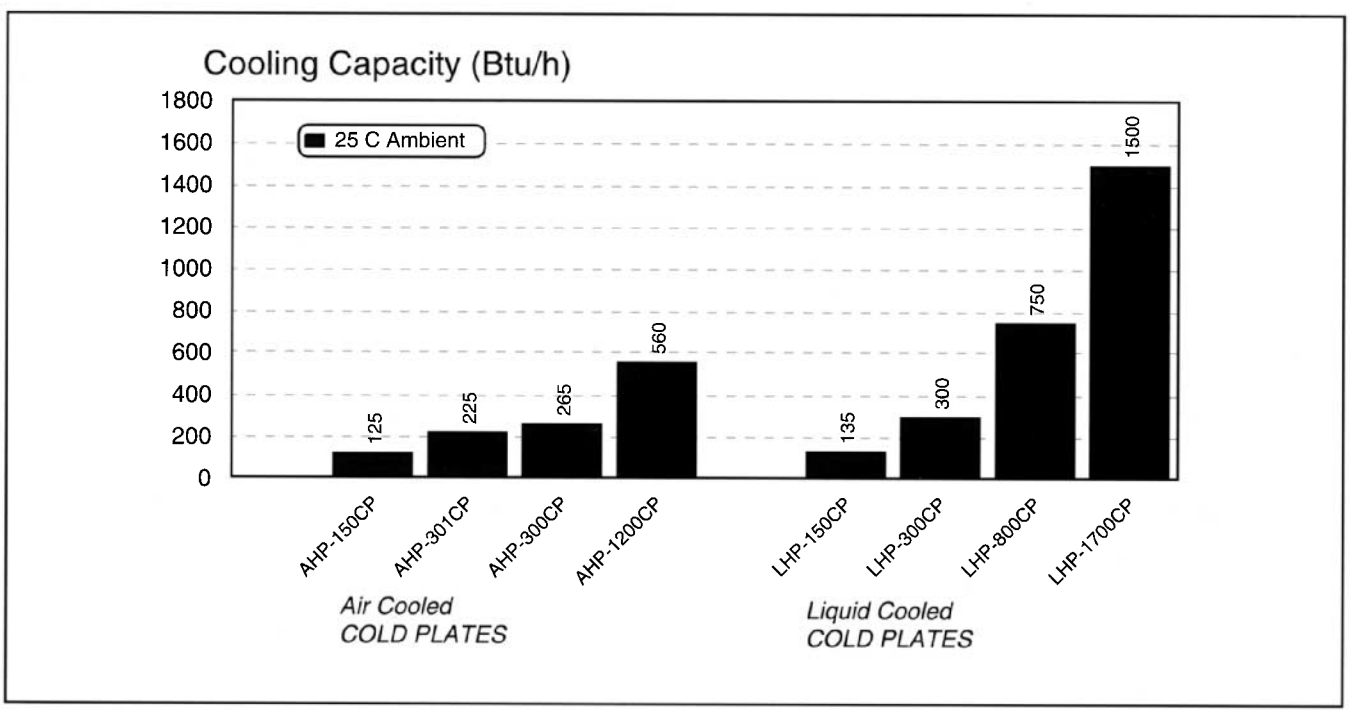


FIG. C-441 Cooling capacities of various air conditioners. (Source: TECA.)

ethylene, or gases or vapors of equivalent hazard, acetone, ammonia, benzene, butane cyclopropane, ethanol, gasoline, hexane, methanol, methane, natural gas, naphtha, propane, or gases or vapors of equivalent hazard.

[*Note*: Applies to this OEM's models AHP- (1,200XP, 1,200XPHC, 1,801XP, 1,801XPHC)]

UL/CSA Standards

UL-1604	Hazardous duty operation, Class I and II Division 2, Class III Div 1 and 2 Tested thru ETL and ETLc Testing Laboratories, Report # 532015 Applies to models AHP- (1,200XP, 1,200XPHC, 1,801XP, 1,801XPHC)
UL-1995	Heating & Cooling Equipment, Categories 169 & 294, No. 236-M90
CSA 22.2	Tested thru ETL and ETLc Testing Laboratories, Report # 532015 Applies to models AHP- (1,200, 1,201, 1,200HC, 1,201HC, 1,200X, 1,200XHC, 1,801, 1,801X, 1,801XHC, 1,801HC)

Reliability and mean time between failure (MTBF)

The life expectancy of a thermoelectric device is high due to its solid-state construction. Service life is typically in excess of five (5) years, under normal conditions.

MTBFs on the order of 2–300,000 h at room temperature, 100,000 h at elevated ambients of 80°C, have been calculated.

Cooling, Thermoelectric

Theory of operation

Thermoelectric cooling, or as it is sometimes called, the Peltier Effect, is a phenomenon discovered by a French watchmaker during the early 19th century. It is described as a solid-state method of heat transfer generated primarily through the use of dissimilar semiconductor materials. To understand the cooling method, it is first necessary to know how thermoelectric cooling systems differ from their conventional refrigeration counterparts.

Like conventional refrigeration, thermoelectrics obey the basic laws of thermodynamics. Both in result and principle, then, thermoelectric cooling has much in common with conventional refrigeration methods—only the actual system for cooling is different.

The best way to show the differences in the two refrigeration methods is to describe the systems themselves. In a conventional refrigeration system, the main working parts are the evaporator, condenser, and compressor. The evaporator surface is where the liquid refrigerant boils, changes to vapor and absorbs heat energy. The compressor circulates the refrigerant and applies enough pressure to increase the temperature above ambient level. The condenser helps discharge the absorbed heat into the ambient air.

In thermoelectric refrigeration, essentially nothing has changed. The refrigerant in both liquid and vapor form is replaced by two dissimilar conductors. The cold junction (evaporator surface) becomes cold through absorption of energy by the electrons as they pass from one semiconductor to another, instead of energy absorption by the refrigerant as it changes from liquid to vapor. The compressor is replaced by a DC power source that pumps the electrons from one semiconductor

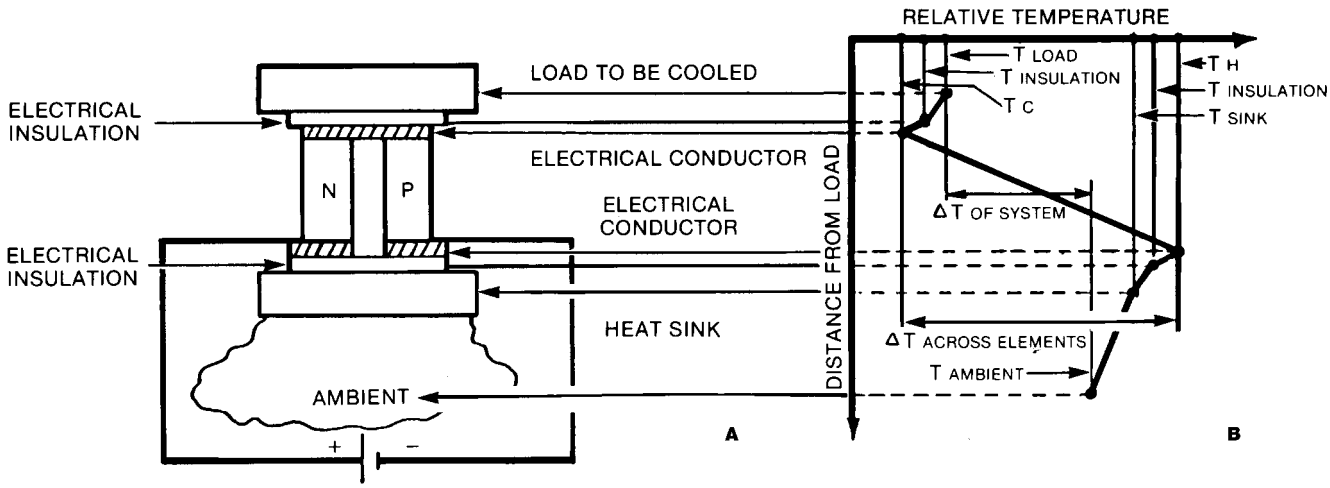


FIG. C-442 Semiconductor materials with dissimilar characteristics are connected electrically in series and thermally in parallel, so that two junctions are created (*A*). (Source: TECA.)

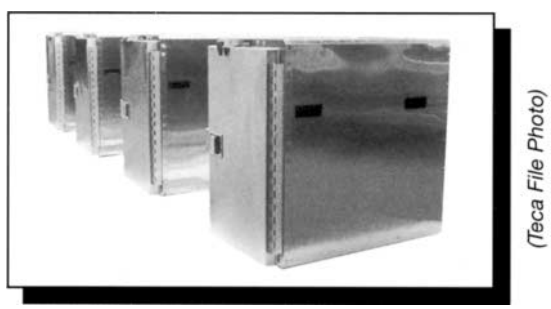


FIG. C-443 Food service refrigerators for airborne application. (Source: TECA.)

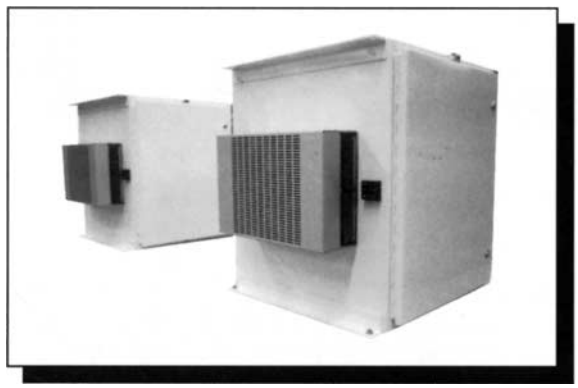
to another. A heat sink replaces the conventional condenser fins, discharging the accumulated heat energy from the system.

The difference between the two refrigeration methods, then, is that a thermoelectric cooling system refrigerates without use of mechanical devices, except perhaps in the auxiliary sense, and without refrigerant.

The semiconductor materials are N and P type (see Fig. C-442), and are so named because either they have more electrons than necessary to complete a perfect molecular lattice structure (N-type) or not enough electrons to complete a lattice structure (P-type). The extra electrons in the N-type material and the holes left in the P-type material are called carriers and they are the agents that move the heat energy from the cold to the hot junction.

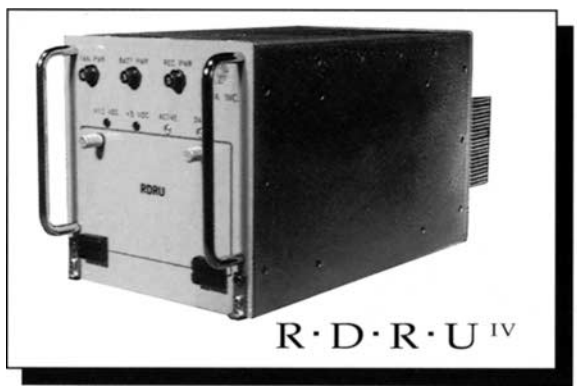
Heat absorbed at the cold junction is pumped to the hot junction at a rate proportional to carrier current passing through the circuit and the number of couples. Good thermoelectric semiconductor materials such as bismuth telluride greatly impede conventional heat conduction from hot to cold areas, yet provide an easy flow for the carriers. In addition, these materials have carriers with a capacity for carrying more heat.

Heat sinks. Some applications of thermoelectric coolers are illustrated in Figs. C-443 through C-448.



(Photo courtesy of Scientific Atlanta-Jet Propulsion Lab)

FIG. C-444 Cooled enclosure system for Tower Mt. Horn/electronics assembly. (Source: TECA.)



(Photo courtesy of Veda Incorporated)

FIG. C-445 R.D.R.U. (ruggedized digital recording unit) utilizes a thermoelectric heat/cool system for reconnaissance data collection, flight test and evaluation, and automotive test and instrumentation. (Source: TECA.)



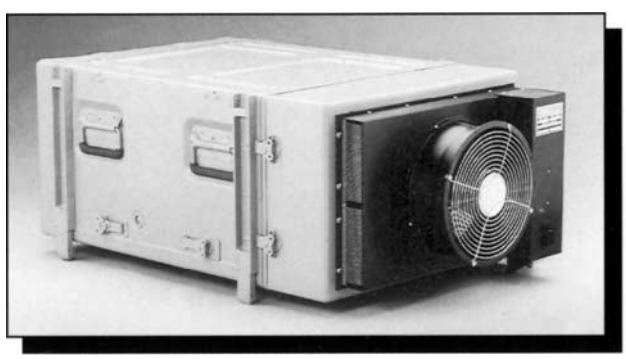
(Photo courtesy of New Zealand Dairy Research Institute)

FIG. C-446 One of the world's leading centers for dairy research uses thermoelectric cold plates with temperature control for tempering fat samples prior to pulsed nuclear magnetic resonance measurement of solid fat content. (Source: TECA.)



(Photo courtesy of Noah Precision)

FIG. C-447 A manufacturer in the semiconductor industry uses a solid-state liquid chiller to precisely control fluid temperatures for water-jacketed columns and etch baths. (Source: TECA.)



(Photo courtesy of EDAK)

FIG. C-448 A manufacturing specialist of transport equipment uses a solid-state cooling system to protect electronic equipment from harsh, high-stress conditions. (Source: TECA.)

The design of the heat exchanger is a very important aspect of a good thermoelectric system.

Figure C-442*B* illustrates the steady-state temperature profile across a typical thermoelectric device from the load side to the ambient. In Fig. C-442*B*, the total steady-state heat that must be rejected by the heat sink to the ambient may be expressed as follows:

$$\underset{\text{rejected}}{\text{Heat}} = \underset{\text{from the load}}{\text{Heat absorbed}} + \underset{\text{input}}{\text{Power}} + \underset{\text{leakage}}{\text{Heat}}$$

$$Q_S = Q_C + V \times I + Q_1$$

If the heat sink is not capable of rejecting the required Qs from the given system, the temperature of the entire system will rise and the cold junction temperature will increase. If the thermoelectric current is increased to maintain the load temperature, the COP (coefficient of performance) tends to decrease. Thus, a good heat sink contributes to improved COP.

TABLE C-36 Air Conditioner Sizing

To size an air conditioner proceed with the following 7 steps.
A free-standing enclosure (3′ × 3′ × 2′) with 1″ insulation has been provided as an example.

Step			English	Metric
1	(Ta)	Determine the maximum ambient (outside) air temperature ✍ *(9/5 × °C) + 32 = °F or 5/9 (°F – 32) = °C*	+120°F	+50°C
2	(Te)	Determine the maximum allowable enclosure air temperature	+100°F	+38°C
3	(ΔT)	Determine temperature differential (Step 2 – Step 1) ✍ *9/5 × Δ °C = Δ °F or 5/9 × Δ °F = Δ °C*	–20°F	–12°C
4	(Sa)	Determine exposed surface area = 2(H × W) + 2(H × D) + 2(W × D) ✍ *(Exclude nonexposed surfaces) 1 m^2 = 10.76 ft^2, or 1 ft^2 = .0929 m^2*	42 ft^2	3.9 m^2
5	(Qa)	Estimate ambient load *(example uses 1-in insulation; see Fig. C-449)* *(Positive if cooling below ambient, negative if cooling above ambient)*	140 Btu/h	41 watts
6	(Qe)	Determine enclosure internal load ✍ *(Fill in actual, or use either method 1,2,3)*	341 Btu/h	100 watts

Method 1: For resistive loads, measure the electrical power into the enclosure and subtract the electrical power out. This approximates the electrical load generated inside the enclosure.
✍ *Voltage (volts) × Current (amps) = Power (watts) [1 Watt = 3.414 Btu/h]*
Method 2: If power cannot be measured directly, check with the manufacturer of each device and add the load (watts) from all internal components.
Method 3: Measure the steady-state temperature rise from ambient to internal with the enclosure completely sealed. See Fig. C-449 to estimate the internal load.

7	(Qt)	Total Load (Step 5 + Step 6) ✍ *(Add additional loads at this time, i.e., solar or radiated loads)*	481 Btu/h	141 watts

TABLE C-37 Using Performance Curves (See Fig. C-450)

X (Horizontal Axis)	Total Load Line	(Qt)	(watts or Btu/h)
Y (Vertical Axis)	Temperature Differential Line	(ΔT)	(°C or °F)

✍ *(ΔT) from ambient to enclosure. Values are (–) for below ambient cooling and (+) for above ambient cooling.*

There is usually a blue-shaded region on each performance curve. The upper end is performance at +25°C/+77°F ambient. The lower end is performance at +60°C/+140°F ambient. The shaded region includes performance from 25°C to +60°C.

✍ *Please note: A thermoelectric cooler is typically more efficient at higher ambients, due to inherent properties in the material. This is the opposite of conventional fluorocarbon systems.*

To use the performance curves, you need to know total load (Qt from Step 7 of Table C-36) and temperature differential (ΔT from Step 3). From the example you know that the total load is 141 watts. Draw a vertical line to intersect at this load.

Next, place a point within the shaded region and on the vertical line to approximate the actual performance at ambient of 50°C. Since 50°C is close to the lower border, place a point slightly above the border. Then extend a horizontal line through the point to intersect the y-axis (or ΔT line). For our example, we are at roughly –13°CΔT or –23.4°FΔT, which is greater than the required –12°C/–20°F.

Result: ✓ *Adequate cooling capacity with this model.*

Line equations are also provided below each performance curve. You can solve for ΔT by substituting the load, or solve for load by substituting ΔT.

Energy may be transferred to or from the thermoelectric system by three basic modes: conduction, convection, and radiation. The values of Qc and Q1 may easily be estimated; their total along with the power input gives Qs, the energy the hot-junction heat sink must dissipate.

Air conditioner sizing

See Tables C-36 and C-37. Note that ambient temperature is the control parameter. From these calculations, the air conditioner can be selected. Figures C-438 through

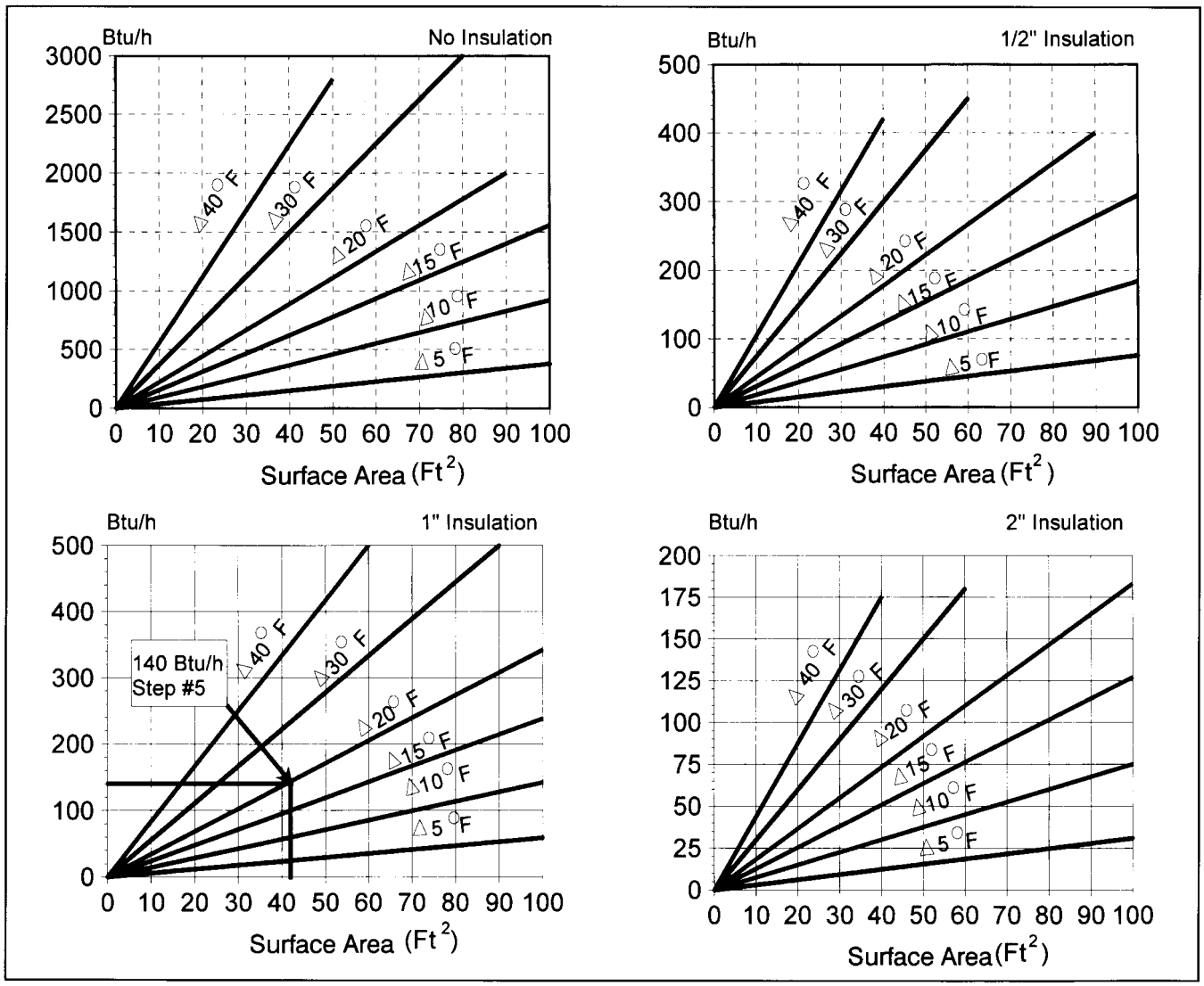


FIG. C-449 (Source: TECA.)

C-441 provide selection options based on ambient temperature. Note that model designations in these figures are specific to this information source. When looking at other original equipment manufacturers' products, look for equivalent or close to equivalent capacities.

Cooling Towers (*see also* Fans, Centrifugal)

Cooling towers cool process water using either direct or indirect contact with cooling air. Some tower designs have no contact with air, some have partial contact with air.

There are *evaporative-type* towers that get most of their cooling from the evaporation that occurs when air and water make direct contact. In the *dry tower*

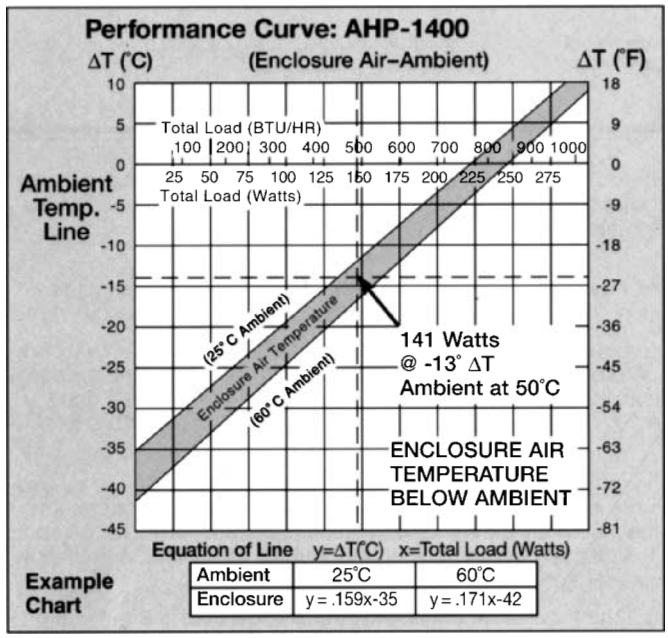


FIG. C-450 Typical performance curve for an air enclosure air conditioner. (Source: TECA.)

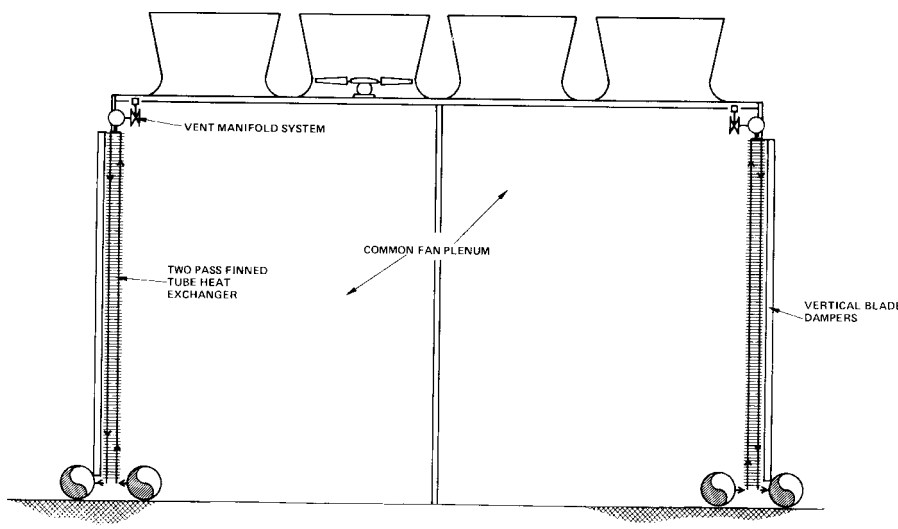


FIG. C-451 Dry-type cooling tower, cross-sectional elevation. (Source: The Marley Cooling Tower Company, USA.)

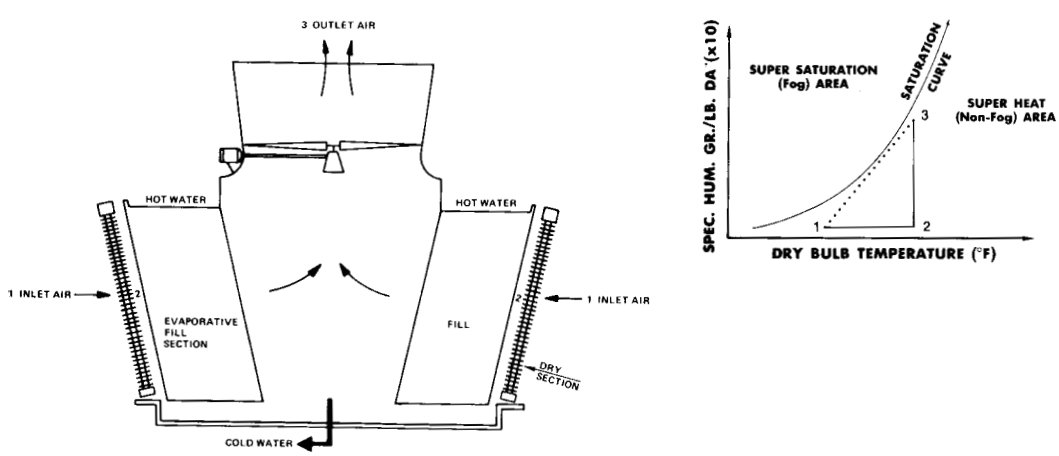


FIG. C-452 Plume abatement tower and psychrometrics (coil before fill). (Source: The Marley Cooling Tower Company, USA.)

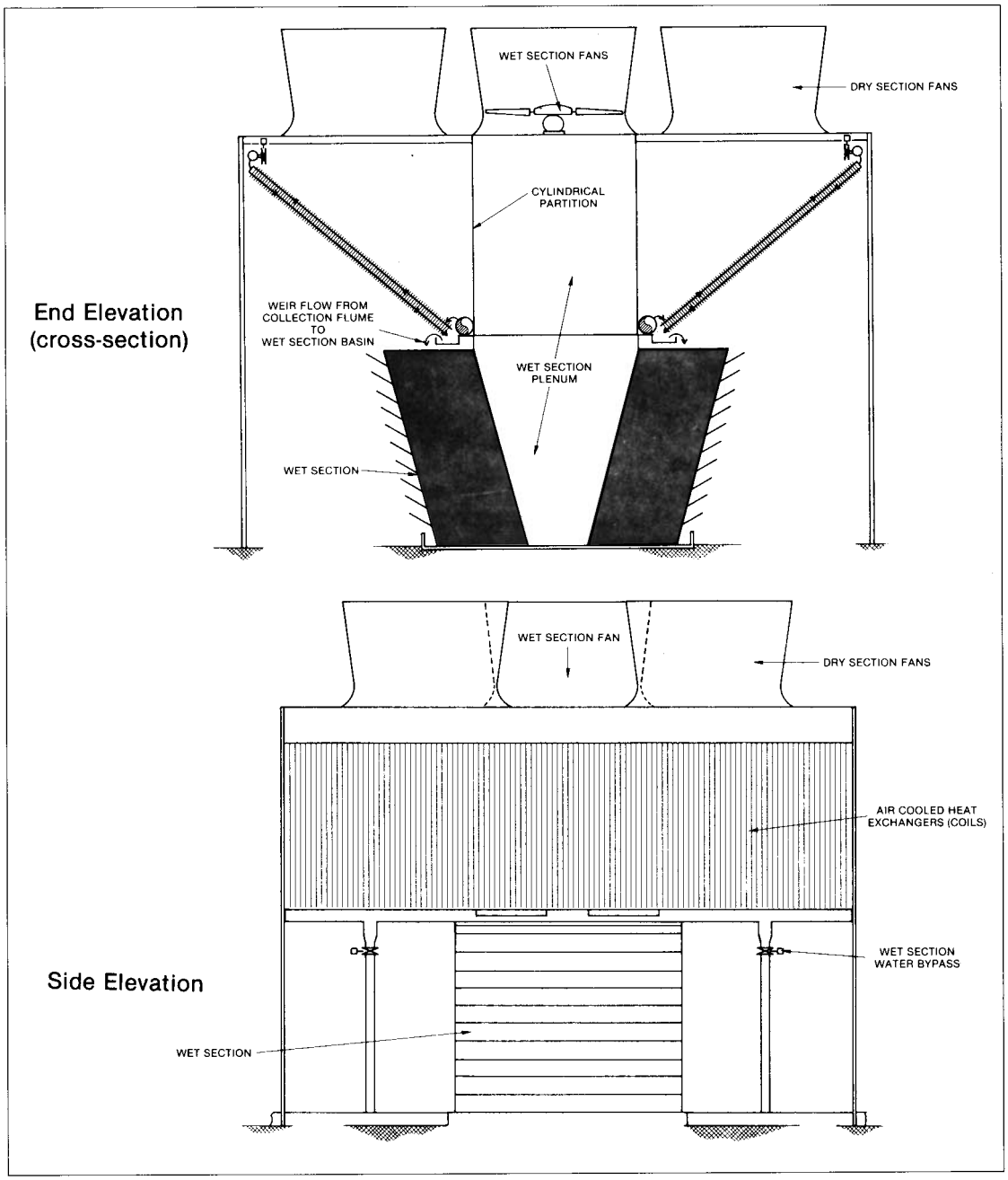


FIG. C-453 Water conservation cooling tower. (Source: The Marley Cooling Tower Company, USA.)

(Fig. C-451), by full utilization of dry surface coil sections, no direct contact (and no evaporation) occurs between air and water. Hence the water is cooled totally by sensible heat transfer.

In between these extremes are the *plume abatement* (Fig. C-452) and *water conservation* (Fig. C-453) towers, wherein progressively greater portions of dry surface coil sections are introduced into the overall heat transfer system to alleviate specific problems, or to accomplish specific requirements.

Reference and Additional Reading

1. Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Corrosion; Anticorrosion Coatings (*see also* Metallurgy)

Under "Metallurgy," a few basic methods of corrosion prevention and control, such as coatings on turbomachinery components, are discussed. The entire field of corrosion prevention is quite vast and includes techniques such as applying specific welding joints in certain applications. *See* Some Commonly Used Specifications, Codes, Standards, and Texts as well.

Couplings (*see* Power Transmission)

Crushers

Typically crushers use rollers, gyration of workpiece, or jaws to exert their crushing force. They are commonly used in agriculture and civil engineering type work to produce aggregate/pellets of a certain size. *See* Some Commonly Used Specifications, Codes, Standards, and Texts.

D

Dampeners (*see* Pulsation Dampeners)

Desalination

How Desalination Works

There are many different techniques for desalination. Among the main ones are the multistage flash (MSF) process and reverse osmosis. Countries that do not have an abundance of fresh water are prime candidates for the use of this technology. Quatar's installations include the Ras Abu Fontas B power and desalination plant, which cost $1 billion to build. Dubai in the United Arab Emirates (UAE) has a 60-million-gal/day desalination plant at Jebel Ali. The plant's eight MSF units are part of a cogeneration power facility. The fresh water would have been at least as much incentive in the Middle East as the increased total thermal efficiency. The desalination equipment in both cases was supplied by Weir Westgarth (WW).

WW designed the MSF process. The principle of the system is simple: Water and steam in a closed system can be made to boil at temperatures lower than at standard temperature and pressure by reduction of the system pressure. MSF plants contain a series of closed chambers—as many as 20—each held at a lower pressure than the preceding one.

Heated salt water is passed through the overall system. Some of the salt water in each chamber vaporizes into steam. Moisture droplet separators remove salt water droplets. The steam condenses to fresh water when faced with cold tubes and is collected for storage. The last chamber's brine is quite cool, and it is, in fact, used as the coolant fluid. Then it starts to pick up the latent heat of condensation and increases in temperature. Only a small amount of additional heat is required to prepare this steam for entry into the first flash chamber. One source of this steam is low-pressure steam from a power station.

The key to the reverse osmosis (RO) process is a suitable semipermeable membrane. Improvements in membrane technology now mean that the process can apply to industrial-scale plants. Common contemporary membrane selections are made of cellulose-based polymer or a polyamide layer applied to a microporous polymer film. This membrane is bonded to a porous polyester sheet for structural stiffness. This composite is rolled into a spiral. Spun hollow fine fibers are the finished product. The semipermeable layer is on the outside of the fibers. The total thickness of the composite is about 24 μm. The outside diameter of the tube is about 95 μm, making for a large surface area for rejecting salt. The fibers are made into bundles that are sealed with epoxy in a fiberglass pressure container.

The Global Drive for Desalination

The motivation level for incorporating desalination plants into plant infrastructure varies. The Middle East has fuel resources in abundance—natural gas, clean oil, and residual oil—and technology available that will help it burn each of these options with acceptable efficiency. Desalination technology also helps the Middle East achieve its major operational objective, which is to extend the time between overhauls (TBOs) and therefore parts life [as defined by life-cycle analysis (LCA)],

over the previous generation of power-producing machinery operated. Saving fuel for its own sake is not as critical to this market as it is to areas that do not have natural gas and oil in abundance. However, the Middle East has another reason for caring about efficiency—other than TBOs and LCA—that has emerged in the last decade: desalination. The Middle East is very short of fresh water.

The main forces behind the optimized commercialization of desalination technology are ironies in light of conditions in the Middle East. Concern regarding seawater contamination of freshwater aquifers in more freshwater-rich countries was one. (Sea water takes the place of fresh water as the aquifer pressure drops with water extraction.) The need to give nuclear energy a better image (by using its waste heat to produce fresh water) was another.

With respect to the first concern: desalination alleviates use of underground freshwater aquifers, which then reduces the risk of sea water seeping in to make up the balance of fresh water pumped out of the ground. Japan is gas and oil poor. A major user of nuclear power, it has been instrumental in promoting the use of desalination processes in conjunction with its nuclear facilities. The water is produced only in enough quantities to be used by the plants themselves. However, the Nuclear Power Technology Development section of the International Atomic Energy Agency (IAEA) has been seeking to promote large-scale desalination in conjunction with nuclear power production since 1989. The IAEA no doubt hopes that a freshwater supply would overcome the general public's reluctance to be situated close to a nuclear station. Water rises in cost if it has to be transported further.

Interestingly, despite the IAEA agreeing that MSF is promising, they are not contemplating using it for any large-scale attempts at freshwater production because of its inflexibility at partial load operation. MSF's tendency to corrosion and scaling versus other desalination techniques is also a factor.

Material Metallurgical Selections

The amount of sea water handled for a given membrane selection and seawater temperature varies directly with the applied pressure. Gulf sea water typically contains 19,000 ppm of salt. Typical temperature gradients are 20 to 30°C. Metallic corrosion at 30°C is 4 times what it would be at 20°C (double the rate of corrosion for each 5°C rise in temperature). This then requires corrosion-resistant alloy selections for the pumps.

In a design for a 25-year pump life for a Danish nuclear plant, Alfa Laval used copper nickel alloys in the evaporator and titanium for the heat-transfer tubing.

Desalinated water requires fewer chemical additives for water treatment. This is an advantage in terms of overall system cost.

Dialysis; Electrodialysis

Dialysis or *electrodialysis* is a specialized chemical process to separate components using differences in diffusion rates through membranes. A difference in concentrations drives the flow of the components in question. This process is used in chemical purification systems.

Distillation; Fractional Distillation (*see also* Towers and Columns)

Distillation is the evaporation and subsequent collection of a liquid that is a component in a mixture. *Fractional distillation* is the evaporation of two or more

liquids from a parent mixture by using the differences in their boiling points. Both processes may be used for purification or separation. A typical example of distillation is extracting pure solvent (such as water) from a mixture of solute and solvent (such as brine).

A typical example of fractional distillation is in the separation of various hydrocarbons, for instance, butane, pentane from a hydrocarbon mixture that results from some process in the overall refining process. In this example, a refinery fractional distillation apparatus is commonly termed a *fractional distillation column* or a *fractionating column* and can be several hundred feet high, depending on the refinery throughput. This column then consists basically of an outer shell, several metallic trays arranged through the column's height to enable the drawing off of various liquids. Pumps and pipes inside and outside the column conduct the fluid in required directions. Pressure and temperature in the column is controlled to the most suitable values for the process in question. Degrees of vacuum are used to accelerate separation rates.

A vessel of this nature is custom designed, generally by the design contractor for the overall plant. Besides all the relevant physical property tables, designers will use past experience to a considerable extent to determine the final parameters and dimensions for the column. In prototype applications some estimation or guesswork is unavoidable, which may have consequences not necessarily within the column itself, but with higher-precision elements of the process downstream. For example, a fractionating column-handling process oil in an oil sands plant had various molecular weights of hydrocarbon taken off at various points along its length. Based on flash point, one of these streams was designated as the source supply for the purged seal oil system in the plant. That stream contained more colloidal coke than was expected, making it unsuitable for the application. The problem was solved by taking the required supply from higher up the column (a "lighter end" of lower molecular weight).

Diverter; Diverter Damper; Diverter Valve; Flapper Valve

Diverter commonly describes a flat-plate-type valve hinged at one end that is moved to divert flow from one stack or passage to the other. There is normally no appreciable degree of speed or mobility attached to this type of device, as there might be with a butterfly valve in a control system. In a 150,000-barrel/day refinery, for instance, a flapper valve or flapper diverter valve can be made of concrete and weigh up to 5 tons. The absolute closure of the closure seal may not be critical, and the allowable gap between the flapper surface and the mating flange it sits on may be as large as 0.010 in.

Doctor

A *doctor* is a paper industry device. It is used to lead the paper sheet and keep the paper roll(s) clean.

Drives (*see also* Power Transmission; Turbines)

Drives is the term given to power-transmission equipment. The simplest form of power transmission is a belt drive. A belt drive can have a flat, V-belt, ribbed belt, or toothed belt design. These drives are common in applications such as conveyors. Conveyor manufacturers should be consulted for their catalogues on power-transmission capability.

In modern process industry, drive equipment might be classified as power-transmission equipment.

Drum; Knock-Out Drum; Knock-Out Vessel (*see* Separators)

Drying

These are many types of drying media and methods. A variety of methods to produce heat for drying (ovens) or air for drying (fans) or moisture absorption chemical (desiccant) are used and generally customized for a specific application. There are fluid-bed dryers, where solid particulates disperse heat to a gas. Sophistication may be added by using different stages for the dryer with dust collection (cyclone separators or otherwise) at each phase. In a pneumatic conveying system, an agitator supplied with warm air can be used to accelerate drying. With vacuum dryers, moisture removal is completed below atmospheric pressure. Drying temperatures, however, may vary considerably. Vacuum drying is particularly popular in pharmaceutical manufacturing, electronics, metallurgical, and food industries. *See* Some Commonly Used Specifications, Codes, Standards, and Texts.

Drying Equipment; Driers

The simplest kind of drying is "open-air" drying. From there, we can progress to fans, heaters, hot-air blowers, ovens, conveyor and oven systems, heat-exchanger provided heat, and a variety of other options, too numerous to concentrate on extensively in this book.

Drying, Freeze*

Freeze drying is sometimes what is meant by the term "drying" in the process engineer's terms. The following information is based on the solvent Tebol 99. It indicates what freeze drying is and the properties sought and the performance parameters measured in a typical freeze-drying agent.

Freeze drying or lyophilization is a process that removes a solvent, typically water, from a frozen solution by sublimation. Studies in the 1930s and 1940s were done on blood serum and foods. More recently, research has focused on using freeze drying for pharmaceuticals, cosmetics, and chemicals. An increasing number of parenteral products have been prepared by freeze-drying techniques. The method reduces particulate contamination, improves product quality and stability, and enhances the dissolution rate on reconstitution.

The pharmaceutical industry takes advantage of the freeze-drying process to maintain the activity and viability of various delicate biological materials. These materials include antibiotics, peptides, proteins, vaccines, and microbial cells. While freeze drying with water has proven useful, it has several inherent limitations:

- Uneven moisture distribution in the freeze-dried product
- Uneven stability or unpredictability of the final product
- Useless for water-insoluble or hydrolyzable products
- High energy costs
- Long process cycles

---

*Source: ARCO Chemical, USA.

Perhaps most important, freeze drying with water is restricted to those materials that are soluble and stable in a water system.

Much attention has been devoted to optimizing freeze-drying cycles. Recent studies have shown that addition of tertiary butyl alcohol (TBA) can markedly improve the freeze-drying process.

TBA as a processing aid:

- Helps dissolve products that are difficult to dissolve in water
- Gives a product with a high specific surface area
- Accelerates the drying process by reducing dried product resistance
- Prevents the product from reaching the collapse temperature
- Produces a pharmaceutically elegant product that can be reconsitituted easily

Typical physical properties related to freeze drying (see Table D-1)

TBA-water phase diagram. Figure D-1 shows the phase diagram for the TBA/water system developed by Kasraian and DeLuca. Water and TBA form a TBA hydrate. This complex phase diagram essentially consists of two simple eutectic phase diagrams placed side by side. The left side represents the eutectic phase diagram for water–TBA hydrate; the right side represents that of TBA hydrate–TBA. The maximum at 70 percent TBA corresponds to the melting of the pure TBA hydrate. The TBA-water system has two eutectic compositions, one at 20 percent TBA (eutectic A) and the other at 90 percent TBA (eutectic B). For the purpose of accelerating the freeze-drying process, only 5–10 percent of TBA is needed. In the 5–10 percent TBA concentration range, TBA and water form a eutectic mixture that has a melting point of –5 °C. Therefore, during freeze drying the product temperature should be kept below –5 °C.

Rates of sublimation of TBA/water solution. The relative rates of sublimation for each component depend on its concentration. Figure D-2 compares sublimation rates by plotting the molar ratio of TBA remaining to water remaining versus time. During the sublimation of a 20 percent TBA solution, the TBA/water ratio remains constant, which shows that both sublime at the same rate. Higher TBA concentration solutions have negative slopes, indicating that TBA is subliming faster than ice. Lower TBA concentration solutions show that ice was subliming faster than TBA. Figures D-3 and D-4 show appropriate clothing for operator handling of this product.

TABLE D-1 Typical Properties

1. Flash point (tag closed cup), °C (°F)	11 (52)
2. Vapor pressure at 26 °C (mmHg)	46
3. Density (lb/gal at 26 °C)	6.5
4. Viscosity (cps) at 30 °C	3.3
5. Surface tension (dynes/cm) at 25 °C	19.6
6. Refractive index at 25 °C	1.38
7. Solubility parameter at 26 °C	
Hansen D	7.2
Hansen P	2.6
Hansen H	7.1
Total Hansen	10.4
8. Solubility, in H_2O at 26 °C	Complete

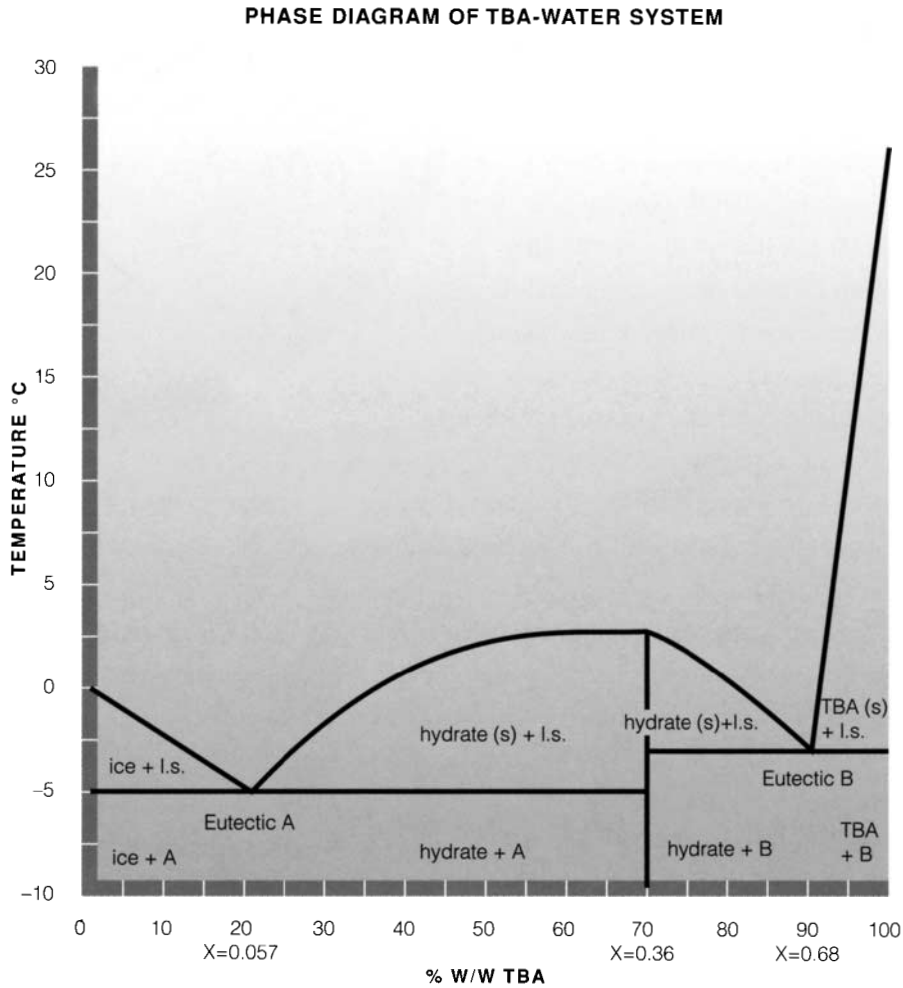


FIG. D-1 Phase diagram for TBA-water system. (l.s. = liq. state.) (Kasraian, K., DeLuca, P., *Pharmaceutical Research*, Vol. 12, No. 4, 1995; permitted by the Plenum Publishing Corporation.)

Frozen TBA-water mixtures. Research using freeze-drying microscopy has shown that TBA affects the crystal habit of ice and therefore the sublimation rate. Adding 3–19 percent TBA resulted in the formation of large needle-shaped ice crystal patterns that can facilitate sublimation. Once these crystals sublime, they leave behind a more porous and lower resistance dry matrix than water alone. Drying can take place more effectively through this matrix. Figure D-5 shows frozen TBA-water mixtures with TBA concentrations varying from 0 to 70 percent w/w. As TBA concentrations increase, the crystal patterns become more ordered and needle-shaped.

Porous resistance of the dry product layer. Research has shown that adding 5% w/v TBA into a 5 percent w/v sucrose solution considerably shortens the primary drying stage by lowering the resistance of the dried cake. Figure D-6 shows that the frozen solution without TBA initially had a high resistance, approximately 60 cm^2 torr hr/gm, due to the formation of a skin. Once the skin cracked, the resistance improved to 10 cm^2 torr hr/gm. The solution containing TBA had a resistance of 0.5–3 cm^2 torr hr/gm. Figure D-7 shows the result. Without TBA, the sucrose solution dried in 100 hours. Adding 5 percent w/v TBA lowered the drying time to 10 hours.

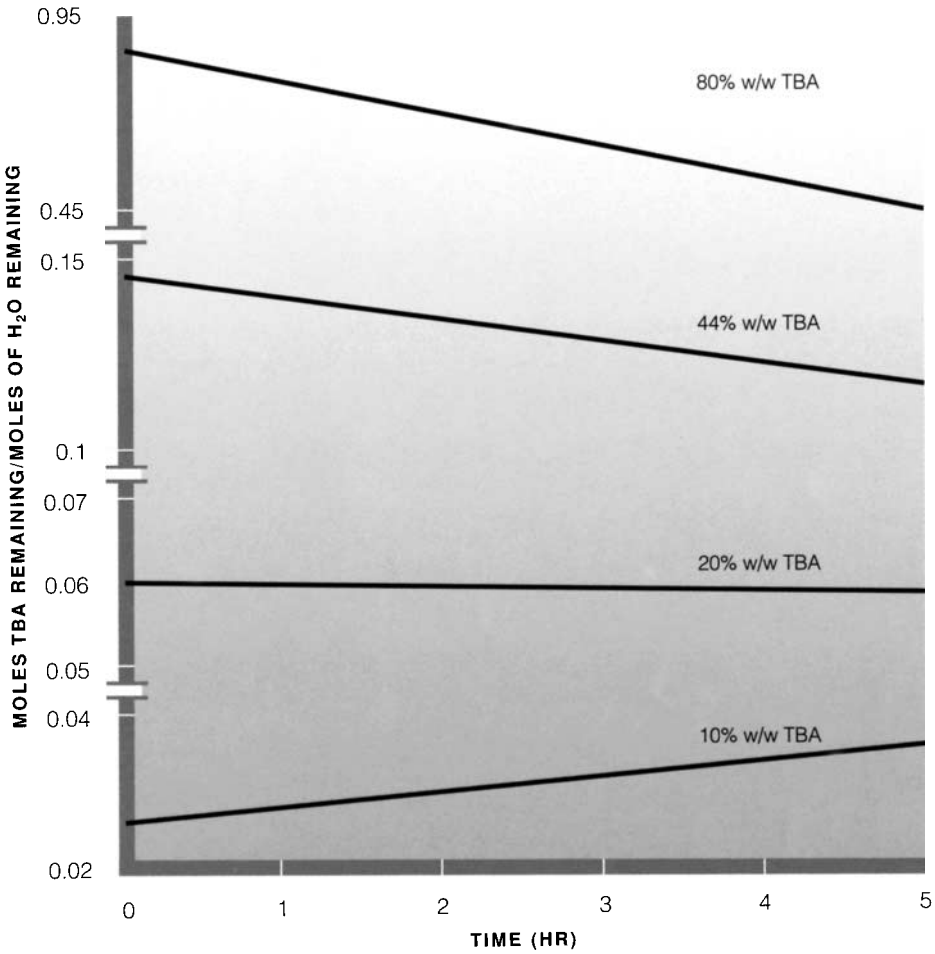


FIG. D-2 Ratio of TBA to water as a function of time for 10%, 20%, 44%, and 80% TBA solutions. (Source: Kasraian, K., DeLuca, P., *Pharmaceutical Research*, Vol. 12, No. 4, 1995; permitted by the Plenum Publishing Corporation.)



FIG. D-3 Suitable clothing for operator handling. (Source: ARCO Chemical.)



FIG. D-4 Suitable clothing for operator handling. (Source: ARCO Chemical.)

Applications

TBA as mass transfer accelerator. Beecham Pharmaceuticals has extensively studied the effect of organic solvents, especially TBA, on freeze-drying efficiency and product properties. (See Figs. D-8 and D-9.)

The use of TBA for freeze drying the common antibiotic gentamicin, in the presence of maltose, has been reported. Adding TBA reduced the drying time from 39 hours to 28 hours and maintained the porous structure of the product.

TBA in biopharmaceuticals. The industry uses sugar and other polyhydroxy compounds as stabilizers for proteins and biological materials in their formulations. Sugars are added to solutions for freeze drying to protect certain protein compounds from freeze, freeze-thaw, and freeze-drying damage. However, freeze-drying cycles of such solutions are excessively long because sugar solutions collapse at very low temperatures. Consequently, low shelf temperatures must be maintained throughout the drying stage.

The effect of adding 5 percent cosolvent on the freeze drying of sucrose and lactose has been extensively studied. Table D-2 compares the freeze-drying performance of organic solvents with water. All solutions, except the TBA ones, failed to freeze dry. The solvent systems without TBA experienced severe bubbling of the cosolvents followed by collapse during the drying phase. Solutions containing TBA resulted in complete drying and yielded good cakes.

The freeze-drying behavior of sugar solutions at various temperatures has been researched. It was shown that adding 5–10 percent w/v TBA increased the drying rate by 3 times (Table D-3). In addition, only the TBA-containing solutions survived at a 30°C shelf temperature.

The effects of using TBA on the properties of the dried sucrose (Table D-4) has been compared. Data indicate that the cake dried from the TBA solution was very porous. The hypothesis is that the porous nature causes reduced resistance to water vapor transfer during sublimation and subsequently gives faster drying rates.

Figures D-10 and D-11 provide scanning electron microscopy (SEM) and photographs of sucrose solutions during freeze drying without and with TBA. Collapse occurs in the samples without TBA. Therefore TBA appears to either

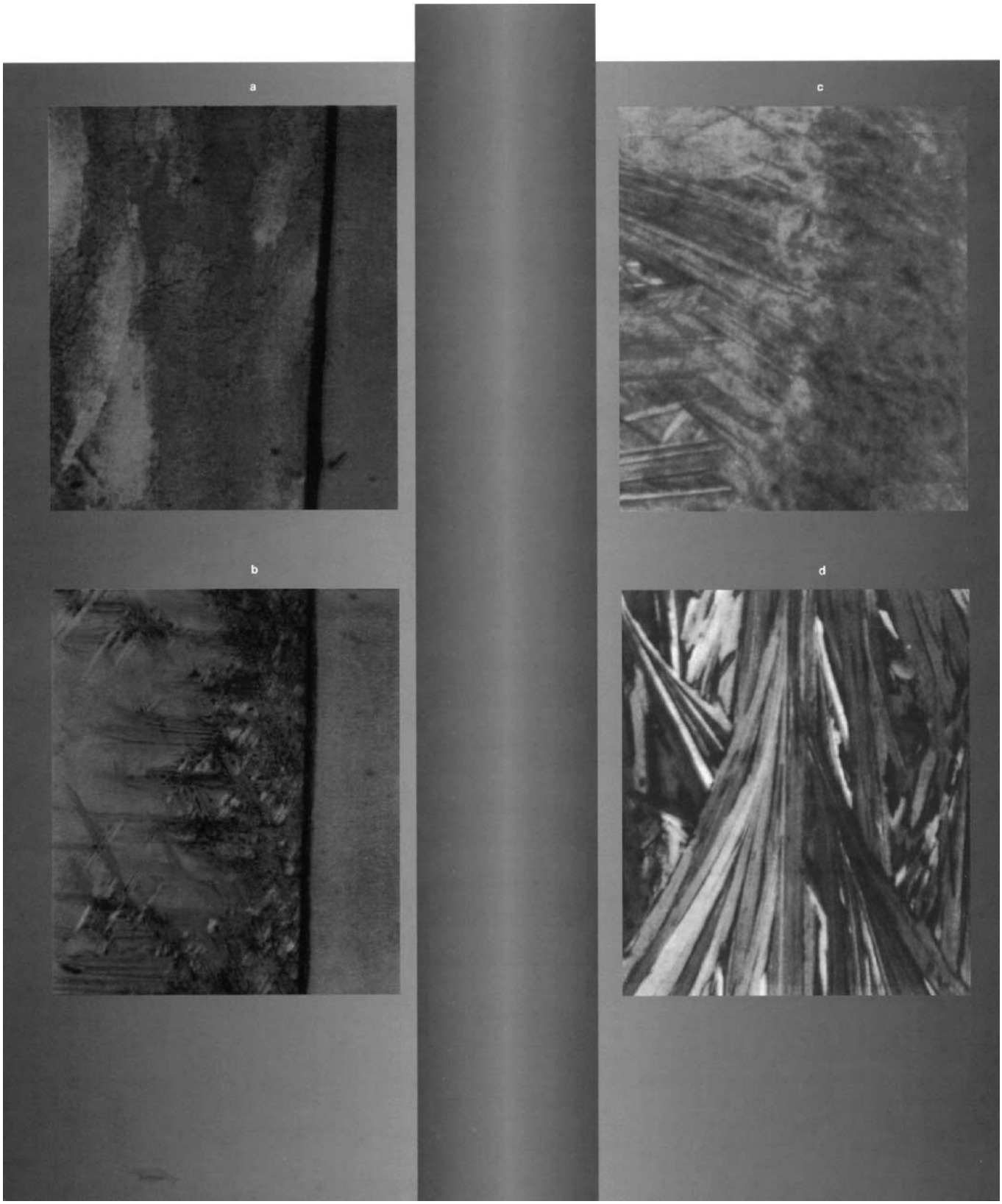


FIG. D-5 Polaroid photographs of frozen TBA-water mixtures with different concentrations of TBA: (*a*) frozen deionized water, (*b*) 10% w/w TBA aqueous solution, (*c*) 50% w/w TBA aqueous solution, (*d*) 70% w/w TBA aqueous solution. (Source: Kasraian, K., DeLuca, P., *Pharmaceutical Research*, Vol. 12, No. 4, 1995; permitted by the Plenum Publishing Corporation.)

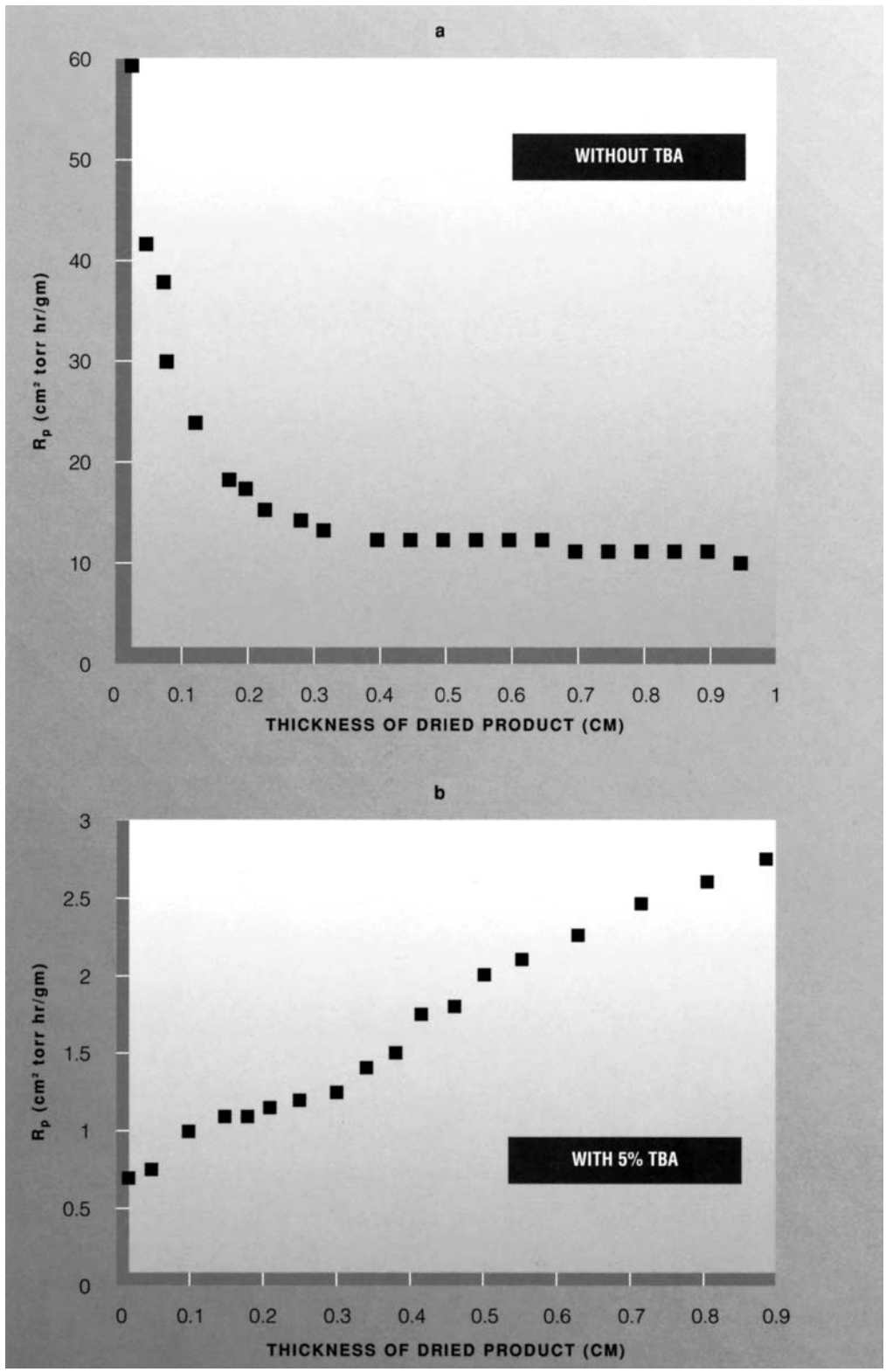


FIG. D-6 Normalized dried product resistance versus thickness of dried product: (*a*) 5% w/v sucrose freeze dried in a microbalance at a temperature of −35 °C, (*b*) 5% w/v sucrose containing 5% w/v TBA freeze dried in a microbalance at −35 °C. (Source: Kastaian, K., DeLuca, P., *Pharmaceutical Research*, Vol. 12, No. 4, 1995; permitted by the Plenum Publishing Corporation.)

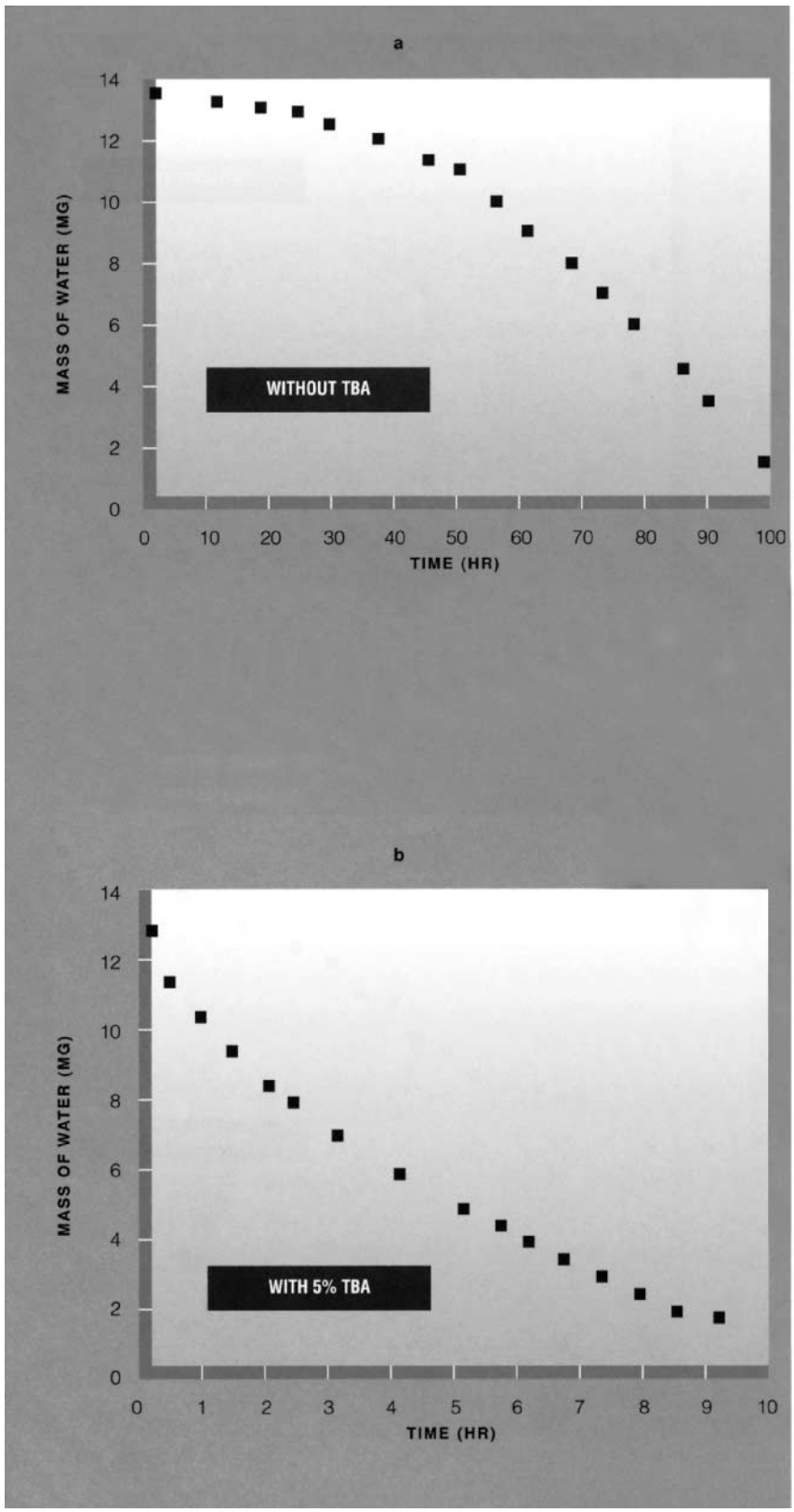




FIG. D-7 Mass loss of water from: (*a*) 5% w/v sucrose solution, (*b*) 5% w/v sucrose solution containing 5% w/v TBA. Drying temperature −35 °C in the microbalance. (Source: Kasraian, K., DeLuca, P., *Pharmaceutical Research*, Vol. 12, No. 4, 1995; permitted by the Plenum Publishing Corporation.)



FIG. D-8 A vacuum dryer. (Source: Stokes Vacuum Inc.)

elevate the collapse temperature or prevent the frozen product from reaching the collapse temperature due to faster rate of sublimation.

AKZO Pharma Division of Organon International B.V. also reported how TBA concentrations affect the stability of freeze-dried sucrose formulations. Adding 5 percent TBA to a 180 mg/ml sucrose solution resulted in a pharmaceutically acceptable, stable freeze-dried cake with no collapse. Therefore, while TBA addition did not change the collapse temperature of the sucrose solution, it increased the rate of sublimation. The increase thereby prevented the product from ever rising to the collapse temperature.

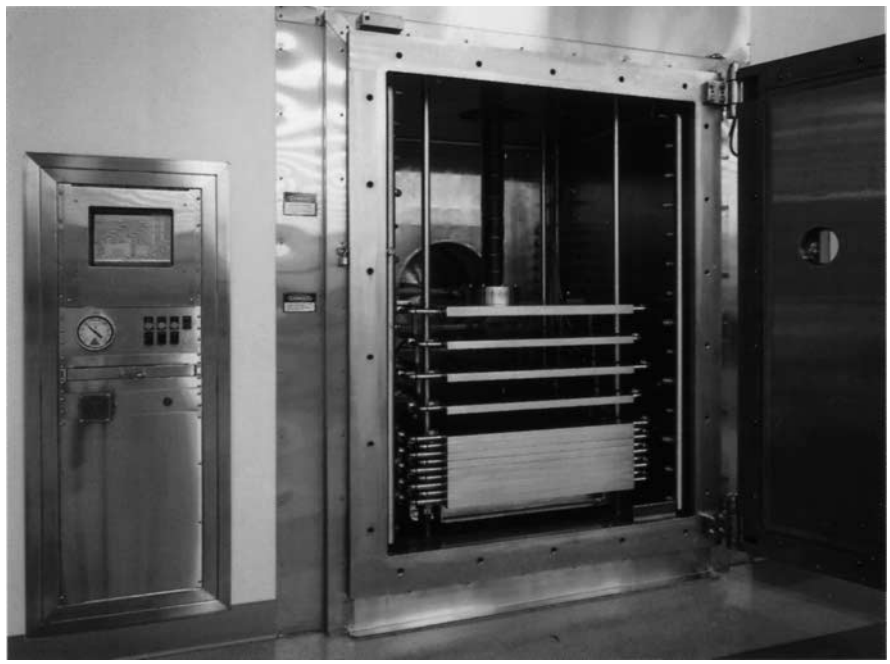


FIG. D-9 A vacuum dryer. (Source: Stokes Vacuum Inc.)

TABLE D-2 Effects of Solvent on Freeze-Drying Sugar Solutions

System	Shelf Temperature (°)	Remarks
Sucrose alone	0	Collapse
Sucrose + methanol	0	Boiling of solvent, collapse
Sucrose + ethanol	0	Boiling of solvent, collapse
Sucrose + isopropanol	0	Boiling of solvent, collapse
Sucrose + acetone	0	Boiling of solvent, collapse
Sucrose + *n*-butanol	0	Boiling of solvent, collapse
Sucrose + dioxane	0	Boiling of solvent, collapse
Sucrose + TBA	30, 45	Good
Lactose alone	0	Collapse
Lactose + methanol	0	Boiling of solvent, collapse
Lactose + ethanol	0	Boiling of solvent, collapse
Lactose + isopropanol	0	Boiling of solvent, collapse
Lactose + acetone	0	Boiling of solvent, collapse
Lactose + n-butanol	0	Boiling of solvent, collapse
Lactose + dioxane	0	Boiling of solvent, collapse
Lactose + TBA	30, 45	Good

Taken from DeLuca, P. P., Kamat, M. S., Koida, Y. Congr. Int. Technol. Pharm., 5th, 1989, 1, 439, permitted by the publisher, Rue J.-B. Clement.

TBA in freeze drying of lipids and liposomes. Use of liposomal formulations is rapidly gaining popularity in pharmaceutical research and development. Liposomes are increasingly serving as carriers for antigens and/or drugs for different routes of administration. The physical stability of the liposomes during long-term storage has been a matter of intense investigation for some time. To ensure its therapeutic properties, a liposomal dosage form must be stable with respect to its drug-carrier characteristics. While several approaches can stabilize liposomes, lyophilization is one of the best available methods to extend shelf-life.

TABLE D-3 Observations during Freeze-Drying Sugar Solutions (10% w/w)

	Freezing Pattern	Collapse Temperature, °C	Shelf Temperature, °C	Product Temperature, °C	Drying Rate, g/hr
Sucrose alone	Spontaneous	–28	–30	–31	0.21
Sucrose + 5% TBA	Slow, needles	–21	+30	–28	0.60
Sucrose + 10% TBA	Slow, needles	–20	+30	–30	0.67
Lactose alone	Spontaneous	–23	–15	–27	0.19
Lactose + 5% TBA	Slow, needles	–22	+30	–26	0.70
Lactose + 10% TBA	Slow, needles	–21	+30	–27	0.79

Taken from DeLuca, P. P., Kamat, M. S., Koida, Y. Congr. Int. Technol. Pharm., 5th, 1989, 1, 439, permitted by the publisher, Rue J.-B. Clement.

TABLE D-4 Effects of TBA on Properties of Dried Sucrose

	Without TBA	With TBA
Initial drying	Slow	Fast
Texture	Coarse	Smooth, soft
Particle shape	Irregular	Irregular
SEM	Plate-like	Plates but porous
Polarized light	Non-birefringent	Partial birefringence
Crystallinity	Amorphous	Partial crystallinity
Residual moisture	1–2%	1–2%
Surface area	0.9 m^2/g	1% TBA = 1.13 m^2/g 5% TBA = 2.25 m^2/g 10% TBA = 2.80 m^2/g
Residual TBA		<0.2%
Total drying time	40 hr	22 hr

Taken from DeLuca, P. P., Kamat, M. S., Koida, Y. Congr. Int. Technol. Pharm., 5th, 1989, 1, 439, permitted by the publisher, Rue J.-B. Clement.

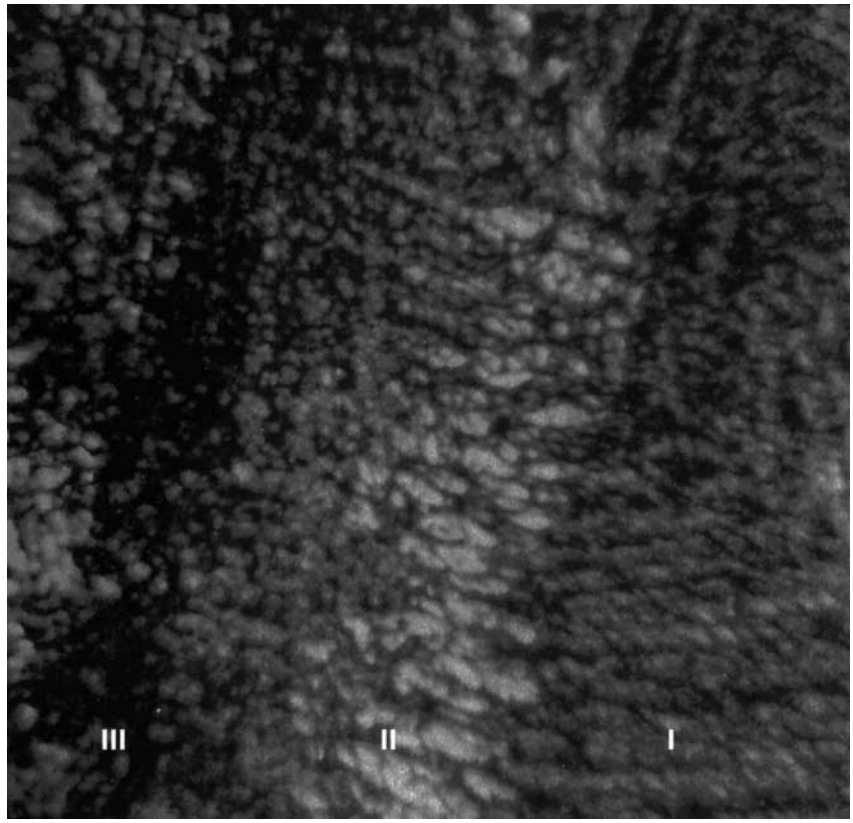
Because water is an essential component of the liposomal bilayer structure, freezing can promote damage due to dehydration. Use of cryoprotectants and control of the freezing rate that can minimize the formation of large ice crystals are of utmost importance. To achieve stability, the quality of phospholipids and the production process must be reproducible.

Research has shown that lipid solubility is four to five times greater in TBA than in other organic solvents such as ethanol. Therefore, more researchers are considering TBA as a lyophilization solvent for dissolution of lipids.

Ciba-Geigy Ltd., reported using TBA and *N*-methyl pyrrolidone (NMP) as water miscible organic solvents in large-scale production of liposomes. TBA was selected to dissolve the phospholipids and NMP to dissolve the dye, zinc phthalocyanine. This organic phase was mixed with an excess of a water phase to yield reproducible unilamellar liposomes with a mean size of 50–150 nm. The liposomes were then sterile filtered and freeze dried in a mixture of lactose and phospholipid. Three batches were tested for particle size, monomeric ZnPc, residual organic solvent, and moisture content in the lyophilized samples. Particle size was comparable after every manufacturing step with all three batches. The fraction of monomeric ZnPc was 100 percent in all three batches after every manufacturing step. Also, the removal of organic solvent and moisture content were reproducible.

Shionogi & Co., Ltd., reported a process for manufacturing a crystalline, lyophilized formulation of fosfomycin sodium (FOS) using aqueous TBA. FOS has an extremely high affinity with water, and the eutectic point is below –40 °C. An aqueous solution of FOS cannot be frozen at the temperatures obtained in common

(a)



(b)

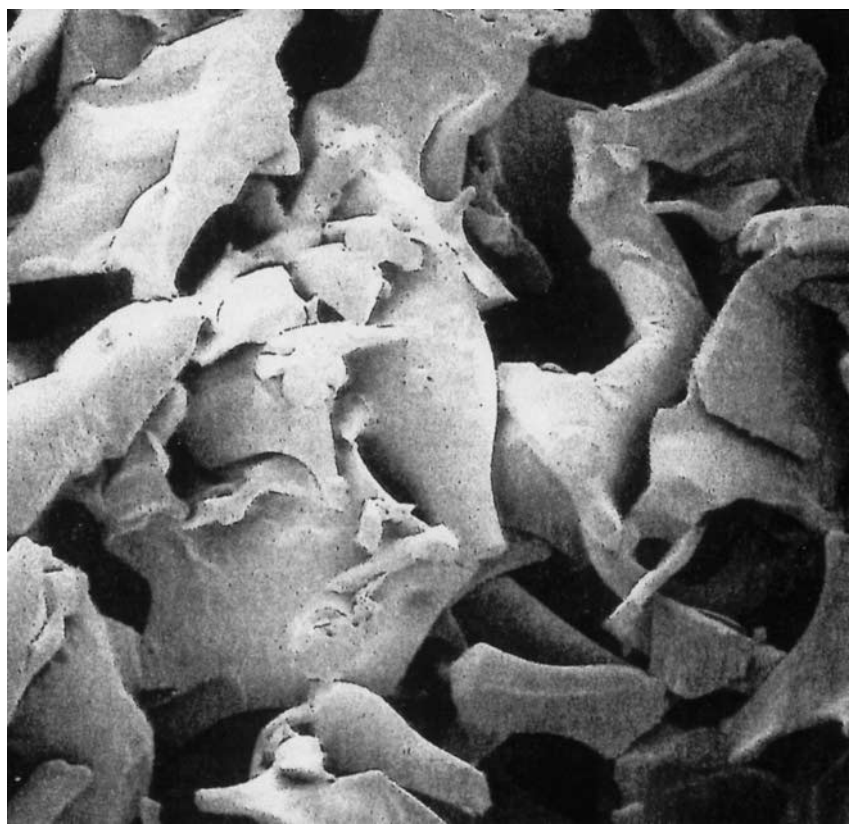
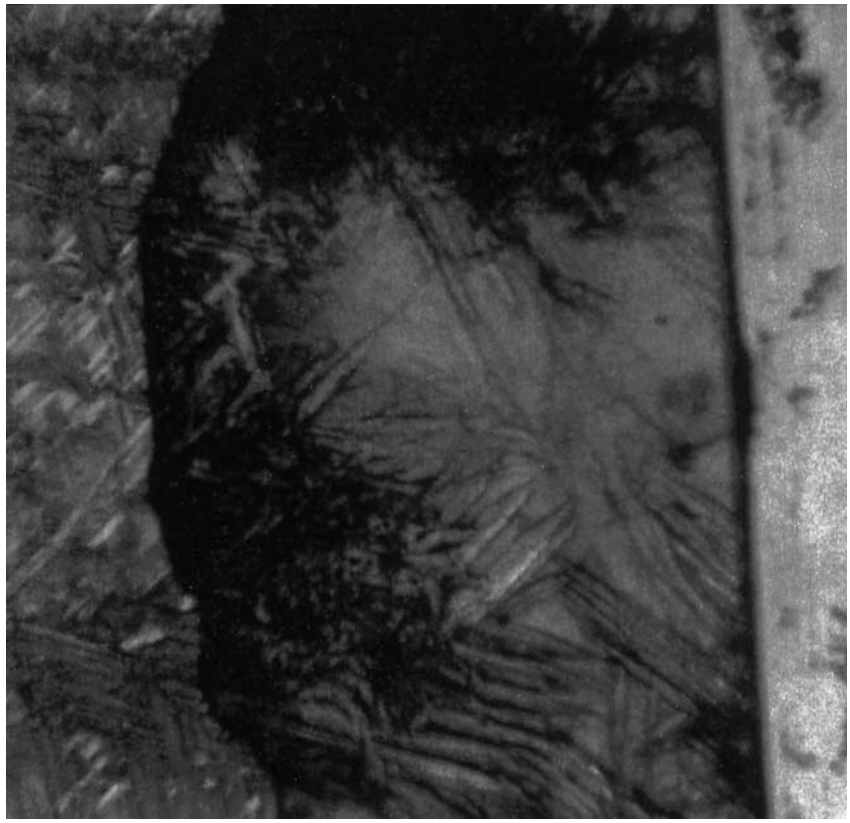


FIG. D-10 (*a*) Polaroid photographs of sucrose during freeze drying without TBA. Region I: dried material. Region II: collapse. Region III: frozen matrix. (*b*) SEM of freeze-dried sucrose (10% w/v). (Source: Kasraian, K., DeLuca, P., *Pharmaceutical Research*, Vol. 12, No. 4, 1995; permitted by the Plenum Publishing Corporation.)

(**a**)



(**b**)



FIG. D-11 (*a*) Polaroid photographs of sucrose during freeze drying in the presence of TBA. (*b*) SEM of sucrose (10% w/v) freeze dried with TBA. (Source: Kasraian, K., DeLuca, P., *Pharmaceutical Research*, Vol. 12, No. 4, 1995; permitted by the Plenum Publishing Corporation.)

freeze-drying devices. Moreover, the sample immediately melts during primary drying. For these reasons, no such formulations are presently available. Shionogi & Co., Ltd., developed a simple manufacturing method that produces a stable product with a long shelf-life. They have obtained the target formulations by dissolving FOS in aqueous TBA and then freeze drying. The TBA allows FOS to be freeze-dried at a temperature far higher than the original eutectic point. Therefore, ordinary freeze-drying equipment and operation can be used.

Sumitomo Pharmaceuticals Co., Ltd., reported a method for preparing a lyophilized formulation of a liposoluble platinum (II) complex. Liposoluble platinum (II) is virtually insoluble in water, but very soluble in TBA. Therefore, researchers dissolved the liposoluble complex in TBA and then lyophilized it for use as an anticancer drug. Analysis revealed that the resulting TBA content was no more than 0.05 wt%.

Many more patents and publications describe the use of TBA for dissolving liposomes. Geo-Centers, Inc., has patented a process for fabricating lipid microstructures using TBA where the dissolved lipid grows into tubular microstructures. Mehta et al. reported using TBA to lyophilize antifungal polyene macrolide-containing liposomes.

Freeze drying of water unstable drugs. The Upjohn Company reported a process to manufacture a stable, lyophilized formulation of prostaglandin E1 (PGE-1) for use in the treatment of erectile dysfunction. Lyophilization of a buffered lactose formulation of PGE-1 from a TBA/water mixture provides superior product stability than when freeze drying from a 100 percent aqueous system. The level of TBA that afforded the product maximum stability appeared to be when the TBA amount ranged from 17–25 percent (v/v). The unique kinetics of the degradation pathway of PGE-1 indicates that it is imperative to keep PGE-1 molecules as far apart as possible in order to minimize the interaction of two PGE-1 molecules. TBA is most likely enabling the PGE-1 molecules to be kept further apart during the freezing and lyophilization phases of manufacture.

Bristol-Myers Company has reported on the use of TBA as a solvent for the in-vial deposition of 7 (dimethylaminomethylene) amino-9a-methoxymitosane in sterile unit dosage form. This compound is not stable in water. It is introduced into a sterile vial in a TBA solution. Then the TBA is removed by lyophilization. The deposited material contains up to 0.5 mole equivalent of TBA as a hemi-solvate and is very stable to heat.

Miscellaneous applications. United States Surgical Corporation patented a process for preparing foamed, bioabsorbable polymer particles by freeze drying. The particles are useful in medical diagnostic procedures such as mammography and in the repair of damaged or defective bone. The use of TBA or other organic solvents enables the manufacturing process to achieve low processing temperatures that allow medicinals, drugs, growth factors, radiopaque substances, and other additives to be incorporated into the foamed polymer. These additives cannot tolerate high processing temperatures. The bioabsorbable polymer particles serve as excellent vehicles for the delivery of drugs, growth factors, and other biologically active substances to surrounding bone or tissue.

Sterling Drug Inc. patented TBA as a drug dispersion medium for surface modified drug nanoparticles. They claim the use of TBA as a dispersion medium for pharmaceutical drugs having a water solubility of less than 10 mg/mL. The excellent dispersion provides pharmaceutical compositions with unexpectedly high bioavailability.

DeLuca reported that a macromonomer solution with TBA was easier to sterilize by filtration and fill since it was free of foaming compared to the water solution.

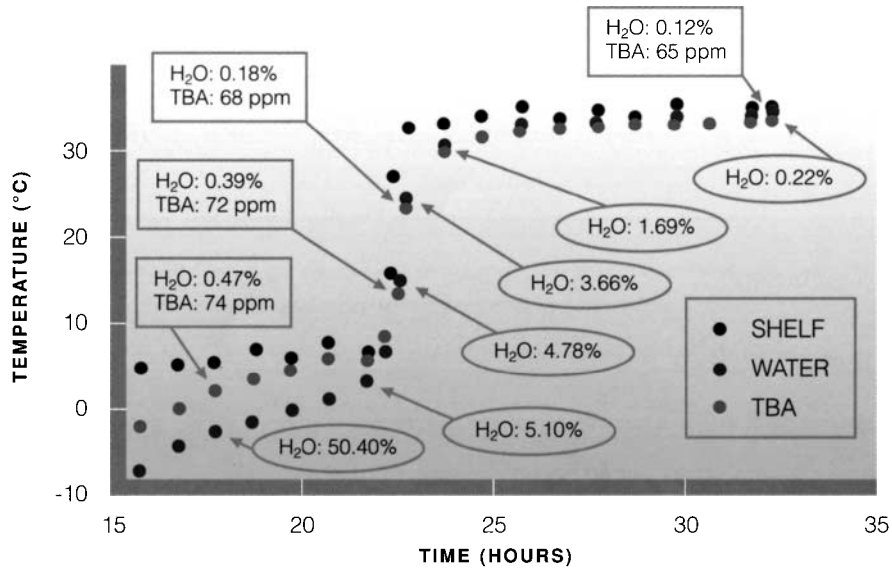


FIG. D-12 Temperature-time profile for freeze-drying cycle of macromonomer. (Source: P. DeLuca, PharmTech Conference Proceeding, 1994, p. 375. Copyright by Advanstar Communications, Inc.)

Figure D-12 illustrates the temperature profile for the samples and the water content at various stages of drying. During the primary drying stage, the TBA solution remained at a lower temperature showing faster drying and the temperature increased after 13 hours showing evidence for lower water content. After 17.5 hours of cycle time, the TBA solution sample reached 1 °C while the water sample remained at 4 °C. The freeze-dried material with TBA showed very low moisture content (0.12 percent) compared to the material freeze dried in water that showed 0.22 percent moisture. The residual TBA was 65 ppm.

Schott Glaswerke has a patent on using TBA to prepare a high purity glass powder with a mean particle size of less than 10 μm. Glass powders having a particle size up to 300 μm are ground to the desired particle size in the presence of a grinding liquid comprising water and TBA. The slurry is then frozen, and the solvent is subsequently removed from the frozen slurry by freeze drying. The resultant glass powder is particularly suitable as a filler for synthetic resins in the dental sector.

Ducting; Ducting and Joints (*see also* Expansion Joints)

Ducting, such as that provided with another major accessory—a gas turbine intake filter system, for example—may be provided by the vendor of the major accessory. If it refers to the gas passageway from the exhaust end of a gas turbine to an HRSG (*see* Cogeneration), the entire package is likely to be provided by the gas turbine vendor. At any rate, ducting of major consequence is generally custom designed for a plant. If well designed in terms of supports and seals and if not subject to fluctuating temperatures, it could well remain a low-maintenance item through the life of a plant. Expansion joints, however, are often subjected to fluctuating temperatures.

E

ECM (Engine Condition Monitoring) (*see* Condition Monitoring)

ECMS (Engine Condition–Monitoring Systems) (*see* Condition Monitoring)

Ecological Parks; Industrial Ecological Parks*

Industrial ecology, based on recycling of waste products, is a sustainable development strategy that is gaining ground. The reductions in overall industrial and commercial fuel use gained by sharing energy among a group of neighboring facilities contributes to resource sustainability. Another aspect of the ecological park concept is waste-heat recovery and energy from waste projects. There is also the potential for many companies to cooperate to reduce waste-management costs by reassessing industrial processes to recycle many liquid and solid waste streams. Many air- and waste-management issues can be dealt with using pollution prevention technology.

A successful ecological park is much easier to integrate into local communities that might otherwise complain about waste, smoke, pollution, and so forth. To succeed, therefore, a high level of public awareness, information, and support is required. Governments, local or otherwise, can assist by being educated on the basics of the technology and providing financial incentives.

Ecosystem

When operating a plant in certain countries such as the Scandinavian countries, the public and government are well educated in environmental issues. In countries such as Canada, the level of the general population's environmental education may not be as consistent, but it is likely to be higher than in the U.S., for instance. Penalties for damage to ecosystems may reflect a higher percentage of revenues in ecologically aware countries.

Ecosystem Approach*

Table E-1 outlines various operational definitions of an ecosystem, and Table E-2 defines various ecosystem approaches. When these are reviewed it becomes apparent that, despite minor differences in detail and wording, they all encompass physical, biological, and chemical properties while focusing on air, water, soil, and biota. In response to decision-making needs and concerns, the classical ecological definitions have been expanded to include specific reference to human beings as an integral part of the biological community and the flexible nature of ecosystem spatial boundaries.

There are many advantages of the ecosystem approach including:

- Focus is on the interrelationships among ecosystem components, which encourages integrated management of these components

---

* Source: Environment Canada. Adapted with permission.

TABLE E-1 A Selection of Definitions of an Ecosystem

". . . a community of organisms and their nonliving environment. Fundamental to the system is the flow of energy via food chains and the cycling of nutrients."
". . . subdivisions of the global ecosphere, vertical chunks that include air, soil, or sediments, and organisms (including humans). Ecosystems occur at various scales, from the global ecosphere to continents and oceans, to ecoregions, to forest, farms, and ponds."
". . . an assemblage of biological communities (including people) in a shared environment. Air, land, water and the living organisms among them interact to form an ecosystem."
". . . a community of organisms, including humans, interacting with one another, plus the environment in which they live and with which they interact. Ecosystems are often embedded within other ecosystems of larger scale."

TABLE E-2 A Selection of Definitions of an Ecosystem Approach

". . . an approach to perceiving, managing, and otherwise living in an ecosystem that recognizes the need to preserve the ecosystem's biochemical pathways upon which the welfare of all life depends in the context of multifaceted relationships (biological, social, economic, etc.) that distinguish that particular ecosystem."
". . . means looking at the basic components (air, water, and biota, including humans) and functions of the ecosystem not in isolation, but in broad and integrated environmental, social and economic context."
". . . a geographically comprehensive approach to environmental planning and management that recognizes the interrelated nature of environmental media and that humans are a key component of ecological systems; it places equal emphasis on concerns related to the environment, the economy, and the community."

- Focus is on long-term and/or large-scale issues, which permits a more "anticipate and prevent" strategy to management, rather than the more common "react and cure" mode
- Role of culture, values, and socioeconomic systems in environmental and resource management issues is recognized
- A mechanism is offered for integrating science and management

Ecosystem Approach to Management

The ecosystem approach to management or ecosystem-based management has been described as a planning and management tool that provides a framework for observing and interpreting nature as well as managing human uses and abuses of nature. This approach recognizes that it is human interactions with ecosystems, not the ecosystems themselves, that must be managed. In other words, it is the sustainable management of human uses of the natural resources within a multiple-use system. (See Tables E-3 and E-4.)

Many process engineers who might have worked in countries with limited environmental legislation have been unpleasantly surprised when they find themselves working in a country with stricter laws in that regard.

Ejectors

Ejectors are a means of optimizing the value of a vacuum condition. A high-energy fluid stream imparts a higher-pressure energy to a fluid of lower-pressure energy

TABLE E-3 A Selection of Definitions of an Ecosystem Approach to Management

". . . requires a broad perspective. It includes knowledge of heritage resources, ecological processes and socioeconomic activities . . . ecosystem-based management must, above all, be sensitive and responsive to the unique status of each ecosystem and its spheres of influence."
". . . is an active process that emphasizes the maintenance of biological diversity, of natural relationships among species, and dynamic processes that make ecosystems sustainable."
"The application of biophysical and social information, options, and constraints to achieve desired social benefits within a defined geographic area and over a specified time period."
". . . recognizes there are ecological, social, and economic considerations to be made when assessing and predicting the impacts of human activities on natural systems and practicing the 'ecosystem approach' means that all stakeholders understand the implications of, and are accountable for, their actions."
". . . implies a balanced approach toward managing human activities to ensure that the living and nonliving elements that shape ecosystems continue to function and so maintain the integrity of the whole."

TABLE E-4 Comparison of Four Approaches to Resolving Human-Made Ecosystem Problems

	Approach			
Problem	Egosystemic	Piecemeal	Environmental	Ecosystemic
Organic waste	Hold your nose	Discharge downstream	Reduce BOD	Energy recovery
Eutrophication	Mysterious causes	Discharge downstream	Phosphorus removal	Nutrient recycling
Acid rain	Unaware	Not yet a problem	Taller smokestacks	Recycle sulfur
Toxic chemicals	Unaware	Not yet a problem	Discharge permits	Design with nature
Greenhouse effects	Unaware	Not yet a problem	Sceptical analysis	Carbon recycling
Pests	Run for your life	Broad spectrum insecticides	Selective degradable poisons	Integrated pest management
Attitude to nature	Indifferent	Dominate	Cost/benefit	Respect

SOURCE: Environment Canada–U.S. Environmental Protection Agency, International Joint Commission, 1995.

via a nozzle. Use of ejectors is particularly common in the agricultural and food industries for processes such as humidification, fumigation, impregnation, cooling, freeze drying, and vacuum drying.

Electric Motors; Electric Motor Controls*

Motor selection is a complex process involving many trade-offs with parameters that include efficiency. The objective of optimum motor selection is to arrive at the best compromise of cost, horsepower, and frame size for the life expectancy, load torque, load inertia, and duty cycle in question.

To fulfill the requirements of a large range of applications, NEMA specifies polyphase AC motors in four different classes, A through D. Each has its own speed torque characteristic (see Fig. E-1).

Motors intended for effectively constant loads and long run times are designed with low slip (less than 5 percent) and are more efficient than design D motors. The latter are used where loads are heavy and sudden, such as hoists and cranes. Design D motors deliver high starting torque and are designed with high slip (greater than

*Source: Reliance Electric, USA. Adapted with permission.

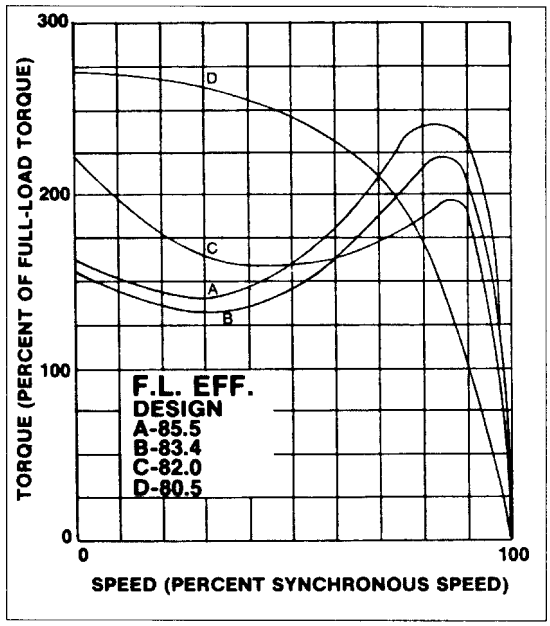


FIG. E-1 Speed-torque curves for a 5-hp motor, NEMA design A and D, and full-load efficiencies. (Source: Reliance Electric.)

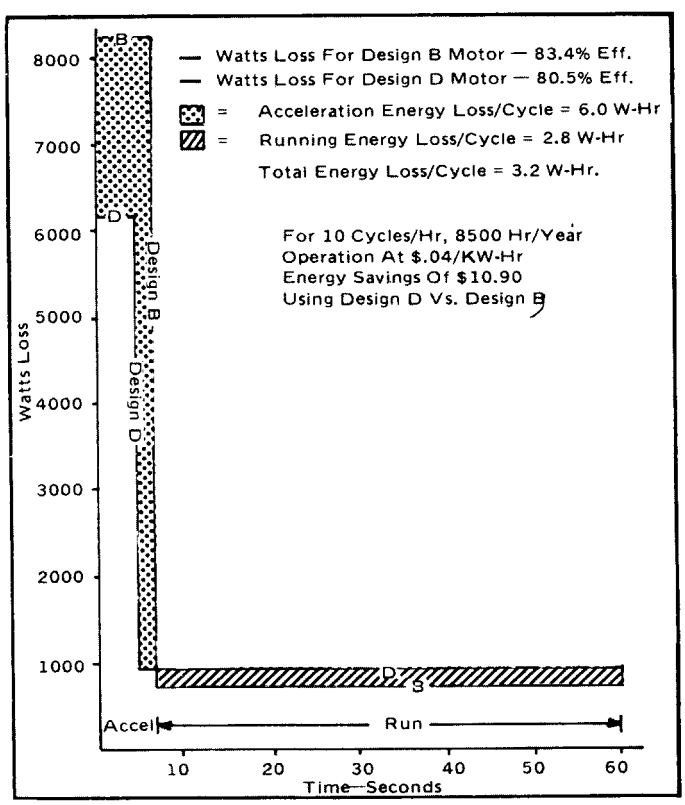


FIG. E-2 Energy usage on duty cycle application 5-hp, 4-pole, TEFC accelerating 27 lb·ft^2 inertia. (Source: Reliance Electric.)

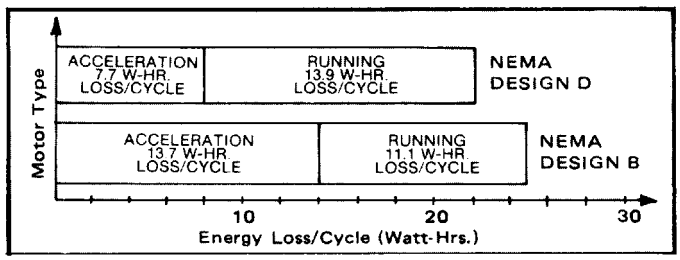


FIG. E-3 Acceleration and running loss per cycle on NEMA D and NEMA B motors. (Source: Reliance Electric.)

5 percent) so that motor speed can drop when fluctuating loads are encountered. Although design D motor efficiency can be less than other NEMA designs, it is not possible to replace a design D motor with a more efficient design B motor, because it would not meet the performance demands of the load.

The motor with the highest operating efficiency does not always provide the lowest energy choice. Figure E-2 compares the watts loss of a NEMA design D and a design B motor, in a duty cycle that accelerates a load inertia of 27 lb·ft^2 to full speed and runs at full load for 60 s. During acceleration, the lower curve represents the performance of the design D motor, while the upper curve reflects the NEMA design B motor. The shaded area between the curves represents the total energy difference during acceleration. In this example, this area is approximately 6.0 watt-hours, the energy saved accelerating this load with a design D motor instead of a design B. During the run portion of this duty cycle, the energy loss differential favors the NEMA design B, because it has a higher operating efficiency. In this example, the energy saved operating this load with a design B motor instead of a design D motor is approximately 2.8 watt-hours.

The bar chart shown in Fig. E-3 summarizes acceleration and running loss/cycle on both the NEMA design B and design D. Comparison of the total combined acceleration and running portions of this duty cycle indicates a total energy savings of 3.2 watt-hours favoring the use of the design D motor, even though the design B motor has an improved operating efficiency. The key is the improved ability of the design D motor to accelerate a load inertia at minimum energy cost.

Components Affecting Efficiency

Because a motor buyer selects the most efficient motor of a given size and type does not mean that energy savings are being optimized. Every motor is connected to some form of driven equipment: a crane, a machine tool, a pump, etc., and motors are often connected to their loads through gears, belts, or slip couplings. By examining the total system efficiency, the component that offers the greatest potential improvements can be identified and money allocated to the component offering the greatest payback.

In the case of new equipment installations, a careful application analysis, including load and duty cycle requirements, might reveal that a $7\frac{1}{2}$-horsepower pump, for example, could be utilized in place of a 10-horsepower pump, thereby reducing motor horsepower requirements by one third. By reducing the mass of the moving parts, the energy required to accelerate the parts is also proportionately reduced. Or, in the instance of an air compressor application, the selection of size and type of compressor relative to load and duty cycle will affect system efficiency

and energy usage. Of course, the most efficient equipment should be selected whenever possible.

Reduced system efficiency and increased energy consumption are also possible with existing motor drive systems due to additional friction that can gradually develop within the driven machine. This additional friction could be caused by a buildup of dust on a fan, the wearing of parts causing misalignment of gears or belts, or insufficient lubrication in the driven machine. All of these conditions cause the driven machine to become less efficient, which causes the motor to work harder. Rather than replace the existing motor with a higher-efficiency model, replacing either critical machine components or the machine itself may result in greater system efficiency and energy savings.

Choosing the best applications

Energy-efficient motors may be the most cost-effective answer for certain applications. Simple guidelines are listed below:

- Choose applications where motor running time exceeds idle time.
- Review applications involving larger horsepower motors, where energy usage is greatest and the potential for cost savings can be significant.
- Select applications where loads are fairly constant, and where load operation is at or near the full-load point of the motor for the majority of the time.
- Consider energy-efficient motors in areas where power costs are high. In some areas, power rates can run as much as $0.12 per kilowatt-hour. In these cases, the use of an energy-efficient motor might be justified in spite of long idle times or reduced load operations.

Using these simple guidelines, followed by an analysis and cost justification based on various techniques, can yield results that will influence motor choice beyond just-in cost consideration.

Determinants of Operating Cost

Voltage unbalance

Although efficiency is a commonly used indicator of energy usage and operating costs, there are several important factors affecting motor operating costs. Rated performance as well as selection and application considerations of polyphase motors requires a balanced power supply at the motor terminals. Unbalanced voltage affects the motor's current, speed, torques, temperature rise, and efficiency. NEMA Standard MG 1-14.34 recommends derating the motor where the voltage unbalance exceeds 1 percent and recommends against motor operation where voltage unbalance exceeds 5 percent. Voltage unbalance is defined as follows:

$$\text{Voltage unbalance (\%)} = 100 \times \frac{\text{maximum voltage deviation from average voltage}}{\text{average voltage}}$$

Voltage imbalance is not directly proportional to the increase in motor losses, as a relatively small unbalance in percent will increase motor losses significantly and decrease motor efficiency as Fig. E-4 shows. An effort to reduce losses with the purchase of premium priced, premium efficiency motors that reduce losses by 20 percent can easily be offset by a voltage unbalance of 3.5 percent that increases motor losses by 20 percent.

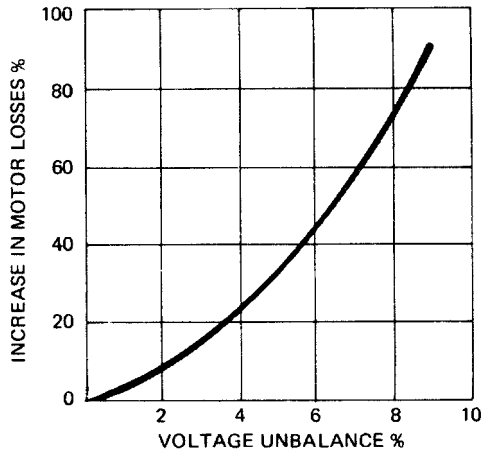


FIG. E-4 Motor loss percentage as a function of voltage unbalance. (Source: Reliance Electric.)

Energy cost can be minimized in many industrial applications by reducing the additional motor watts loss due to voltage unbalance. Uniform application of single-phase loads can ensure proper voltage balance in a plant's electrical distribution system used to supply polyphase motors.

Motor loading

One of the most common sources of motor watts loss is the result of a motor not being properly matched to its load. In general, for standard NEMA frame motors, motor efficiency reaches its maximum at a point below its full-load rating, as indicated in Fig. E-5. This efficiency peaking below full load is a result of the interaction of the fixed and variable motor losses resulting in meeting the design limits of the NEMA standard motor performance values, specifically locked rotor torque and current limits.

Power factor is load variable and increases as the motor is loaded, as Fig. E-5 shows. At increased loads, normally in the region beyond full load, this process reverses as the motor's resistance to reactive ratio begins to decrease and power factor begins to decline.

In some applications where motors run for an extended period of time at no load, energy could be saved by shutting down the motor and restarting it at the next load period.

Maintenance

Proper care of the motor will prolong its life. A basic motor maintenance program requires periodic inspection and, when encountered, the correction of unsatisfactory conditions. Among the items to be checked during inspection are lubrication, ventilation, and presence of dirt or other contaminants; alignment of motor and load; possible changing load conditions; belts, sheaves, and couplings; and tightness of hold-down bolts.

Total Energy Costs

There are three basic components of industrial power cost: cost of real power used, power factor penalties, and demand charges. To understand these three charges

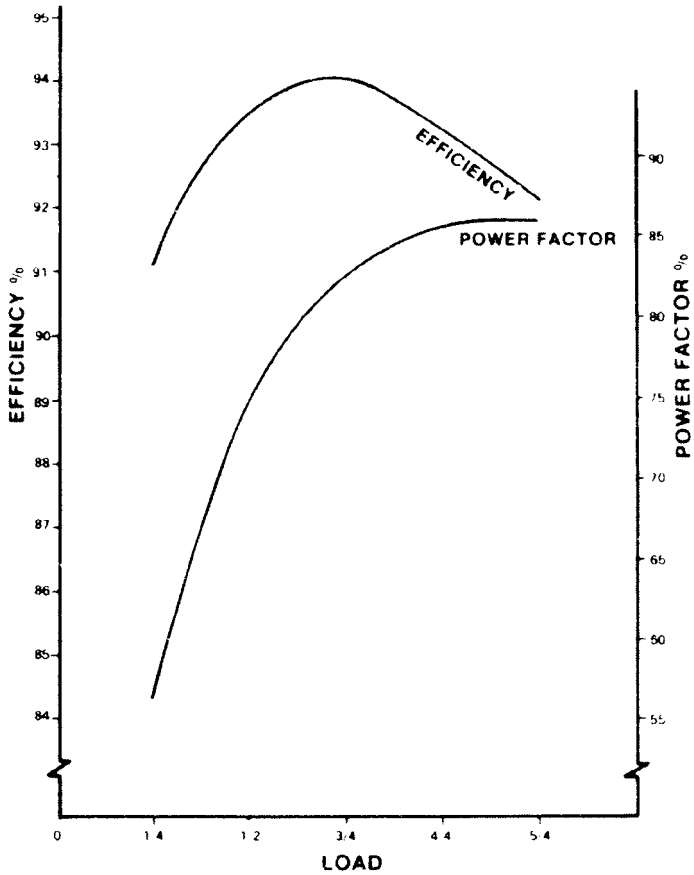
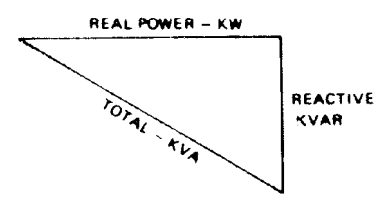


FIG. E-5 Power factor and efficiency changes as a function of motor load. (Source: Reliance Electric.)



DEFINITIONS – POWER FACTOR

KW – Kilowatts
KVA – Kilovolt – Ampere
KVAR – Reactive Kilovolt – Ampere

FIG. E-6 Electrical power vector diagram. (Source: Reliance Electric.)

and how they are determined, a review of the power vector diagram (Fig. E-6) identifies each component of electrical energy and its corresponding energy charge.

Real power

The real power-kilowatt (kW) is the energy consumed by the load. Real power-kW is measured by a watt-hour meter and is billed at a given rate ($/kW-hr). It is the real power component that performs the useful work and is affected by motor efficiency.

$$\text{Power factor} \quad \frac{\text{Real Power—KW}}{\text{Total—KVA}}$$

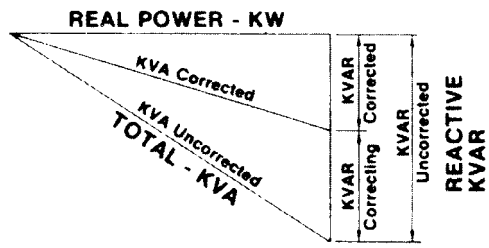


FIG. E-7 Effect of corrective capacitance on total KVA vector. (Source: Reliance Electric.)

Power factor

Power factor is the ratio of real power-kW to total KVA. Total KVA is the vector sum of the real power and reactive KVAR. Although reactive KVAR performs no actual work, an electric utility must maintain an electrical distribution system (i.e., power transformers, transmission lines, etc.) to accommodate this additional electrical energy. To recoup this cost burden, utilities may pass this cost on to industrial customers in the form of a power factor penalty for power factor below a certain value.

Power factors in industrial plants are usually low due to the inductive or reactive nature of induction motors, transformers, lighting, and certain other industrial process equipment. Low power factor is costly and requires an electric utility to transmit more total KVA than would be required with an improved power factor. Low power factor also reduces the amount of real power that a plant's electrical distribution system can handle, and increased line currents will increase losses in a plant's distribution system.

A method to improve power factor, which is typically expensive, is to use a unity or leading power factor synchronous motor or generator in the power system. A less expensive method is to connect properly sized capacitors to the motor supply line. In most cases, the use of capacitors with induction motors provides lower first cost and reduced maintenance expense. Figure E-7 graphically shows how the total KVA vector approaches the size of the real power vector as reactive KVAR is reduced by corrective capacitors. Because of power factor correction, less power need be generated and distributed to deliver the same amount of useful energy to the motor.

Just as the efficiency of an induction motor may be reduced as its load decreases, the same is true for the power factor, only at a faster rate of decline. A typical 10-horsepower, 1800 rpm, three-phase, design B motor with a full-load power factor of about 80 percent decreases to about 65 percent at half load. Therefore, it is important not to overmotor. Select the right size motor for the right job. Figure E-8 shows that the correction of power factor by the addition of capacitors not only improves the overall power factor but also minimizes the fall-off in power factor with reduced load.

Demand charges

The third energy component affecting cost is demand charge, which is based on the peak or maximum power consumed or demanded by an industrial customer during a specific time interval. Because peak power demands may require an electric utility to increase generating equipment capacity, a penalty is assessed when demand

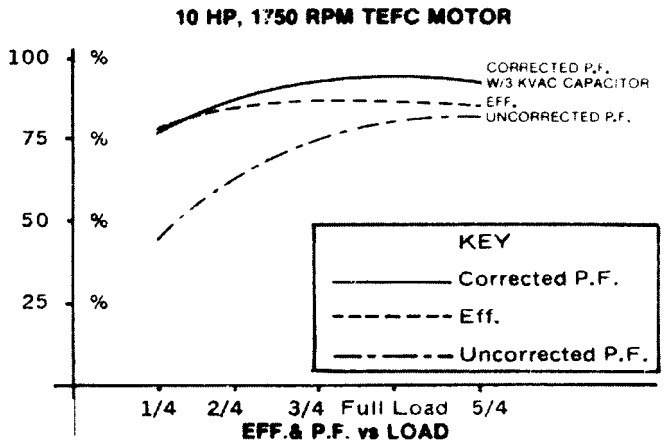


FIG. E-8 Effect of capacitors on fall-off in power factor with reduced load. (Source: Reliance Electric.)

exceeds a certain level. This energy demand is measured by a demand meter, and a multiplier is applied to the real power-kW consumed.

Industrial plants with varying load requirements may be able to affect demand charges by (1) load cycling, which entails staggering the starting and use of all electrical equipment and discontinuing use during peak power intervals, and (2) using either electrical or mechanical "soft start" hardware, which limits power inrush and permits a gradual increase in power demand.

Adjustable Frequency Motor Considerations

Speed control by way of adjusting power frequency is becoming more and more important for economical throughput or pressure capacity variation of modern process machinery. Several key parameters that must be considered when applying induction motors to adjustable frequency controllers include the load torque requirements, current requirements of the motor and the controller current rating, the effect of the controller wave-shape on the motor temperature rise, and the required speed range for the application.

In order to properly size a controller for a given application, it is necessary to define the starting torque requirements, the peak torque requirements, and the full-load torque requirements. These basic application factors require reexamination because the speed-torque characteristics of an induction motor/controller combination are different from the speed-torque characteristics of an induction motor operated on sine-wave power.

The motor current requirements should be defined for various load points at various speeds in order to ensure that the controller can provide the current required to drive the load. The current requirements are related to the torque requirements, but there are also additional considerations due to the harmonics of adjustable frequency control power that must be taken into account.

Temperature rise and speed range must be considered when applying induction motors to adjustable-frequency controllers because this nonsinusoidal power results in additional motor losses, which increase temperature rise and reduce motor insulation life.

Before discussing the speed-torque characteristics of a motor/controller combination, it is useful to review the speed-torque characteristics of an induction

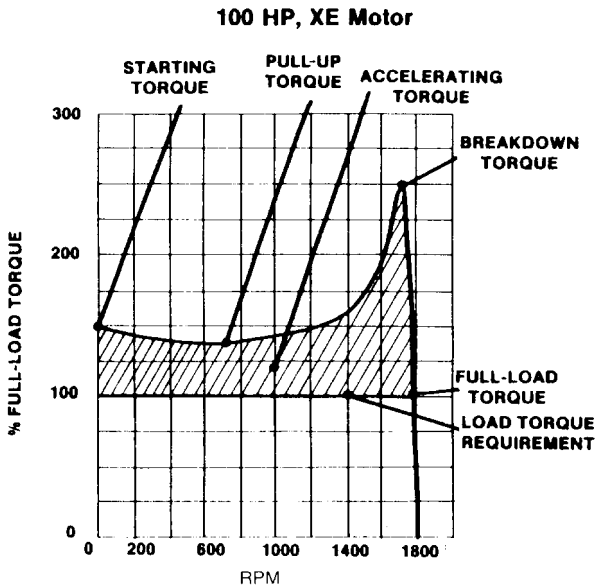


FIG. E-9 Speed-torque characteristics of induction motors started at full voltage. (Source: Reliance Electric.)

motor started at full voltage and operated on utility power (Fig. E-9). Here we see the speed-torque curve for a 100-horsepower, 1800 rpm, high-efficiency motor. When this motor is started across the line, the motor develops approximately 150 percent of full-load torque for starting and then accelerates along the speed-torque curve through the pullup torque point, through the breakdown torque point, and, finally, operates at the full-load torque point, which is determined by the intersection of the load line and the motor speed-torque curve.

In this case, we have shown an application, a conveyor where the load-torque requirement is constant from 0 rpm to approximately 1800 rpm. The difference between the motor speed-torque curve and the load line is the accelerating torque and is indicated by the cross-hatched area.

If the load-torque requirement ever exceeded the maximum torque capability of the induction motor, the motor would not have enough torque to accelerate the load and would stall. For instance, if the load line required more torque than the motor could produce at the pullup torque point, i.e., 170 percent load torque versus 140 percent pullup torque, the motor would not increase in speed past the pullup torque speed and would not be able to accelerate the load. This would cause the motor to overheat. It is, therefore, important to ensure that the motor has adequate accelerating torque to reach full speed.

Normally, the motor accelerates the load and operates at the point of intersection of the load line and the motor speed-torque curve. The motor always operates between the breakdown torque point and the synchronous speed point that corresponds to the 1800 rpm location on the horizontal axis. If additional load torque is required, the motor slows down and develops more torque by moving up toward the breakdown torque point. Conversely, if less torque is required, the motor speeds up slightly toward the 1800 rpm point. Again, if the breakdown torque requirements are exceeded, the motor will stall.

Figure E-10 depicts the same motor speed-torque curve, but now the motor current has been shown for full voltage starting.

Typically, when a NEMA design B induction motor is started across the line, an

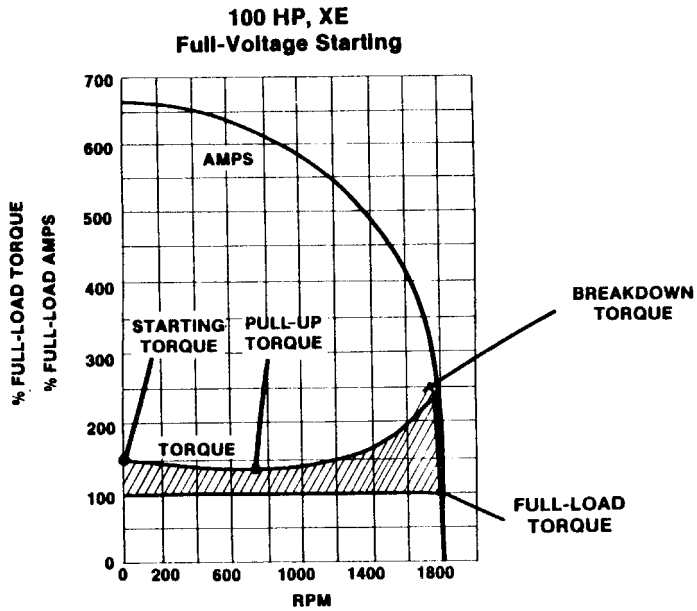


FIG. E-10 Motor current of induction motor started at full voltage. (Source: Reliance Electric.)

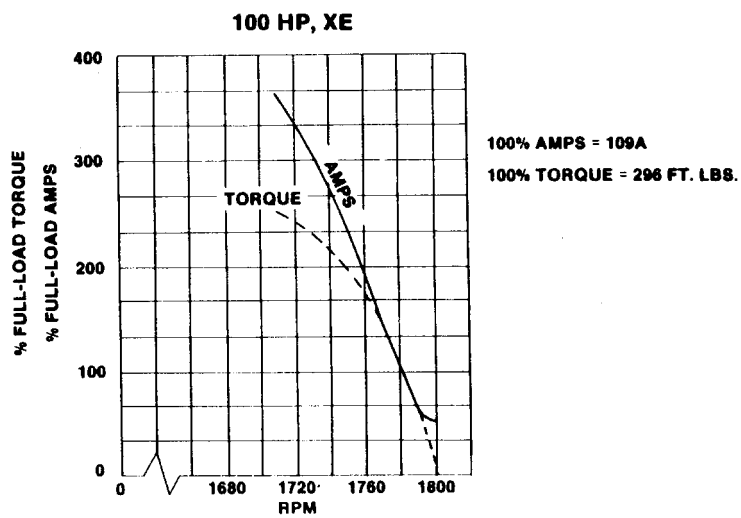


FIG. E-11 Motor current and torque as full operating speed is approached. (Source: Reliance Electric.)

inrush current of 600 percent to 700 percent occurs corresponding to the starting torque point. As the load is accelerated to the full-load torque point, the current decreases to 100 percent full-load current at 100 percent full-load torque. High currents, however, are drawn during the acceleration time.

The amount of time that the motor takes to accelerate the load will depend on the average available accelerating torque, which is the difference between the motor speed-torque curve and the load speed-torque curve, and the load inertia.

Figure E-11 illustrates a blown-up view of the region between the breakdown torque point and the synchronous speed point, which is where the motor would

operate. This is of particular interest because the current for various torque requirements can easily be seen. This would directly affect the size of the controller required to produce a given torque because controllers are current-rates.

At 100 percent full-load torque, 100 percent full-load nameplate current is required. At 150 percent torque, 150 percent full-load nameplate current is required. Beyond the 150 percent full-load torque point, however, the torque-per-amp ratio is no longer proportional. For this case, 251 percent breakdown torque would require 330 percent current.

Adjustable-frequency controllers are typically rated for a maximum of 100 percent continuous or 150 percent for one minute of the controller full-load current. This would generally provide a maximum of 100 percent or 150 percent of motor full-load torque. This would not, however, provide the same amount of torque as the motor could potentially develop if it were operated from utility power, which could normally provide as much current as the motor required.

It would generally be uneconomical to oversize a controller to obtain the same amount of current (torque), since the controller size would actually triple for this example in order to provide 251 percent torque.

Two basic concepts that can explain adjustable-speed operation of induction motors can be summarized as follows:

$$\text{Speed} \propto \frac{\text{frequency}}{\text{poles}}$$

$$\text{Torque} \propto \text{magnetic flux} \propto \frac{\text{volts}}{\text{hertz}}$$

The speed of an induction motor is directly proportional to the applied frequency divided by the number of poles. The number of poles is a function of how the motor is wound. For example, for 60 Hz power, a two-pole motor would operate at 3600 rpm, a four-pole motor at 1800 rpm, and a six-pole motor at 1200 rpm.

The torque developed by the motor is directly proportional to the magnetic flux or magnetic field strength, which is proportional to the applied voltage divided by the applied frequency or hertz. Thus, in order to change speed, all that must be done is to change the frequency applied to the motor. If the voltage is varied along with the frequency, the available torque would remain constant. It is necessary to vary the voltage with the frequency in order to avoid saturation of the motor, which would result in excessive currents at lower frequencies, and to avoid underexcitation of the motor, which would result in excessive currents, both of which would cause excessive motor heating.

In order to vary the speed of an induction motor, an adjustable-frequency controller would have an output characteristic as shown in Fig. E-12. The voltage is varied directly with the frequency. For instance, a 460-volt controller would normally be adjusted to provide 460 volts output at 60 Hz and 230 volts at 30 Hz.

A controller would typically start an induction motor by starting at low voltage and low frequency and increasing the voltage and frequency to the desired operating point. This would contrast with the conventional way of starting induction motors of applying full voltage, 460 volts at 60 Hz, immediately to the motor. By starting the motor with low voltage and low frequency, the inrush current associated with across-the-line starting is completely eliminated. This results in a soft start for the motor. In addition, the motor operates between the breakdown torque point and synchronous speed point as soon as it is started, as compared with starting across the line, in which case the motor accelerates to a point between the synchronous speed and breakdown torque point.

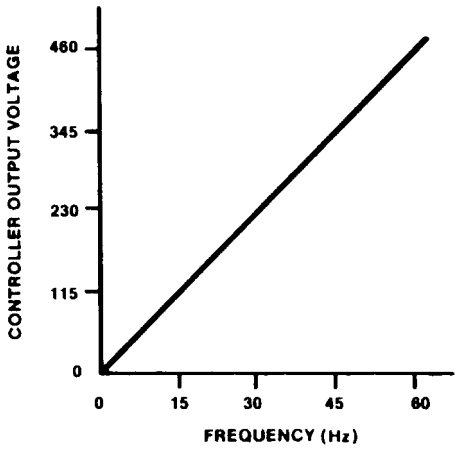


FIG. E-12 Controller output voltage versus frequency relationship for adjustable-speed reduction motors. (Source: Reliance Electric.)

Summary: Motor Selection

- The maximum torque for an induction motor is limited by the adjustable-frequency controller current rating. In order to determine the maximum torque that would be available from an induction motor, it would be necessary to define the motor torque at the controller's maximum current rating.
- The starting torque equals the maximum torque for a motor/controller combination.
- The starting torque current is substantially less for an adjustable-frequency controller/motor combination than the locked rotor current for an induction motor started across the line. This results in a soft start for the controller/motor combination.
- The motor load inertia capability for a controller/motor is much higher, since the controller can limit the motor current to 100 percent or less. This would result, however, in longer acceleration times than starting the motor across the line. Harmonics cause additional motor temperature rise over the temperature rise that occurs for sine-wave power operation. As a rule of thumb, for every 10°C rise in temperature, the motor insulation life is cut in half. This explains why it is important to consider the additional temperature rises associated with adjustable-frequency control power and to follow the suggested rating curves provided by capable motor manufacturers.
- NEMA design C and D motors are not recommended for use on adjustable-frequency control power because these motors have high watts loss due to higher rotor watts loss over design B motors and resulting high temperature rises when operated on adjustable-frequency control power.
- Key application points must be defined in order to properly apply an induction motor to a solid-state adjustable-frequency controller torque, speed range, motor description, and environment. In order to ensure that adequate torque is available to drive the load and adequate current is available to produce the required torque, the starting torque, the peak running torque, and the continuous torque requirements must be defined. The continuous torque is usually defined, but the peak and starting torques are more difficult to define. For the case of retrofit applications, the speed-torque curve of the existing motor might be used as

a reference to define the starting and peak-load torque. Sizing the controller for these points, however, would frequently result in a larger controller than necessary.

- The speed range affects the motor thermal rating. The controllers will typically provide a 10 to 1 speed range below 60 Hz.
- The motor description will permit selection of a controller size for the motor horsepower, voltage, and current rating. The motor insulation class and design type will permit the motor to be rated properly to ensure that its thermal limitations are not exceeded.
- It is necessary to consider the environment to choose the proper motor enclosure. Explosion-proof motors usually have a UL label certifying that they are suitable for the defined classified area. The UL label, however, is suitable only for 60 Hz sine-wave power. When an explosion-proof motor is operated on adjustable-frequency control power, the 60 Hz sine-wave UL label is voided. In addition, induction motors are normally rated for 40°C (104°F) ambient temperature. Use in a higher ambient temperature may require additional cooling or overframing.

Reference and Additional Reading

1. Bloch, H., and Soares C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Special Application Case 1: Recommended Features for High-Corrosion Applications*

Features, depending on level of end-user customization, include:

Total cast iron construction

The motor (see Fig. E-13), including frame with integrally cast feet, end brackets, bearing inner caps, fan cover, conduit box, and cover construction, is cast iron ASTM Type A-48, Class 25, or better. Steel, aluminum, or plastic construction is not acceptable for these features for NEMA sizes.

Insulation system

Motor insulation is a Class F minimum, utilizing materials and insulation systems tested in accordance with IEEE 117 classification tests. The wound stator assembly is to receive a varnish treatment with multiple dips and bakes. Motor leads are to be nonwicking type, Class F temperature rating or better, and permanently numbered along the entire length for easy identification.

Positive lubrication system (PLS)

Motor is to be provided with the positive lubrication system (PLS). (See Fig. E-14.) This system includes open, single-row, deep groove, Conrad-type bearings with a Class 3 internal fit conforming to AFBMA Standard 20. Belted duty applications may require a cylindrical roller bearing.

The PLS is a patented, uniquely designed, open bearing system that provides long, reliable bearing and motor life through positive lubrication directly into and through the bearing track, regardless of mounting position.

The lubrication system consists of a grease inlet on the motor bracket with capped grease fitting. The grease relief plug is 180° from the inlet to provide complete

---

*Source: Reliance Electric test (model referenced is Duty Master XT). Adapted with permission.

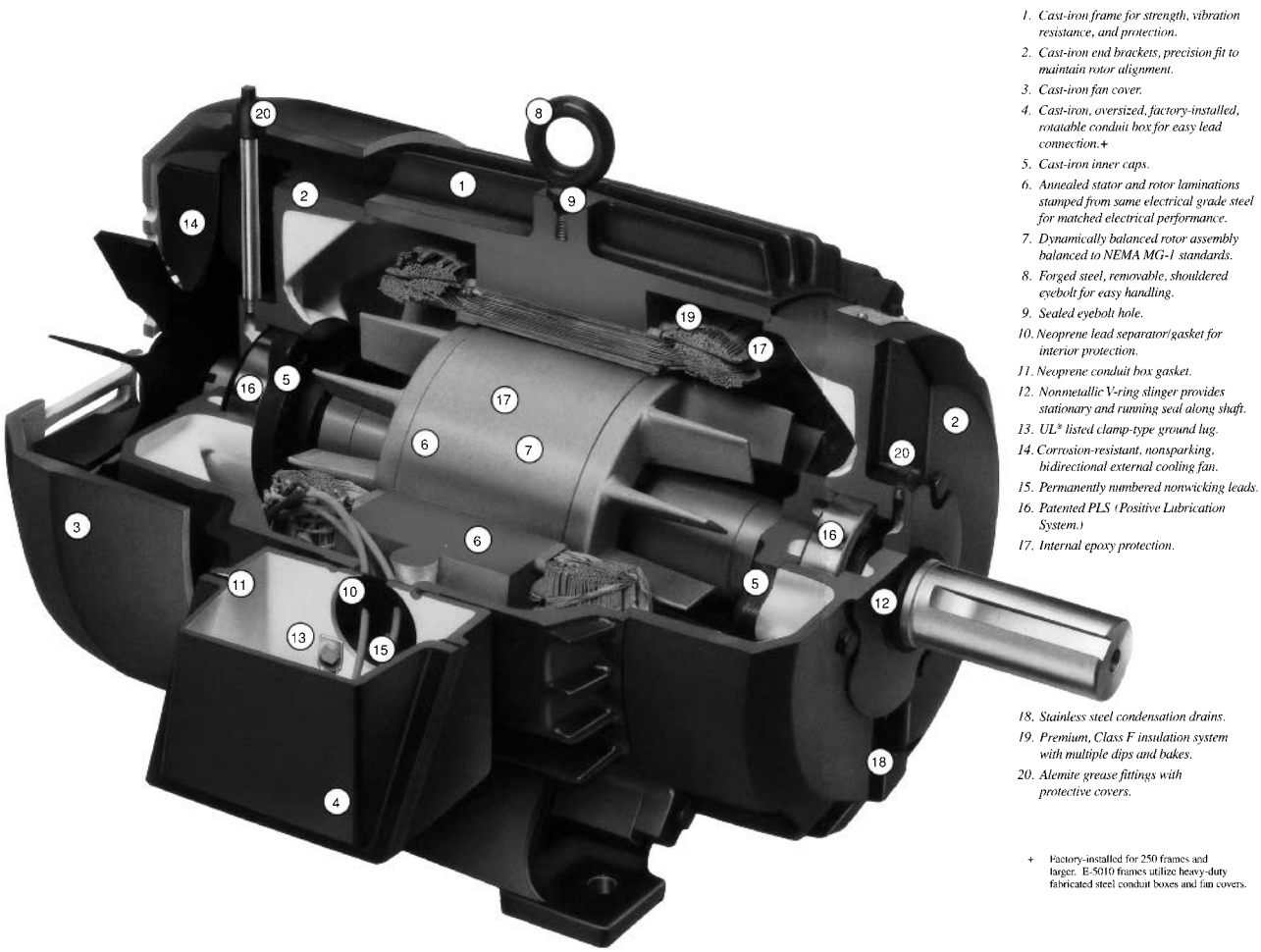


FIG. E-13 Special features of a motor used in high-corrosion applications. (Source: Reliance Electric.)

regreasing without damage to the bearing. The grease entry into the bearing is designed to direct grease into and completely through the open bearing regardless of motor mounting position. Cast iron inner caps are provided, for both bearings, with antichurning grease vanes. The inner caps and end brackets provide a large grease reservoir.

Cooler bearing temperatures. Open bearing (nonshielded) construction (1) minimizes friction, allowing cooler bearing operation. See Figs. E-13 and E-14 for numbered items.

Oil mist lubrication. PLS open bearing design allows for easy conversion to oil mist systems. Positive Lubrication/Relubrication in any mounting position. Exclusive Grease Channeling Passage (2) with minimum grease path entry (3) channels grease directly into the bearing track.

Corrosion control. Small clearance on either side of the grease passage uniformly distributes grease to both inboard and outboard reservoirs (4). Bearing system is completely greased during motor assembly.

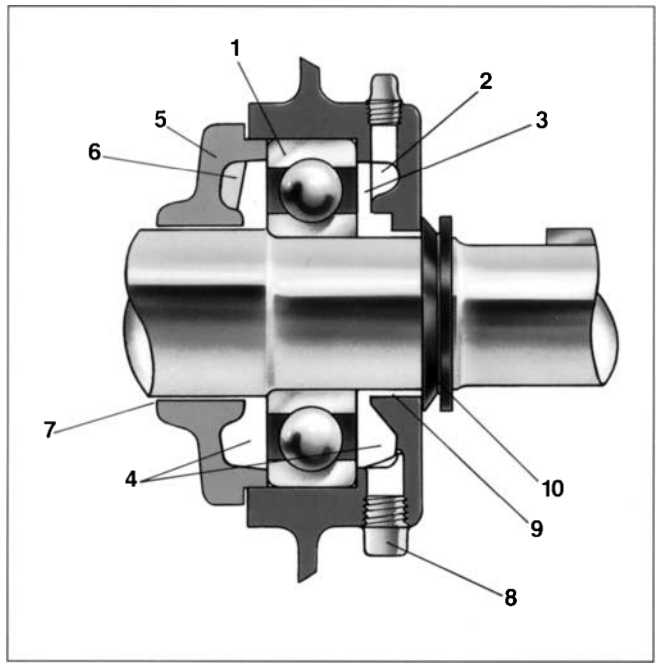


FIG. E-14 PLS features. (Source: Reliance Electric.)

Restricts inboard contaminants. Inner bearing cap (5) with antichurning vanes (6) and close running shaft tolerances (7) minimizes contaminant entry into bearings and grease migration into motor.

Prohibits overgreasing during lubrication/relubrication. Tapped drain (8) ensures grease relief. However, if drain is plugged, PLS design will relieve grease along the shaft (9).

Unique V-ring shaft slinger (10) provides seal for both running and stationary motor protection. Wear-resistant Buna-N shaft seal features a unique V-ring design. This allows the lip to seal flush against the end bracket when the motor is at rest. When the motor is running, centrifugal force pulls the lip away from the end bracket to act as a rotating slinger. Because the seal fits flush with the end bracket rather than being recessed, there is no place for water to pool when the motor is mounted in a shaft-up position.

Standards for quality and testing of XT motors meet or exceed NEMA, NEC, IEEE, ANSI, and ASTM standards where applicable.

Fully gasketed construction

Fits between frame and end brackets are completely sealed with RTV compound to prevent contaminant intrusion.

The cast iron conduit box, diagonally split and rotatable in 90° increments, is provided with tapped NPT threaded conduit hole. A neoprene gasket is used between the conduit box and cover. A neoprene lead separator/gasket, located between the conduit box and frame, is constructed such that there are no unused lead hole openings. The lead separator/gasket is designed to prevent moisture and condensation from migrating into the motor enclosure.

Bidirectional fan

The motor is provided with a bidirectional, corrosion-resistant, nonsparking fan that is clamped, keyed, and shouldered to the motor shaft. Unidirectional fans are specifically prohibited.

Corrosion-protected hardware

All mounting hardware is hex head, high strength, SAE, Grade 5, zinc-plated for corrosion protection. Screwdriver slot fasteners are prohibited. A forged-steel, shouldered, removable eyebolt is provided on all frames. The eyebolt hole is designed to prevent moisture or foreign material from entering the motor when the eyebolt is removed.

Corrosion-resistant stainless steel nameplates are affixed to motor frame with stainless steel or brass drive pins. Nameplates include all required NEMA data and AFBMA bearing numbers, lubrication instructions, and connection diagram (when required).

Internal corrosion protection

The complete internal rotating assembly and stator winding are epoxy coated to maximize corrosion protection of electrical components from dust, acid, moisture, and other contaminants. The patented PLS ensures longer bearing life.

Breather drains

Stainless steel condensation drains are mounted in the lowest part of both end brackets of the motor to provide drainage in any mounting position.

Testing prior to, during, and after assembly

Manufacturing facilities often employ statistical process control throughout the manufacturing process to ensure component integrity. Prior to winding, stator cores receive a core loss check. Wound cores receive a high potential test at twice-rated voltage plus 1000 volts prior to multiple dipping and baking. Rotors are dynamically balanced to commercial standards and shafts are inspected for runout.

After assembly, all motors are surge-tested and checked for key electrical and mechanical characteristics, including no-load watts and amps and locked rotor amps and torque. Vibration is also checked to ensure proper assembly and bearing quality.

Continuous exposure in corrosive environments is common in the petrochemical, paper, water, and waste treatment, mining, and other processing industries. See Figs. E-15 through E-23.

Special Application Case 2: AC Induction Motors Used for Variable Frequency Control*

Until recently the majority of AC variable-speed drives have been applied to variable torque, pump, and fan applications. Advances in drive technology have led to the use of induction motors in high-performance applications that exceed the capability of motors designed for operation on sine wave power. These applications, which have traditionally been served by DC systems, have created the need for definite purpose AC induction motors designed specifically for operation on adjustable-frequency controllers. This application study will highlight the limitations of standard motor designs.

The reasons for operating industrial motors over a range of speeds are as varied as the industries served. The need for variable-speed prime movers is widespread—energy savings on fan drives, constant surface speed cutting on machine tool

---

*Source: Adapted from extracts from Melfi and Hart, "Considerations for the Use of A-C Induction Motors on Variable Frequency Controllers in High Performance Applications," Reliance Electric, USA.

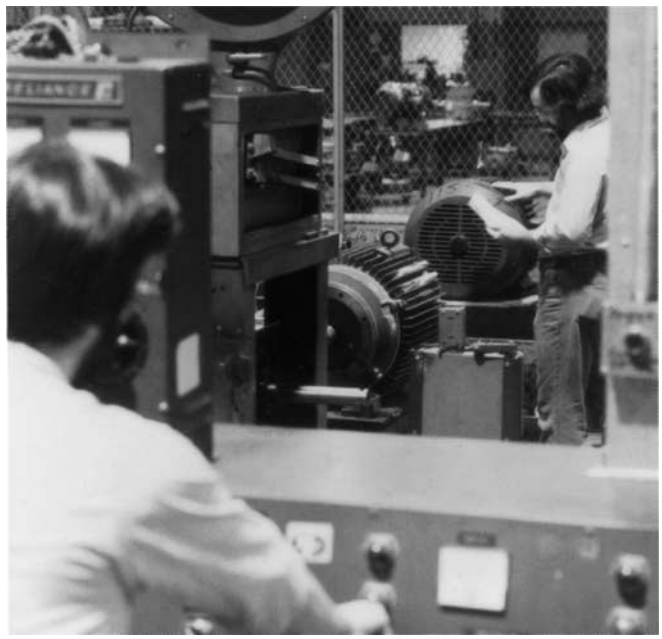


FIG. E-15 Dynamometer testing of Duty Master motors under full-load capacity verifies efficiency and power factors. (Source: Reliance Electric.)



FIG. E-16 Hi-pot testing of wound stator assures electrical integrity per NEMA MG-1 specifications. (Source: Reliance Electric.)

spindles, wind and unwind operations of a bridle drive, etc. Improved performance of these variable-speed drive systems has always been a key means for achieving increased factory productivity. While various methods have historically been used to achieve these speed ranges, advances in technology are making one of the options more attractive than ever.

The low cost and ruggedness of the AC squirrel cage induction motor are benefits that have increased the desire to use it as the electromechanical energy conversion means. Today's control schemes are obtaining higher levels of performance from these AC motors as well. However, a common limiting characteristic of AC induction motors' performance (on adjustable-frequency controls) has not been a technological limitation. Rather, it has been a limitation imposed by the nature of the standardization of industrial AC motors for general-purpose, constant-frequency



FIG. E-17 Complete rotor assembly is dynamically balanced. (Source: Reliance Electric.)



FIG. E-18 Totally assembled motor is vibration tested to NEMA standards. (Source: Reliance Electric.)

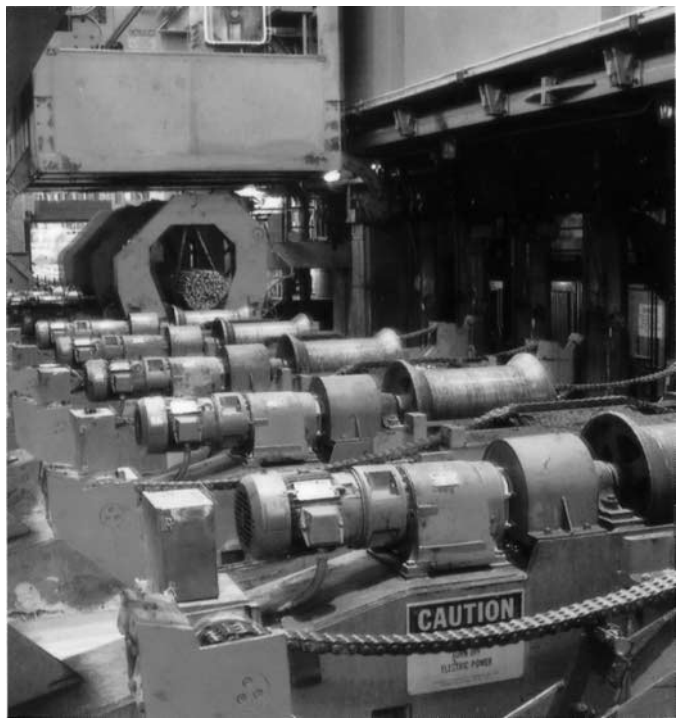


FIG. E-19 Motor-driven power conveyors in a bar mill bundling operation. (Source: Reliance Electric.)



FIG. E-20 Motor-driven paper mill waste handling conveyor. (Source: Reliance Electric.)



FIG. E-21 Motor-driven barge loading conveyor at a coal handling terminal. (Source: Reliance Electric.)



FIG. E-22 Motor-driven aeration pumps in secondary sewage treatment operation. (Source: Reliance Electric.)

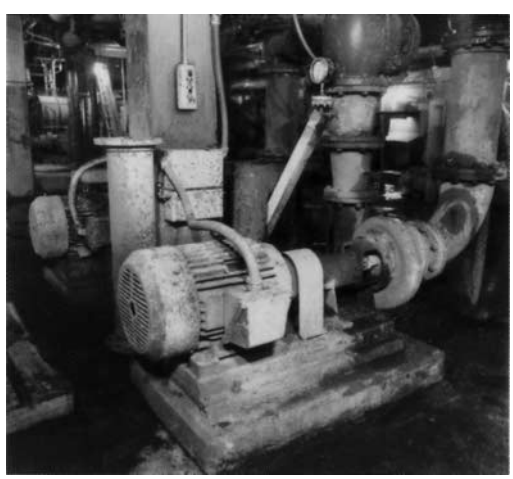


FIG. E-23 Motor-driven sewage water pumps in a sludge dewatering process. (Source: Reliance Electric.)

use. Throughout this highly refined standardization process there has been little consideration for operation on variable-frequency power.

Until recently the majority of high-performance industrial applications have been satisfied using DC motors and controls. This technology has well-defined standards and has been dominated by a limited number of manufacturers where the control supplier assumes responsibility for the performance of the control/motor system.

The rapid development of adjustable-frequency AC technology has encouraged a large number of new control manufacturers to enter the market. As would be expected, their primary experience is in electronics and not variable-speed system application. Also, few of these control manufacturers produce motors. The majority of AC motor manufacturers have limited variable-speed experience as their products have traditionally operated at a fixed frequency and speed. Due to the large number of possible control and motor design combinations in the marketplace, it is impractical to assume all combinations have been tested extensively. In this environment the machine builder and user accept greater responsibility for the total system performance and greater knowledge of the component design considerations and limits is needed.

High-performance drives

When the drive (motor and control) performance requirements are minimal, a standard industrial AC induction motor can often be successfully applied to adjustable-frequency power, variable-speed applications. Indeed, some applications can be converted from constant speed to variable speed while utilizing an existing induction motor. However, when the performance level required is more demanding, a definite-purpose motor design is appropriate. This is usually the case when maximum process productivity is the goal.

While the definition of a high-performance application is not precise, these applications will typically have one or more of the following characteristics:

- Continuous constant torque required below 50 percent of base speed
- Continuous constant horsepower required above 150 percent of base speed
- High starting loads or overloads
- High dynamic performance
- A process (driven machine) that cannot be started or run without a variable-speed control

The vast majority of adjustable-frequency AC controls applied to date have been on low-performance applications such as pumps, fans, and mixers. Only recently have significant numbers been applied to applications such as extruders, winders, and coordinated web processes that meet the criteria above. As improvements in control technology make these applications commonplace there is a need for definite purpose motors designed specifically to optimize the performance of the drive.

General design considerations

The first task is to design a basic motor configuration that is matched to the general needs of adjustable-frequency power and variable-speed operation. Second, the design must be adaptable to match the specific needs of many different drive applications. Third, by relaxing inappropriate constraints associated with fixed frequency, voltage, and speed applications, the design can be tailored to meet the performance objectives by making typical design tradeoffs as outlined in Table E-5. Also, when the controller design is known, more subtle techniques that include

TABLE E-5 Changing Motor Parameters to Meet Performance Objectives

Objective	Parameter Change
Wide constant HP speed range	Increase peak torque at base
Higher peak torque	Oversize motor
	Decrease stator and rotor inductance
	Decrease stator resistance
Lower primary time constant	Increase stator resistance
	Decrease inductances
Higher stator resistance	Increase stator coil turns
	Decrease stator wire/slot size
Lower inductances	Decrease stator coil turns
	Increase flux densities
	Change slot shapes
Lower flux density	Increase volume of core
	Increase stator coil turns
Lower magnetic noise	Decrease slot size
	Decrease flux density
	Alter shape/volume of material
Higher efficiency	Decrease stator resistance
	Decrease rotor resistance
	Reduce flux density

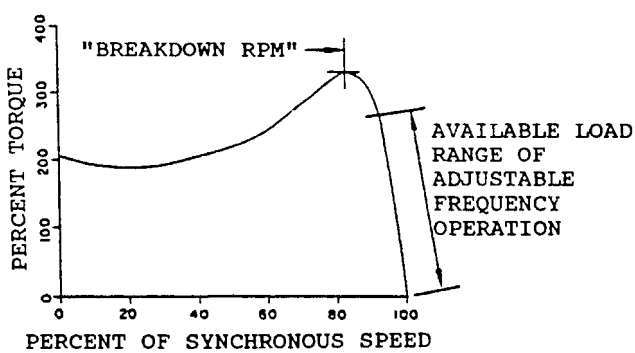


FIG. E-24 "Breakdown rpm" region. (Source: Reliance Electric.)

the controller can be used. An example is the use of a lower than usual voltage at the low speed end of a region of constant horsepower, so that the flux level (hence, peak load capability) at the highest speeds can be maximized to produce sufficient torque without having to oversize the motor. Of course, this must be weighed against the increased current required of the controller at the low speed.

Starting characteristics

Since adjustable-frequency controllers typically accelerate a motor and load by slewing the motor voltage and frequency in such a way as to remain in a region of operation above "breakdown rpm" (as illustrated in Fig. E-24), the usual constraints of fixed voltage, fixed frequency, starting, and acceleration do not apply. Starting torque and current are no longer functions of the 1.0 per unit slip characteristics of the motor but are limited by the overload capability of the control. Thus, the controller can be matched to the motor in such a manner as to produce

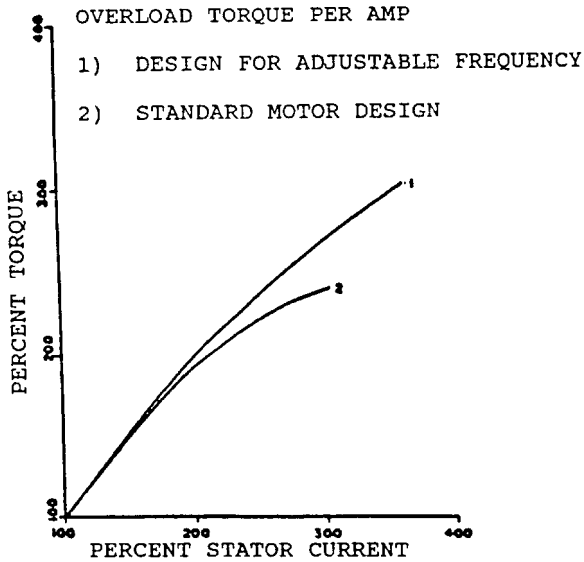


FIG. E-25 Use of overload torque per amp as a design feature. (Source: Reliance Electric.)

the appropriate starting torque based on a torque/amp ratio equal to that under full-load conditions. By evaluating the drive as a motor and control package, the motor designer can take advantage of this to enhance the level of starting torque as well as overload torque per amp as shown in Fig. E-25.

Peak currents

In addition to the RMS (root mean square) current level, an important rating point for a transistor (typically used in adjustable-frequency controllers) is the peak current capability. The high-frequency transient current that results from the electronic switching of the control output voltage is inversely proportional to the leakage inductance of the motor. As noted in Table E-5 the leakage inductances can be increased by altering the design of the windings and the magnetic cores in the motor. The use of an electromagnetic design specifically for adjustable-frequency power can significantly reduce the peak current required for a given level of power output (see Fig. E-26). This will not only improve the reliability of the drive, but often can prevent costly oversizing of the AC controller and provide the most cost-effective solution.

Motor heating

One of the more obvious sources of increased stress on an induction motor insulation system is higher operating temperature when run on variable-frequency controllers. The higher operating temperatures are the result of increased motor losses and often reduced heat transfer as well. As a result, many standard efficient, fixed-frequency design motors will not achieve their nameplate rating when operated on an adjustable-frequency control at 60 Hz while remaining within temperature limits. While these elevated temperatures may not lead to an immediate insulation failure, they will result in a significantly shorter life. In most modern insulation systems, a 10°C increase in operating temperature will result in a 50 percent reduction in expected life. This is one of the reasons why high efficient designs,

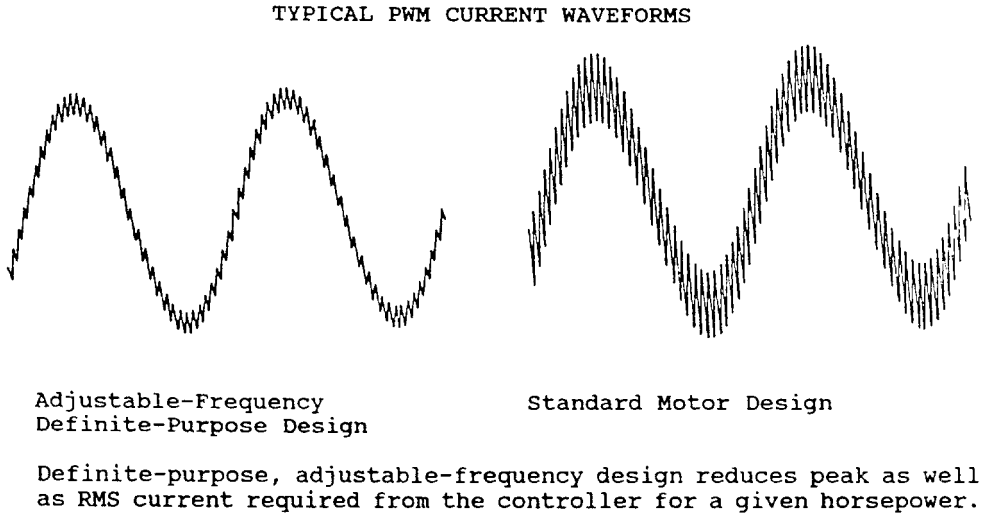


FIG. E-26 Adjustable-frequency design reduces peak and RMS current. (Source: Reliance Electric.)

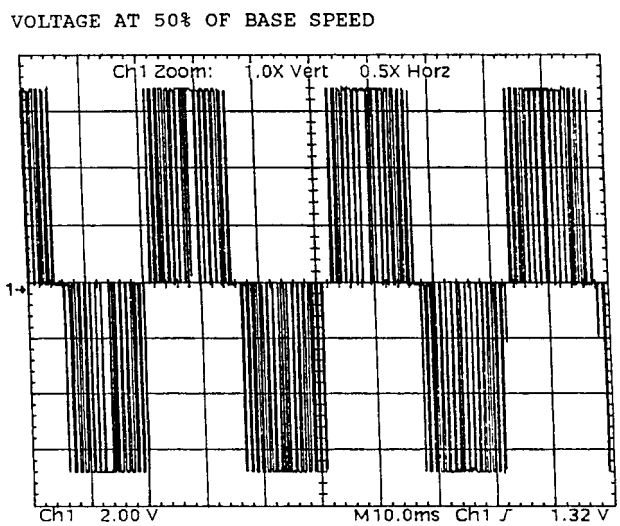
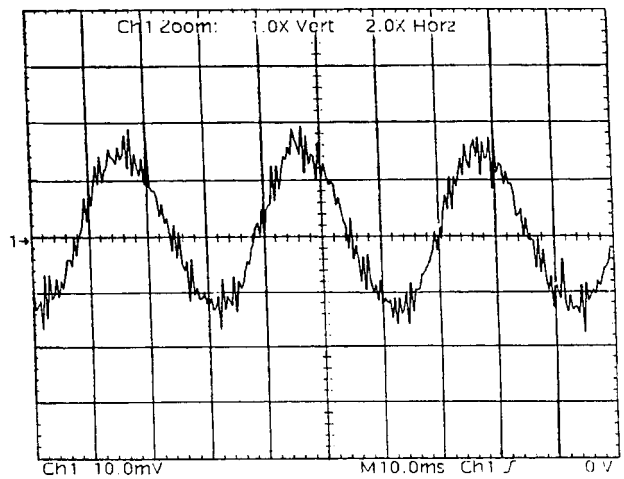


FIG. E-27 Typical waveforms from adjustable-frequency controllers. (Source: Reliance Electric.)

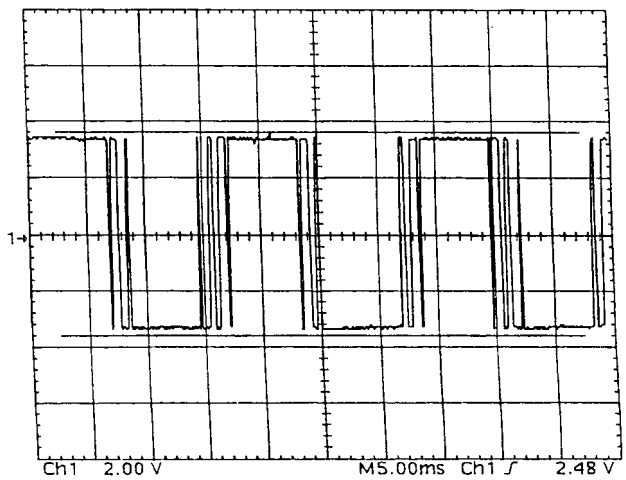
which have inherently greater thermal reserves, are often recommended for operation on adjustable-frequency controls.

When an induction motor is run with voltage and current waveforms as seen in Fig. E-27, the deviation from the ideal sinusoidal waveshapes creates additional losses without contributing to steady-state torque production. The higher-frequency components in the voltage waveform do not increase the fundamental air gap flux rotating at synchronous speed. They do, however, create secondary hysteresis loops in the magnetic steel, which, along with high-frequency eddy currents, produce additional core losses and raise the effective saturation level in the lamination material. As another consequence of these higher-frequency flux variations, there are higher-frequency currents induced in the rotor bars that generate additional losses. Appropriate electromagnetic design, including rotor bar shape, can minimize these added losses.

CURRENT AT 50% OF BASE SPEED



VOLTAGE NEAR BASE SPEED



CURRENT NEAR BASE SPEED

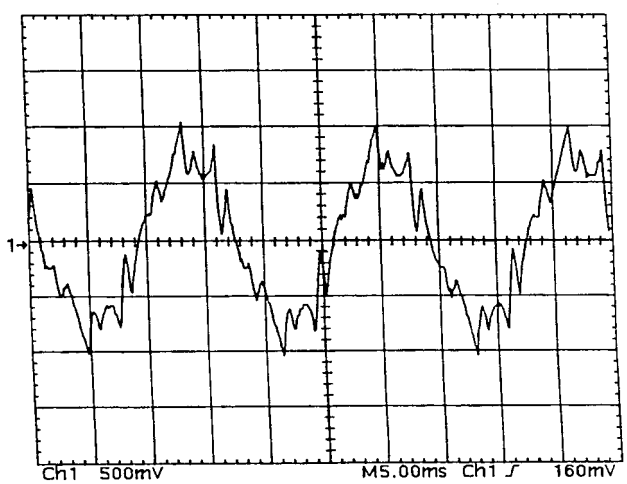


FIG. E-27 (*Continued*)

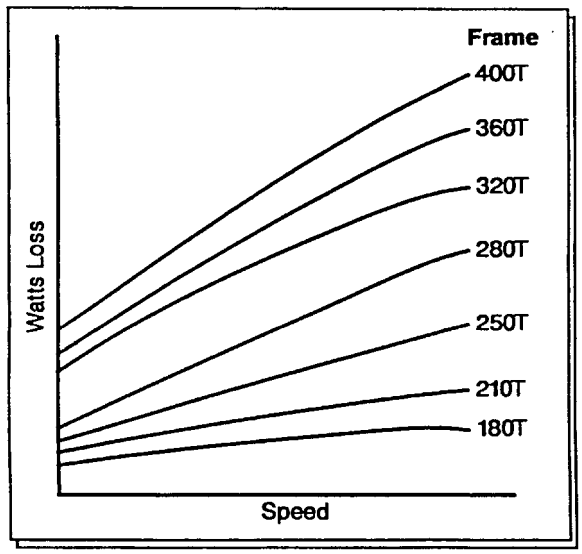


FIG. E-27 Cooling curves for tefc motors. (Source: Reliance Electric.)

The higher-frequency components of the current waveform also do not contribute to the steady-state torque. They do, however, increase the total RMS current resulting in added IR (I, current; R, resistance) losses in the stator winding. In addition to higher-frequency current components there can also be low-frequency "instabilities" in the currents seen by the AC motors on variable-frequency controllers. These asynchronous components of current again cause added losses without contributing to the steady-state torque production. Motor designs that help minimize harmonic currents lead to low IR losses.

Motor cooling

As has been well documented in the literature, when AC motors are run across a wide speed range their heat-transfer effectiveness will vary a great deal. Cooling fans whose rotation is directly supplied by the motor are subject to high windage losses and noise at high speeds. Modern AC controllers are capable of operating across a very wide frequency range, often up to several hundred hertz. While this provides great flexibility in the control, it places the motor cooling fan well above its fixed-frequency design operating point, which often leads to inefficient airflow and objectionable noise. In low-speed operation the fan's effectiveness falls off with the motor's speed. Figure E-27 shows typical cooling curves for a family of totally enclosed fan-cooled motors. In variable-torque applications this reduction in cooling air often stays in balance with the reduction in motor losses as the load is reduced with speed. However, in constant-torque applications the motor's temperature limits will likely be exceeded. An independently powered blower can provide an essentially constant heat-transfer rate. Although not a standard fixed-frequency motor feature, depending on the load/speed profile required by the application, this can be a very effective choice and is often specified for high-performance applications.

In addition to fan speed, the operating temperature of the motor is determined by how effectively the heat generated in the motor can be conducted to surfaces that are in contact with the cooling medium (generally air) and the ability to transfer this heat via convection to the cooling medium. In a conventional totally enclosed fan-cooled motor the heat must be transferred from the laminated steel

stator core to the cast iron frame and finally to the air. Since the fan is located opposite the drive end of the motor, there is generally greater airflow and heat transfer at one end of the motor than the other. Square laminated frame AC motors have been offered by a variety of manufacturers as a method to improve heat transfer. The laminated frame design eliminates the stator-to-frame interface and provides a more direct and effective heat-transfer path to the cooling air while integral cooling ducts trap the air in contact with the frame along the motor's length. This laminated frame construction has been common in variable-speed DC motors for over twenty years.

An offshoot of motor cooling is the need to protect the motor should the motor cooling system fail. While thermostats and thermistors are not common in fixed-frequency AC motors they should be required for variable-speed applications. A standard AC motor operates at a fixed speed on a well-defined power supply that allows the shaft-driven fan to provide adequate cooling air in all normal circumstances. By design a variable-frequency control will allow the motor to operate at very low speeds where little or no cooling is provided. This might occur during maintenance, jog, or threading operation, for example. On the other hand, if a separately powered blower is provided, the drive motor must be protected from a potential blower failure. As is the case with DC motors, over-temperature protection is recommended.

Disadvantages of oversizing (derating) motors

In applying variable-frequency controllers, attempts are often made to use either in-place AC motors or standard sinewave power designs. To do this and operate across a speed range the motor is often oversized relative to the rating required by the application. This can sometimes be done successfully, but there are a number of potential pitfalls. These can range from something as basic as a motor insulation system, which is fine on sinewave power but inadequate for the voltage and current waveshapes on the controller, to drive system instability due to a lack of damping. The oversized motor will have correspondingly higher rotor inertia, which could lengthen acceleration and deceleration times and reduce process productivity. Also, since no-load current tends to be a fairly constant percentage of full-load current within a motor product line, the higher no-load current of a derated motor could result in lower power factor and higher current at the load point required by the application. This current may exceed the capability of the variable-frequency controller requiring a costly oversizing of the controller as well. A derated motor will have a lower nominal slip at the application load than a matched motor, which can cause problems either with load sharing in the case of multimotor drives or with IET trips whenever the load changes quickly. While it often appears to be economic to oversize a standard motor to achieve a greater speed range, this course of action should be approached cautiously while weighing all factors of the desired performance of the drive.

The effect of fast power transistors

As power transistor technology has evolved, there has been a proliferation of variable-frequency controllers operating at an AC input voltage of 460 V, using these transistors as the power-switching device. As the transistor manufacturers have continued to push toward devices with lower losses and the capability of the higher switching rates, a result has been very rapid transition times between the off and on states. This is the case for both bipolar (BJT) and insulated gate (IGBT) transistors.

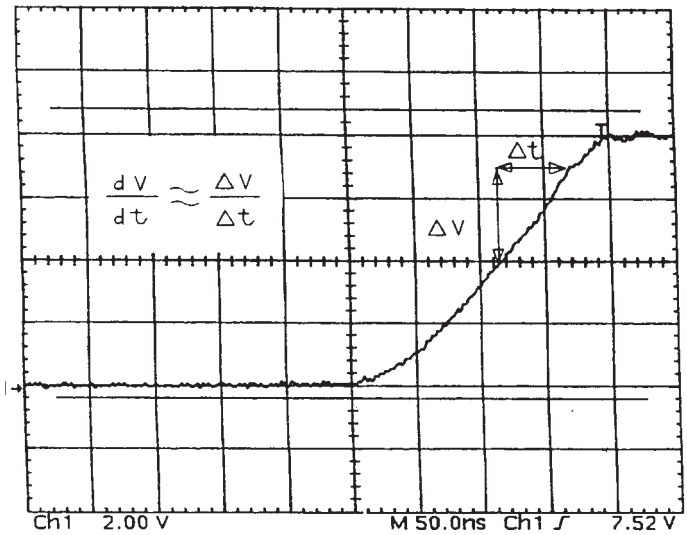


FIG. E-28 Typical transistor transition voltage. (Source: Reliance Electric.)

The combination of fast transitions (turn-on time) and the DC bus voltages of 460 VAC (input) controllers results in the high dV/dt levels as seen in Fig. E-28. What is typically referred to as dV/dt is the time derivative of the voltage, or the slope of the voltage versus time curve.

Increasing the dV/dt levels at the variable-frequency controller output (and motor input) can have effects that need to be considered in the design of motors for such applications. The significance of these effects can be shown by the following equation:

$$I = C \times dV/dt$$

(C is capacitance.) As can be seen from this equation, as dV/dt increases, the capacitively coupled current increases linearly with it. While items such a lead wires and motors are not usually thought of in terms of capacitance, their phase AC motor windings have a capacitance to ground as well as between phases. The leads between the controller and motor also exhibit similar effects. While these capacitance values are normally considered negligible, given enough dV/dt, it does not take much C to get quite a bit of I.

A second way of viewing the high dV/dt levels is to use transmission line theory to compute the voltage distribution due to the propagation of the steep wavefront. This involves careful modeling of the leads and motor windings as well as transition points such as conduit box connections. Reflected as well as incident wavefronts must be computed and combined. This type of analysis will not be described in this paper. Analyses done by this methodology are susceptible to errors due to many things including the choice of appropriate complex impedance models for circuit components. Generally, the results of this type of analysis have indicated that the first length of wire in a motor will see higher voltages than will subsequent parts of the winding. This type of modeling is typically used for the analysis of high voltage surges incident on the terminals of very large machinery.

Another result of the very fast transition time of today's transistors is that the voltage at the inverter output and the motor terminals is not the same. The voltage waveshapes in Figs. E-29 and E-30 demonstrate typical differences. Using the transmission line model mentioned previously, the two major differences in these

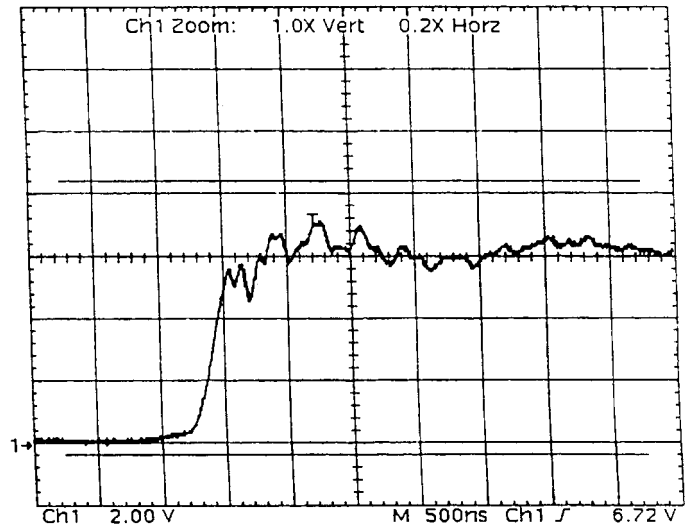


FIG. E-29 Voltage wavefront at inverter output. (Source: Reliance Electric.)

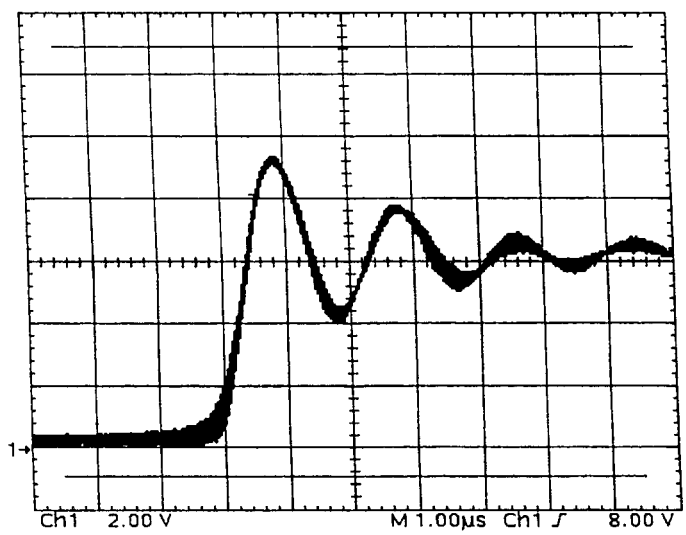


FIG. E-30 Voltage wavefront at motor terminals. (Source: Reliance Electric.)

waveshapes can be explained as follows. The impedance of the leads results in the voltage wavefront being distributed to some extent across those leads, softening the wavefront to a lower dV/dt level at the motor terminals. Secondly, the termination of the transmission line (leads) at the motor results in a reflected wave, producing the overshoot and dampened oscillation seen in Fig. E-30. This waveform could also be modeled as the response of an L (inductance), R, C circuit to an impulse input.

The end result of these waveshapes being applied to the motor terminals is increased stress on the insulation system. Since these waveshapes do not exist in sinewave applications it is clear that their effect has not been considered in standard AC motor insulation systems. The motor insulation system must be capable of withstanding both the increased thermal stress as well as the capacitively coupled currents and voltage stresses. Appropriate selection of

individual materials properly integrated into a motor insulation system is needed to withstand the demands of operation on variable-frequency controllers.

Motor flux level

The fundamental frequency component of the voltage output of a variable-frequency controller can be as high as the AC input to the controller. However, this is often not achieved. In order to maintain PWM modulation, for example, the output voltage may be limited to 90 to 95 percent of the incoming AC voltage. As long as this situation is recognized, and appropriated design choices made, it does not usually present a problem. When an existing motor design (expecting 460 V at 60 Hz, for example) is applied to a controller that delivers only 420 V, there can be problems.

While NEMA standards for fixed-speed AC motors allow for a 10 percent voltage variation from nominal, it is important to recognize that at 10 percent lower than nominal flux, performance, including the nominal horsepower (HP) rating, will vary. For example, it may require 10 percent more current than nominal to deliver rated HP. While this additional current is almost always available from the incoming line, it may not be available from the variable-frequency controller. Users that are familiar with static DC drives and their characteristics in low line conditions may be unpleasantly surprised to find that AC variable-frequency controllers often do not provide the same rating capability at low line conditions. Operation of an AC motor at lower than nominal flux levels will result in increased slip and rotor heating, which is self-compounding and may lead to a thermal runaway condition. High-efficiency AC motors designed for sinewave operation are often particularly susceptible to poor performance when the controller output voltage is low, because they usually employ low flux density designs at nominal terminal conditions.

Measurements in a PWM environment

Another effect of the rapid-rise-time pulses that today's variable-frequency controllers can apply to motors is to challenge existing measurement tools and techniques. The high dV/dt voltage pulses are themselves not trivial to measure. Typically, an oscilloscope with a single shot bandwidth greater than 10 MHz, plus a high-voltage probe with high-frequency capability (carefully impedance matched) is required. Since voltage isolators typically cannot faithfully reproduce these waveshapes, the scope must be "floated" unless the variable-frequency controller is operating on a floating power system. This then requires appropriate care to avoid electrical shock to the operator.

Not only is measuring the voltage pulses difficult, all other measurements on the equipment are exposed to this high dV/dt environment. This requires the use of equipment that has high noise immunity and excellent rejection of common mode voltages. Common devices such as thermocouple and tachometer readouts often provide unreliable readings if they are not capable of faithful operation in these high dV/dt conditions. This effect makes activities such as drive startup and troubleshooting difficult as specialized equipment is required to take even basic measurements.

Noise

Operation of standard industrial AC induction motors on adjustable-frequency power over a speed range often results in unacceptable sound power levels as well as an annoying tonal quality. While the actual sound power level has proven to be unpredictable due to the large number of possible motor and controller designs,

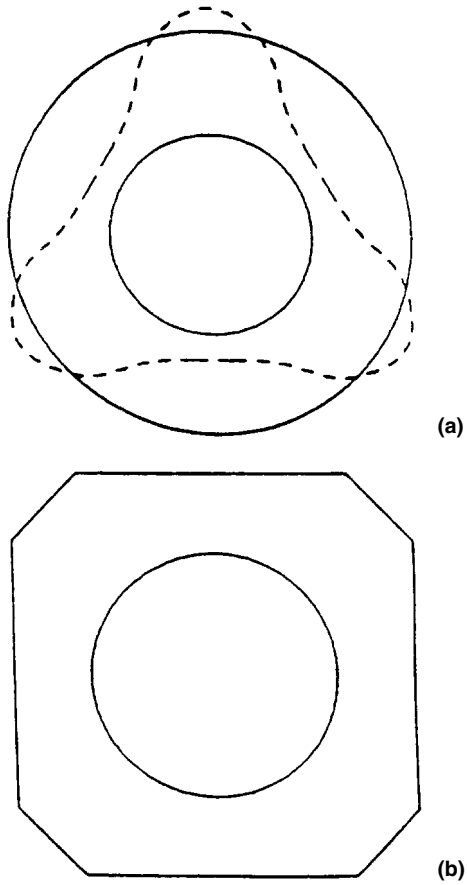


FIG. E-31 (*a*) Third-order mode of vibration in a round-bodied configuration. (*b*) Square configuration tends to suppress odd modes. (Source: Reliance Electric.)

the increase in sound level is typically in the range of 7 to 10 dB. There has been some success in reducing these sound levels by pushing the variable-frequency controller's carrier frequency above the motor structure natural-frequency spectral band. However, there are also motor design considerations that will improve sound levels.

As discussed earlier, one source of acoustic noise is the air noise caused by running shaft-driven fans above their design speed to achieve a wider speed range. A separately powered, unidirectional, constant-speed cooling fan provides a consistent level of air noise independent of motor speed and eliminates annoying sound level changes as the motor accelerates and decelerates.

A second source is the magnetic noise from flux harmonics that drive the magnetic core steel into a saturated condition. A well-planned design will use lower than nominal flux levels with particular emphasis on avoiding localized regions of higher flux density or pinch points. Air gap length and rotor slot bridge thickness, which reduce saturation in localized areas, are two contributing areas where additional reductions in sound power level can be achieved.

Electromagnetic-mechanical noise from parasitic forces, which are caused by flux and current harmonic interactions, produce mechanical vibrations within the motor and contribute to an overall increase in sound power levels. This mechanism will usually become a problem when amplified by mechanical resonances in the motor

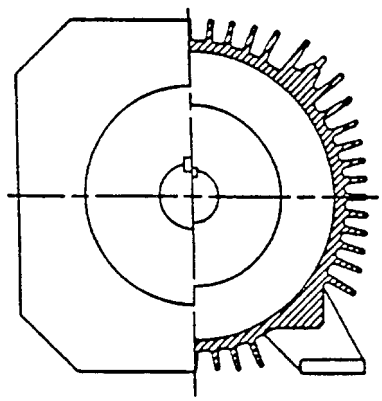


FIG. E-32 Optimized space utilization of square, laminated configuration allows increased active materials and higher power density. (Source: Reliance Electric.)

or driven machine. To offset this source rotor and stator slots can be designed to reduce harmonic flux that contributes to parasitic torques. Also, the use of a laminated frame construction eliminates a separate frame and stator structure, which simplifies the mechanical system and reduces the richness of possible noise-producing natural frequencies and modes of vibration. If a square frame configuration is used it will tend to suppress odd-ordered modes of vibration, which are present round-bodied configurations. This is illustrated in Fig. E-31.

In summary, there are many factors that combine and ultimately result in noise at the motor. The motor and controller must be considered as a system to ensure the desired results.

Mechanical flexibility

A motor designed for operation on a high-performance, variable-frequency drive must have considerable flexibility inherent in its construction to accomplish the variety of tasks it will be called upon to perform. A comparison of the standardized NEMA enclosures for fixed-frequency AC motors to the wide variety of DC motor constructions available demonstrates the difference in the fundamental design approach. Since high-performance, variable-frequency drives will typically be used in DC-like applications as opposed to converting fixed-frequency AC (pumps and fans, etc.) to variable speed, it can be assumed that more DC-like construction will be required in definite-purpose AC motors.

One consideration is to achieve the maximum output from the smallest possible motor. High-performance, adjustable-frequency drives are often incorporated as part of specialized machinery or processes where machine real estate is at a premium. The standardization of NEMA fixed-frequency dimensions creates unnecessarily large motors and offers few alternatives. The practice of oversizing the rating in order to achieve a speed range aggravates the problem. The replacement of the inactive frame material of conventional AC induction motors with active materials (conductors and magnetic steel) in a laminated-frame construction allows a larger air gap diameter and increased power density (Fig. E-32). Often up to two frame diameters can be reduced by using this technique.

Also, to take full advantage of the variable-frequency controller the motor must be capable of operating above its fixed-frequency design speed at 60 Hz. The

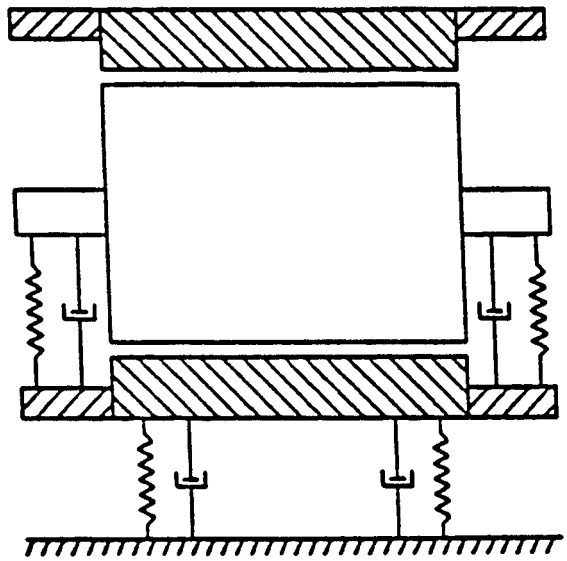


FIG. E-33 Standard feet-on-frame design can result in a two-level dynamic system with lower operating speed capability. (Source: Reliance Electric.)

standard motor design considers only acceleration up to and operation near its synchronous speed. As a result few of these designs are expected to operate above 3600 rpm. The conventional AC motor rotor support to ground system (via bolted joints to the frame, etc.) can give rise to a low stiffness-to-ground and to second-order modes of vibration (two level dynamic systems, as shown in Fig. E-33), which tend to reduce the value of the lowest critical speed. While all elements of a high-speed motor system (bearings, rotor balance, strength, etc.) must be evaluated for suitability, the use of integral feet on the end brackets provides increased stiffness-to-ground by eliminating one of the joints. This can result in increased values of the lowest critical speed and permit operation at higher speeds.

Finally, the motor design must be capable of accepting a variety of accessory devices that are typically mounted on the motor. This includes not only a motor mounting flange but also combinations of brakes, speed feedback devices, and a variety of cooling airflow methods and directions. The design must allow for these devices to be accessed, removed, and replaced in service with little difficulty. Providing these features results in a design approach very similar to DC designs and conflicts with much of the standardization in standard AC motors.

Providing high-performance, variable-speed drives for maximum process productivity has always required complex engineering considerations. Rapid improvements in AC control technology, combined with the ready availability of standard fixed-frequency AC motors has increased the number of possible solutions. However, a component approach (control a, motor b) will not lead to an optimal solution in many cases. In order to utilize the present (and next) generation of adjustable-frequency controllers to meet application needs equal to or better than DC motors have in the past, a definite-purpose AC motor is required. A square laminated-frame configuration with integral feet on the end brackets and adaptable electromagnetic designs is one approach that meets this objective (Fig. E-34).



FIG. E-34 A definite purpose laminated frame AC motor. (Source: Reliance Electric.)

Emergency Power Generation (Packages) (*see* Engines, Gas)

Emissions; Air Emissions (*see also* Turbines)

Emissions from Burning Fossil Fuel

Emissions from burning fossil fuel are:

- SO_x (generally with coal-fueled boilers and steam turbine trains)
- NO_x (with gas turbines)
- CO_2 with all turbines
- Particulates (depending on the fuel)

If gas turbines (fueled by gas or acceptable sulfur-free liquid fuel) conform to current best available technology (BAT), as a rough rule of thumb, emissions generally fall within national guidelines in most countries. Coal-fired power emissions will depend on the quality of coal (sulfur and other impurities content).

As a guideline, the emission standards in a sample country (Canada) for new stationary turbines follow.*

National Emission Guideline for New Stationary Combustion Turbines

1. Emissions of oxides of nitrogen

The emission targets (see Fig. E-35) for various types of combustion turbines are determined by calculation of the allowable mass of NO_x (grams) per unit output of shaft or electrical energy (gigajoules), as well as an allowance for an additional quantity of NO_x emitted if useful energy is demonstrated to be recovered from the facility's exhaust thermal energy during normal operation. Allowable emissions over the relevant time period equal:

$$(\text{Power output} \times \text{A}) + (\text{heat output} \times \text{B}) = \text{grams of } NO_2 \text{ equlvalent}$$

where:

---

* Source: Environment Canada, document CCME-EPC/AIT-49E, pp. 4, 5, and 9. Adapted with permission.

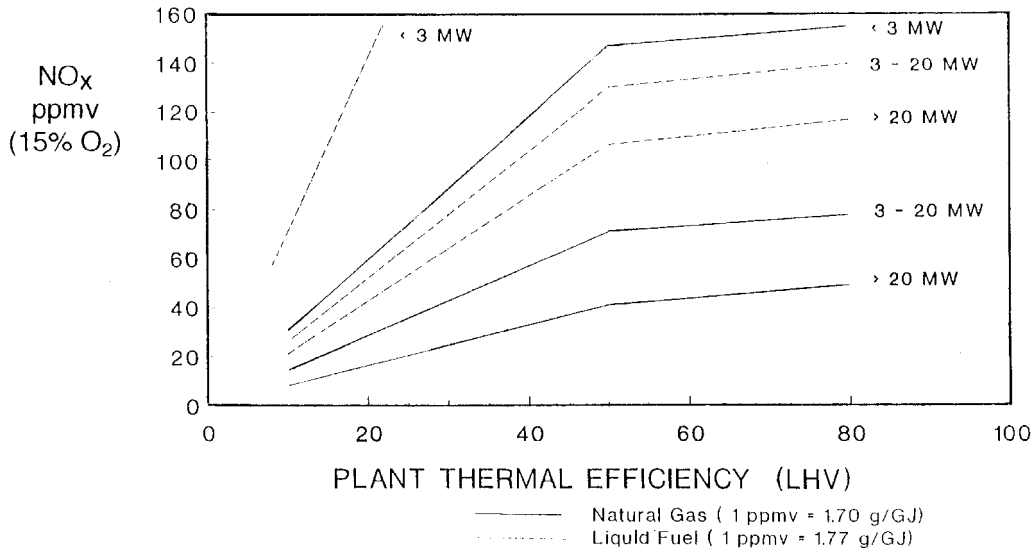


FIG. E-35 NO_x gas turbine emission guidelines: sample conversion from grams/GJ to ppmv (for nonpeaking units). (Source: Environment Canada.)

- Power output is the total electricity and shaft power energy production expressed in gigajoules (3.6 GJ per MW-hour).
- Heat output is the total useful heat energy recovered from the combustion turbine facility.
- A and B are the allowable emission rates, expressed in grams per gigajoule, for the facility's power and heat recovery components, respectively, as summarized below.

Power Output Allowance A (g/GJ)

	Natural Gas	Liquid Fuel
Nonpeaking turbines		
Less than 3 MW	500	1250
3–20 MW(1)	240	460
Over 20 MW	140	380
Peaking turbines		
Less than 3 MW	Exempt	Exempt
Over 3 MW	280	530

Notes:
(1) For the initial two-year period of implementation of the guideline, the emission allowances for 3–20 MW sized nonpeaking units are 350 g/GJ and 600 g/GL, for gaseous and liquid fuels, respectively.
(2) The value of A has been set at 500 g/GJ for solid-derived fuels, which recognizes that the competing alternative technology option is a conventional coal-burning steam electric powerplant.

Heat Recovery Allowance B (g/GJ)

For all units:	Natural gas	40
	Liquid	60
	Solid-derived	120

The heat recovery allowance is a NO_x emission allowance for energy recovered from turbine exhaust gases as heat. The allowance corresponds to the emissions savings

that result from not serving a heating or cooling load by burning additional fuel in an industrial boiler, but by recuperating heat during normal operation from combustion turbine exhaust gases and other sources of waste heat such as condensate.

2. Emissions of carbon monoxide (CO)

For units that are covered by the NO_x provisions of this guideline, emissions of CO corrected to ISO conditions at 15 percent oxygen and on a dry volume basis should not exceed:

- 50 parts per million at its power rating

3. Emissions of sulfur dioxide (SO_2)

Sulfur dioxide emissions from combustion turbines can be limited by using low sulfur content fuels, or by using technologies that reduce the fuel sulfur content or that capture sulfur dioxide emissions in the exhaust. Sulfur dioxide emissions should not exceed:

- *For liquid and gaseous fuel:* For nonpeaking units, 800 g/GJ of output, and for peaking units, 970 g/GJ of output, all based on the lower heating value of the fuel.
- *For solid-derived fuels:* 770 g/GJ of output for those fuels whose uncontrolled SO_2 emissions based on fuel sulfur content would be between 770 and 7700 g/GJ of output, or a minimum of 90 percent sulfur capture for those fuels whose uncontrolled SO_2 emissions based on fuel sulfur content would be greater than 7700 g/GJ of output.
- Units with a power rating of less than 3 MW that are used exclusively to power natural gas field compressors upstream of natural gas processing facilities are exempt from the SO_2 limits.

4. For other emissions in air, *see* Pollutants

Engines, Gas

An *engine* is the term used to describe a reciprocating piston machine that is fueled by either gas or oil (sometimes diesel oil). It can be either a two-stroke or a four-stroke engine. Diesel-driven engines are sometimes used to provide a source of emergency power if the main turbine generators in a plant have a temporary outage. Most turbine vendors either know sources of emergency power-generation packages (of which a diesel engines is part) or they package and supply them themselves.

With* twice as many power strokes per revolution, the two-stroke-per-cycle engines tend to be smaller than four-stroke-per-cycle versions built for the source power output. Moreover, the two-stroke machine is generally less complex because it can dispense with valves and their associated mechanisms.

Reciprocating gas engines display typical efficiencies in the 28 to 42 percent range. The upper portion of this range belongs to turbocharged engines, whereas the lower range is populated by naturally aspirated machines. (See Fig. E-36.)

---

* Source: Cooper-Bessemer Reciprocating, USA.

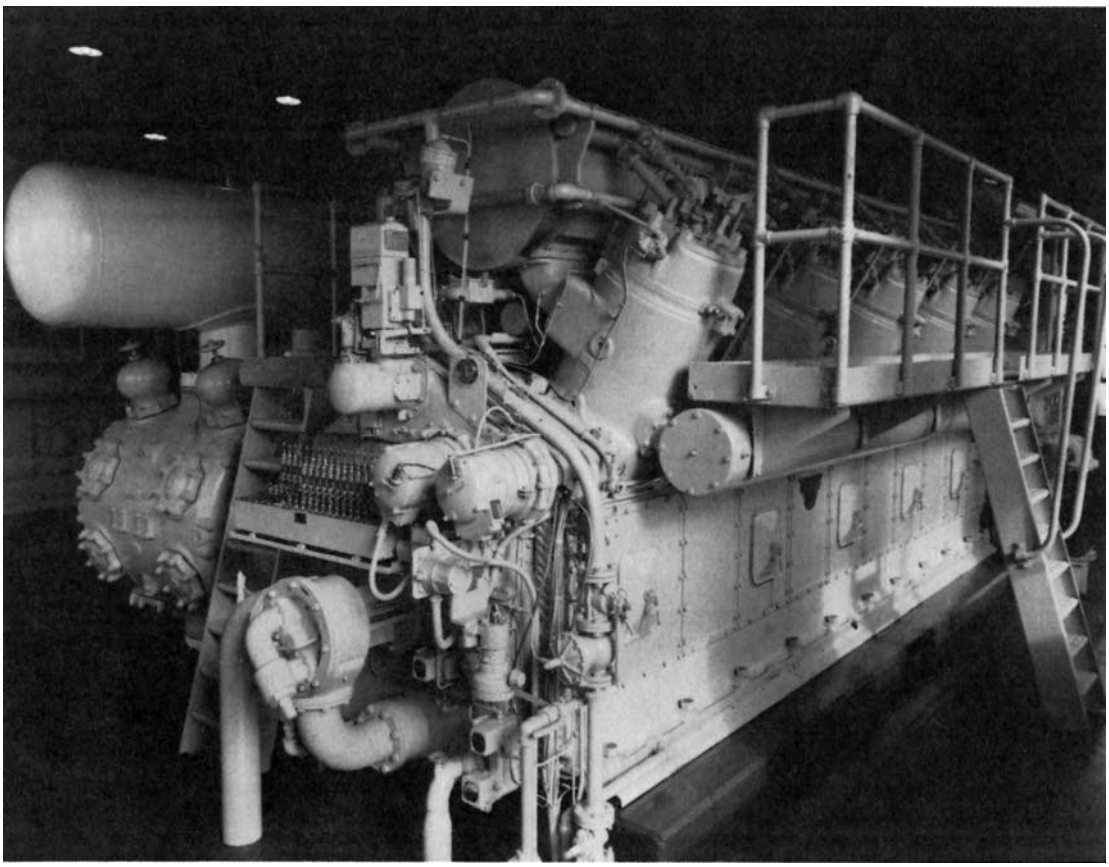


FIG. E-36 Large gas engine (ten cylinders) with built-on reciprocating compressor cylinders. (Source: Cooper-Bessemer Reciprocating.)

Turbocharged equipment uses exhaust gas to drive a blower that forces combustion air through a suitable heat exchanger into the intake manifold. This cooled, compressed air is used for combustion and, in certain engine types, "scavenging." Scavenging air purges exhaust gases from engine cylinders before combustion air is admitted.

Reciprocating gas engines suffer from a few disadvantages that must be considered when selecting process plant equipment:

- They require a fair amount of competent surveillance and routine maintenance. Minor overhauls are typically needed after about 2500 operating hours. Major overhauls should be anticipated every three to five years.
- Their low speed, typically in the 180 to 900 rpm range, requires stepup gears for such process duties as pump drives.
- Fluctuations in the heating value of the fuel may require constant adjustment of spark timing and could also require derating of the power output capability.
- The sulfur content of gas engine fuels may adversely influence the extent and frequency of maintenance required.

Gas engines typically consume between 6500 and 8000 Btu/BHP-hour (British thermal units/brake horsepower-hour).

Two-Stroke Gas Engines

It could be stated that gas engines are basically blown-up versions of the conventional automotive engine with just a few important modifications: they are slow-running, they use a gaseous fuel/air mixture instead of the liquid fuel/air mixture typically found in most automotive engines, and they are often integrally arranged (or combined) with the process gas compressor cylinders that they are driving.

Since the combustion process takes place inside the cylinder, gas engines belong to the family of internal combustion engines. Like their cousins the automotive gasoline and automotive diesel engines, gas engines are either of the two-stroke or of the four-stroke per cycle variety. Two-stroke engines have one power stroke for every full revolution of the crankshaft, whereas in four-stroke engines, only every second revolution is accompanied by a power stroke. Four-stroke engines have inlet and exhaust valves; two-stroke engines have inlet and exhaust ports. Each type of engine has a spark plug; the two-stroke engine also incorporates a fuel admission valve. Two-stroke engines are considerably more prevalent in the process industries.

Reference and Additional Reading

1. Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Environmental Accountability

Corporate accountability for releasing environmental hazards (or practicing other aspects of operation) that create, add to, or aggravate environmental hazards or damage to a greater extent than current BAT can help a company avoid or mitigate is termed *environmental accountability*.

Whether the company can be prosecuted or not for lack of environmental accountability depends on the perceived severity of the hazard in question. In many cases, a company's acceptance of environmental accountability is voluntary, as the following extracts from the 1997 annual report of the Kodak company in Rochester, N.Y., indicate. Apart from community-service value is the inestimable revenue that results from increased goodwill and positive publicity. See Table E-6 and Figs. E-37 and E-38.

Case Study 1: Environmental Programs Conducted by Major Film Manufacturing Corporation, Kodak*

Kodak has invested $15 million in new air scrubbing technology at one of its key film manufacturing buildings at Kodak Park in Rochester. The new technology, called a regenerative thermal oxidizer, can oxidize solvents with up to 99 percent efficiency. Consequently, the company expects an 86 percent reduction in solvent emissions from this building in 1998, exceeding the level of performance required by EPA regulations. Even though film production is expected to double at that building by 2000, the solvent emissions will still be less than 65 percent of 1997 levels.

In operation, solvent-laden air in the building is pulled through a heat exchanger, into the oxidizer, and out through an exit heat exchanger. The gases are then scrubbed before being released through a 100-ft stack.

An EPA-Kodak pilot project completed in 1997 shows that it is possible to predict

* Source: Kodak, USA. Adapted with permission.

TABLE E-6 TQM and Environmental Accountability

Sectors/Divisions	Quality Management System	Environmental Managements System
Cultor Food Science		
Xyrofin		
Kotka, Finland	ISO 9002/1994	ISO 14001/certification target 1999
Terre Haute, U.S.	ISO 9002/certification target 1998	Documented procedures partly in place
Thomson, U.S.	ISO 9002/certification target 1998	Documented procedures partly in place
Flavor		
C.A.L., France	ISO 9002/certification target 1998	—
Flavoring, Sweden	ISO 9001/1995	Documented procedures partly in place
Protectants[1)]	—	—
Cultor Feed Ingredients		
Enzyme Operations	ISO 9001/1994	ISO 14001/certification target 1999[2)]
Betaine Operations	ISO 9002/1997	ISO 14001/certification target 1999[2)]
Pacific Protein	ISO 9003/1994	—
Cultor Baking		
Vaasan Baking		
Frozen Bakery Products	ISO 9001/certification target 1998	
Local bakeries	—	—
Industrial Baking	ISO 9001/certification target 1998	ISO 14001/certification target 1998
Vaasamills[3)]	ISO 9001/1995	ISO 14001 certificate 1997
Cultor Nutrition		
Suomen Rehu	ISO 9001/1994	ISO 14001 certificate 1996
SIA Baltic Feed	ISO 9001/certification target 1998	—
Teknosan/Agrivit	ISO 9001/1997	ISO 14001/certification target 1998
Finnsugar		
Finnsugar	ISO 9002/1995	ISO 14001/certification target 1998
Sucros	ISO 9002/1996	ISO 14001/certification target 1998
Sugar Beet Research Centre	ISO 9001/1997	ISO 14001 certificate 1998
Neson	ISO 9002/1994	Documented procedures partly in place
EWOS Group	ISO 9001/multisite certification target 1998[4)]	ISO 14001/multisite certification target 1999[4)]
Canada	ISO 9003/1992	
Scotland	ISO 9002/1993	
Genencor International		
Hanko, Finland	ISO 9002/1994	ISO 14001/certification target 1999
Jämsänkoski, Finland	ISO 9002/1994	—
Brugge, Belgium	ISO 9002/1994	—
Cedar Rapids, U.S.	ISO 9002/1995	—

Note the integrated status of the implementation of documented quality and environmental management systems.
1) Products are toll-manufactured by major chemical companies, which have their own documented quality and environmental systems in place. These are audited regularly by Cultor Food Science.
2) ISO 14001 certificate covering both betaine and enzyme operations.
3) Part of Industrial Baking since 1998.
4) Multi-site certification including EWOS R&D and the division's companies in Chile, Canada, Norway, and Scotland.
SOURCE: Cultor.

a chemical's potential health and environmental effects so early in development that hazards and waste can be avoided and significant dollars can be saved.

In the two-year effort to evaluate the EPA-developed test method, Kodak was able to reformulate five photochemicals under development. This improved their environmental performance significantly.

EPA's test method, called the Pollution Prevention Assessment Framework, uses computer software, mathematical formulas, and tables to predict a chemical's risk-related properties based on its molecular structure.

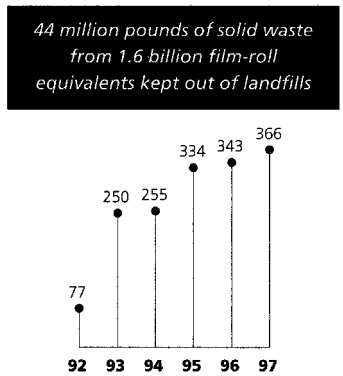


FIG. E-37 Film container recycling program to recycle steel magazines, plastic spools, cartridges, and canisters. (In millions of rolls of film.) (Source: Kodak.)

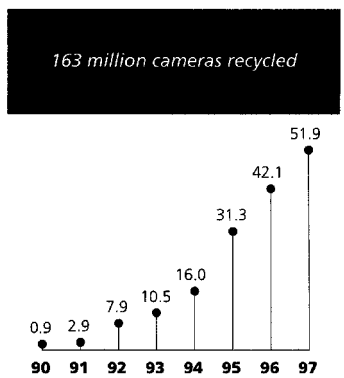


FIG. E-38 Worldwide single-use camera collection and recycling. (In millions of cameras recycled.) (Source: Kodak.)

The project not only determined that the method was effective, but that it was transferable, meaning other companies could apply it to their chemicals in the early stages of development.

European companies generally take environmental accountability far more seriously. For instance, the Sulzer conglomerate, which makes products ranging from turbines to pumps to locomotives to fabric-weaving machines and medical products, is allowing its environmental proactivity to permeate its subsidiaries. It is highly likely that corporations with Sulzer's business acumen and global presence would not take on activity of this magnitude just for goodwill reasons. Sulzer's integration of environmental accountability with its quality control program (total quality management) indicates that these measures are fundamental to its operations.

Environmental accountability takes on particular significance in Scandinavian countries, whose history indicates they practiced this method of total involvement with sustaining their environment long before there were definitions in any language for it. Their reporting parameters support this fact, as the following extracts from the Cultor corporation's annual report indicate.*

* Source: Cultor, Finland. Adapted with permission.

Case Study 2: Cultor

This corporation has adopted a sophisticated, challenging work program based on sustainable development. Environmental concerns are balanced with safety, health, and environmental (SHE) considerations.

The findings in this case study are based on visits to six sites* in Finland, Norway, and the U.S., interviews with management at divisional and corporate level, and review of drafts of this report and supporting data. Cultor has also sought evidence that the trends in the data over a three-year period (1994–1997) are due to changes in performance, rather than changes in the way data are measured or calculated.

Sustainable development findings

Since 1996, "sustainable development" has been a core value along with customer concerns, corporate competence, and shareholder value. Progress has been made in some divisions to integrate sustainable development into strategic business development. See Table E-7. Life-cycle inventories (LCIs) are used to help divisions focus on their key environmental issues.

An area of particular public interest is the relationship between Cultor's gene technology and sustainable development. Cultor's innovative response has been to enter into an agreement with WWF (World Wildlife Fund) which aims, in part, to clarify this relationship.

Much progress has been made by the environmental team. There is now growing recognition at the divisional and plant level that the team is driving a broad range of issues that must be addressed at all levels of the corporation. See Tables E-6 through E-9. Regular meetings of, and informal communication within, the corporation help to address communication lapses between Cultor and its sites.

Glossary of Terms and Abbreviations for Case Study 2

Antinutritive factors	Any dietary component that can impair an animal's utilization of a potentially useful dietary nutrient.
Auditing	A systematic and independent examination to determine whether quality and environmental activities and related results comply with planned arrangements and to ensure that these arrangements are implemented effectively and are compatible with achieving objectives.
Betaine	A nitrogen-containing compound found in sugar beet and other plants.
Bioconversion	A chemical conversion process based on biological catalysis (i.e., enzymes). An example of this type of process is the conversion of glucose to fructose using the glucose isomerase enzyme.
Biofilter	A unit in which the volatile organic compounds in exhaust gases generated by production processes are broken down by micro-organisms into carbon dioxide, water, and nitrogen.
BOD_7/BOD_5	Biological oxygen demand. The amount of oxygen needed by micro-organisms to break down organic matter in effluent, usually measured over a period of seven or five days under standard conditions.
Enzyme	A protein molecule produced by living cells that is able to accelerate biochemical reactions.

* EWOS, Florø, Norway; CFI Vaasa, Finland; Joutsenolainen Bakery Joutseno, Finland; Sucros Salo, Finland; Suomen Rehu Vassa, Finland; CFS Thomson, USA.

Erythrobates	Antioxidants that act as oxygen scavengers, removing oxygen from food products.
Extraction	An operation that uses liquids and other fluids (i.e., extractants) to recover chemical compounds from a solid matrix, such as sugar beet, or from another liquid.
Fermentation	A process that uses the growth of microorganisms to produce cellular material, such as enzymes and vitamins. Fermentation can also be used to convert complex organic compounds into simpler substances. An example of the latter is the use of yeast to convert sugar into carbon dioxide and alcohol.
Functional food	A food is said to be functional if it contains a food component, either a nutrient or non-nutrient, that affects one or more functions in the body in a targeted way and has a positive effect in line with the claims made for it.
Genetic modification	A new branch of biotechnology that allows heritable material within an organism to be modified in a specific and controlled way.
Genetically modified microorganism (GMM)	A microorganism that has been modified using genetic engineering. Microorganisms of this type include bacteria, molds, yeasts, protozoa, and algae.
Genetically modified organism (GMO)	An organism that has been modified using genetic engineering, such as plants, microbes, and animals.
ISO	International Organization for Standardization. The ISO standards on quality management (ISO 9000 series) and environmental management (ISO 14001) have been widely adopted in industry.
Life-cycle assessment (LCA)	A study that compiles and evaluates the inputs, outputs, and environmental impact of a product throughout its entire life cycle.
Life-cycle inventory (LCI)	Part of an LCA comprising a summary-style compilation of the inputs, outputs, and environmental impact of a product throughout its entire life cycle.
Modern biotechnology	The use of living organisms, cells, or enzymes to produce foods, medicines, and other products. Molecular biological tools, such as genetic engineering, play an important role in modern biotechnology.
Organic production	A type of farming based around significantly reducing the use of fertilizers and pesticides that may have detrimental effects on the environment or result in the presence of residues in agricultural produce.
Regulatory affairs	A function within a company responsible for ensuring that its business activities are compatible with legislation and ethics relating to quality, safety, and consumer protection.
Self-assessment	A systematic approach for identifying strengths and development areas and for targeting resources to key improvement areas critical for business success. Self-assessment tools are used for measuring performance against a wide range of key indicators, covering areas such as customer satisfaction, products, services, operations, and finance. A number of national self-assessment criteria have been developed, such as the Malcolm Baldrige award, to provide a framework for performance excellence.
Sex hormone-related diseases	Diseases such as breast cancer and prostate cancer.
Verification	A process making use of examination and objective evidence to confirm that specified requirements have been fulfilled.

Risk management

Minimizing risk of incidents and emergency events is an important part of environmental management systems. Data on environmental incidents was collected for the first time for Cultor's environmental database in 1997. The most significant events are reported in Table E-8.

Feeding management

Environmentally advanced feeding management is used as a tool to reduce the environmental loading of livestock farming by controlling nitrogen and phosphorus emissions. Suomen Rehu (a corporate division) has developed feeding programs for poultry and pigs that can reduce phosphorus emissions by one-third. These

TABLE E-7 Environmental Performance

Business sectors		Energy consumption: Heat and steam TJ	Energy consumption: Electricity TJ	Air releases[1]: SO_2 t	Air releases[1]: CO_2 1,000 t	Waste[2]: Sent to landfill t	Waste[2]: Incineration/ destruction t	Waste[2]: Sent for recycled/ recovered t	Packaging usage[3]: Total t	Water[4]: Consumption 1,000 m^3	Waste water[5]: Volume 1,000 m^3	Waste water[5]: COD_{Cr} t	Waste water[5]: BOD_7 t	Waste water[5]: Total P t	Waste water[5]: Total N t
Cultor Food Science[6]															
Xyrofin	1996	1,023.2	295.6	139	85.8	x[7]	x	x	x	3,822	3,616	1,393.0	x	6.3	4.5
	1997	1,042.7	317.1	101	87.5	3,030	5	90,100	1,160	3,736	3,767	781.0	x	6.8	4.5
Flavor	1996	23.6	7.5	4	1.6	x	x	x	x	38	24	x	x	x	x
	1997	16.1	8.2	2	1.1	870	30	10	130	43	43	x	x	x	x
Cultor Feed Ingredients[8]															
	1996	604.4	49.6	230	59.0	x	x	x	x	318	595	x	x	x	x
	1997	642.2	55.5	347	70.2	900	35	40	370	350	728	x	x	x	x
Cultor Baking[9]															
Vaasamills	1996	31.0	98.2	10	2.6	x	x	x	x	33	18	x	x	x	x
	1997	33.3	96.5	10	2.8	670	1	940	1,680	37	24	x	x	x	x
Vaasan Bakeries															
	1996	409.5	165.9	9	25.2	x	x	x	x	294	x	x	x	x	x
	1997	395.4	176.3	9	26.6	x	x	x	1,580	253	x	x	x	x	x
Cultor Nutrition															
Finnsugar[10]	1996	1,548.4	242.6	1,133	252.2	x	x	x	x	1,643	1,771	142.4	20.3	1.6	28.4
	1997	1,792.4	265.7	1,298	277.6	980	95	241,400	2,030	1,740	2,044	159.1	26.8	2.1	34.1
Suomen Rehu[11]															
	1996	193.8	146.6	75	15.2	x	x	x	x	87	x	x	x	x	x
	1997	195.1	149.0	65	16.4	1,110	140	160	1,330	82	x	x	x	x	x
EWOS[12]	1996	235.0	76.3	25	18.6	x	x	x	x	304	x	x	x	x	x
	1997	320.5	109.7	46	26.1	1,070	140	3,840	1,240	433	x	x	x	x	x

1. Releases of sold energy (850 TJ) are included, but releases of purchased electricity (1000 TJ) are excluded.
2. Only 1997 waste data is reported, since 1996 data is not reliable. Waste storaged (7600 t) at the site is not included. In addition, figures cover soil and stones carried with sugar beet.
3. Only 1997 packaging amounts are reported (pallets and reusable packages not included), since 1996 data is not reliable.
4. Noncontact cooling water not included.
5. COD_{Cr}, BOD_7, total P, and total N reported only for sites with waste water treatment plants, except CFI.
6. The figures cover the following CFS Xyrofin sites: Kotka, Thomson, Terre Haute. The figures do not include the Lenzing site (production started in 1997). The figures for the Flavor Division cover Flavoring AB sites in Sweden, Flavor Technology Corporation in East Windsor, and C.A.L. in France.
7. X indicates no data available or data unreliable.
8. The figures cover both enzyme operations in Vaasa and betaine operations in Naantali, Finland, as well as Pacific Protein in Chile.
9. The Vaasamills figures include Siljans Knäcke AB in Sweden and the Kotka site in Finland. Vaasan Baking figures include 27 bakeries/production units in Finland and operations in the Baltic countries.
10. The figures include the sites in Porkkala, Salo, Säkylä, Turenki and Jokioinen in Finland.
11. The figures include the Suomen Rehu sites at Kotka, Seinäjoki, Turku and Vaasa in Finland, Teknosan AB in Sweden, and SIA Baltic Feed in Latvia.
12. The figures include the sites in Norway, Scotland, Canada, and Chile (since 1997).

TABLE E-8 The Most Significant Environmental Incidents at Business Units between 1995 and 1997

Cultor Business Sector/Division	Accident or Leakage
CFS Xyrofin, Kotka, Finland	A discharge valve of a caustic (NaOH) storage tank was accidentally left open in summer 1996, causing a leak of 22.5 tonnes of caustic soda into the sea. A capital project for a new storage tank and distribution system was undertaken as corrective action.
CFS Xyrofin, Kotka, Finland	In summer 1997, wastewater was discharged into the sea for seven days (approximately 120 hours in total) due to pretreatment, electricity, and wastewater treatment system failures.
CFS Flavor Technology Corporation, East Windsor, U.S.	A package containing ethanol-based flavoring products was shipped improperly from the facility in summer 1997. Penalties of FIM 52,000 have been paid. This led to changes in risk management in the shipping of hazardous materials
Cultor Baking, Vaasan Baking, Nelo Bakery, Finland	An oil leakage took place in summer 1996 as a result of old pipelines and corroded valves. The soil was treated immediately in cooperation with the authorities.
Cultor Nutrition Finnsugar, Kantvik, Finland	Fire damage in summer 1997. No damage to the environment or production resulted.
Cultor Nutrition EWOS, Scotland	1) Land near the factory was previously used for landfill between 1993 and 1996. Landscaping and clean-up was carried out during 1997. Clean-up costs totalled FIM 640,000. 2) Fish oil leakage into a stream took place in summer 1996. A total of FIM 3600 in fines were paid. 3) Fish oil leakage into a stream in late 1996 and early 1997. The case was treated immediately in cooperation with the authorities. No legal action was taken. 4) Fuel oil leakage in summer 1997 as a result of the failure of old pipelines. Clean-up costs totalled FIM 210,000. A total of FIM 18,000 fines were paid. The wastewater system mentioned on Page 29 in this report will reduce the risk of contamination of the local water system from taking place in the future.

programs focus on raw material selection and a more precise phosphorus allocation, and the addition of phytase enzyme to improve phosphorus utilization.

Suomen Rehu has been able to reduce the amount of added phosphorus by almost 700 t/a in poultry and pig diets between 1995 and 1997. This enables the reduction of the phosphorus content of manure by 30 percent, which in turn promotes environmentally compatible livestock production and enables farmers to increase the number of animals they keep.

Feeding management can also be utilized in developing poultry diets without any added growth promoter substances with an antibiotic effect; and pork diets without any added growth promoters. As a result of these types of feeding programs, the proportion of feeds produced by Suomen Rehu without such additives has steadily increased. In 1997, 85 percent of the pigs fed with Suomen Rehu feeds received diets without any added growth promoters.

ISO-VILJA™ technology, developed by Suomen Rehu, is a commercial quality and environmentally oriented cultivation concept. Targeted nutritional content and feeding value, controlled quality with respect to product hygiene and residues, and traceability from cultivation to transportation and the feed factory are the most significant features of ISO-VILJA™ technology.

ISO-VILJA™ is used by Suomen Rehu to enhance the quality of raw materials used for feed manufacturing. A transparent, documented quality chain for animal production and livestock-based foodstuffs is employed.

The benefits of ISO-VILJA™ technology to the farmer include better profitability as a result of higher yields and significant savings in fertilizers. In addition, residual nitrogen in the soil can be reduced by up to 60 percent. In 1997, 1600 farmers adopted this cultivation method, 3000 farmers received training on the

TABLE E-9 Environmental Report 1995 Objectives

	Progress
Cultor Feed Ingredients	
■ To reduce odor generated by betaine production at the Naantali plant.	■ A study was initiated in 1997 to explore suitable biological methods for an odor treatment investment. This will be completed in mid-1998.
■ To implement documented quality and environmental management systems.	■ Enzyme operations received ISO 9001 certification in 1994, and Betaine operations ISO 9002 certification in 1997. The certification target for ISO 14001 covering both operations was 1999.
■ To reduce effluent loadings and improve energy efficiency at Pacific Protein.	■ An investment has been made in pretreatment of wastewater and heat recovery systems.
Cultor Baking	
■ To develop the concept of controlled farming.	
■ To improve health and safety at bakeries.	■ Investments in flour dust removal systems at various bakeries.
Cultor Nutrition	
FINNSUGAR DIVISION	
■ To further reduce the environmental impact of sugar beet cultivation. To incorporate environmental management in the quality management system.	■ Further work has been done, together with the Sugar Beet Research Centre. Goals have been set as part of implementing the ISO 14001 standard.
■ To finalize the LCI (Life Cycle Inventory) on sugar beet and its main products in 1995.	■ The LCI was completed in 1995.
■ To further reduce the environmental impact of drying beet pulp.	
■ To promote the recovery/recycling of packaging waste and increase the use of bulk transportation.	
EWOS	
■ To further reduce the environmental impact of fish farming.	■ The overall nitrogen loading associated with fish feed has been reduced by 80% over the last 20 years, and has now reached the level where further improvements will be difficult to obtain without affecting feed quality properties.
■ To develop alternative protein sources for fish feed production.	■ Work has been done on soya as an alternative protein source.
SUOMEN REHU	
■ To further improve the division's environmental management system and reinforce the image of Finnish food as clean and wholesome.	■ Suomen Rehu received ISO 14001 certification in 1996. Value network thinking and supplier auditing have been promoted. Improvements have also been made to feeding management systems, and ISO-VILJA™ technology has been introduced.
New objectives	
Implementation of Cultor's sustainable development value process	■ Develop Cultor's environmental database to better meet the needs of both the Group and divisions. Improve the reliability of the data collected through the database.
	■ Develop indicators for environmental, quality, regulatory, and ethical issues to measure performance; and conduct a pilot study to test the suitability of the indicators chosen.
Environment	■ Implement Cultor's new Quality and Environmental Policy in operations.
	■ Develop documented quality and environmental management systems for all of Cultor's major business units Divisions to set detailed targets and objectives.
	■ Continue to conduct LCIs for major products and use LCIs as an internal environmental management tool.
	■ Start monitoring transportation data at Group level.
	■ Improve the follow-up of H&S indicators at both Group and divisional level. Develop the data collected and definitions used.
Quality	■ Start using self-assessment as a Group-wide tool for continuous improvement.
	■ Continue internal benchmarking for process improvements and start external benchmarking.
	■ Start divisional cross-auditing.
Regulatory and ethical issues	■ Implement the new Regulatory Policy in divisional operations. Create standard operational procedures based on the policy and set goals for Cultor's regulatory work.
	■ Implement the Animal Trial Policy in divisional operations.
	■ Implement the Modern Biotechnology Position Paper in divisional operations.
	■ Continue proactive dialogue with stakeholders, particularly in the area of modern biotechnology.



FIG. E-39 Taking a chlorophyll measurement on one of the farms using Suomen Rehu's ISO-VILJA™ advanced cereal farming concept. (Source: Cultor.)

technology, and 14,000 technical leaflets were distributed. (See Figs. E-39 through E-41.)

Environmental reports

Objectives and targets. Table E-9 describes the progress that Cultor has made in implementing the objectives detailed in its 1995 Environmental Report. New objectives are also given, linked to four key areas: implementation of Cultor's sustainable development value process, the environment, quality, and regulatory and ethical issues. The objectives and corporate target objectives are updated annually.

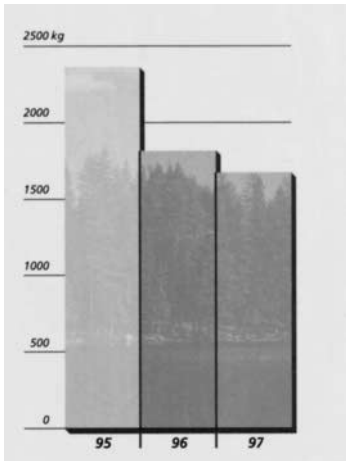


FIG. E-40 Amount of inorganic phosphorus used at Suomen Rehu. (Source: Cultor.)

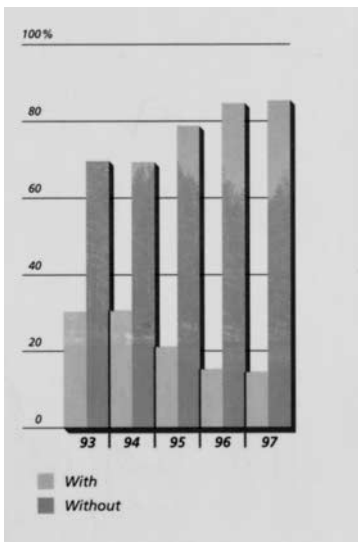


FIG. E-41 Pig feed production with and without growth promoters. (Source: Cultor.)

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.

Environmental Air Monitoring (*see* Emissions)

Environmental Economics*

Environmental resources and effects are difficult to quantify using the usual economic terms and definitions. "Environmental economics" includes special

*Source: AssiDomän, Sweden. Adapted with permission.

additional costs and revenues generated by environmental measures, whether compulsory or voluntary. A discussion of the term follows with specific reference to the information source's corporate policy.

Yardsticks of Environmental Economics (Reference Corporation: AssiDomän)

Biological diversity

External, independent certification of forestry preserves biological diversity at the same time as active forestry is pursued. By means of the certification process, the market can complement and hasten necessary legislation.

The development and introduction of new ecological forestry methods and the certification process are viewed as investments in the future by companies that practice them. The additional costs of these efforts within AssiDomän in 1995, 1996, and 1997 have been estimated at MSEK (millions of Swedish kronor) 100, 120, and 150, respectively. As the new methods gradually become the accepted norm, the actual voluntary additional costs are estimated at half this amount.

Certification raises the value of forest assets, and certified products are expected to yield additional revenues amounting to several tens of millions of Swedish kronor for AssiDomän over many years.

These measures contribute strongly to improving global image as a proactive and leading force in the environmental field. This is a particularly strong corporate objective in western European countries, such as Sweden. In AssiDomän's case, much of the corporation's holdings and the head office are in Sweden. However, it has assets elsewhere, notably Germany.

Reduced emissions

Environmental measures to reduce emissions from the plants are undertaken in response to local legislation, license conditions, environmental charges, and taxes, and as voluntary investments. The Swedish legislation and licensing procedure requires the best available technology within reasonable economic limits—an effective stimulus to advances in environmental technology. Within AssiDomän, efforts are being concentrated on reducing discharges of oxygen-demanding substances to water, reducing emissions of acidifying sulfur and nitrogen oxides to air, and energy conservation measures. Large environmental investments are often undertaken as part of other major capital investments, particularly at pulp and paper mills. The environmental capital cost is between 10 and 25 percent of the project (around MSEK 300 in 1997 for AssiDomän). Environmental charges and taxes amounted to approximately MSEK 50.

Note that some of these taxes are specific to individual countries. They are not yet globally accepted practice. For instance, Sweden charges NO_x and SO_x taxes; as of 2001, the United States does not.

Roughly 10–20 percent of the environmental investments can be regarded as voluntary. In some cases they can yield direct additional revenues for products with an environmental profile (goodwill investment, enhancing the company's image, name, and trademarks). See also Table E-10.

Some examples:

- The new biofuel boiler in Frövi reduces the oil requirement, permits increased biofuel use, and is projected to become increasingly profitable as energy taxes rise.
- Ash restoration: Recovering the ashes from the wood-processing mills, treating them and returning them to suitable soils will eventually be necessary due to the

TABLE E-10 Some Environmental Key Ratios

Type of Activity	Purchased Electricity (kWh/SEK[1]) 1997	Purchased Electricity (kWh/SEK[1]) 1995	Fossil Fuels (kWh/SEK[1]) 1997	Fossil Fuels (kWh/SEK[1]) 1995
AssiDomän's five pulp and paper mills in Sweden	0.88	0.39	0.89	0.37
Industry, pulp and paper mills	—[2]	0.46[3]	—[2]	0.27[3]
AssiDomän's corrugated board and sack production[4]	0.06	—[2]	0.13	—[2]

1. Value added is the sum of operating profit after depreciation and payroll expenses.
2. No data available.
3. Energy consumption: source—ÅF-IPK, *Energy Consumption in the Pulp and Paper Industry*, 1994 (in Swedish). Value added: source—SCB, *Industrial Statistics*, 1995.
4. Applies to all plants in the Group.

NOTE: It is AssiDomän's ambition to contribute to the development of environmental key ratios that can describe how the environmental work affects resource use and financial position. Since energy use is strongly environment-related and furthermore an important cost item for the forest products industry, AssiDomän's energy use is reported here in relation to value added. This reflects both the energy intensity of the company's own activities, and sensitivity to energy price changes.

The difference between 1995 and 1997 is explained for the most part by the fact that the price level for the end products, and thereby the value added, was considerably higher in 1995. It is of interest to note that energy use in relation to value added is nearly 10 times lower in corrugated board and sack manufacture than in the pulp and paper mills. The table also shows that AssiDomän is at roughly the same level as the rest of the Swedish pulp and paper industry.

Energy price sensitivity is shown by the fact that when converted to energy costs, the expenditures for electricity and fossil fuels correspond to 10–20 percent of the value added.

threats of soil acidification and nutrient deficiency in combination with future waste requirements and charges. Several trials are under way.

- Reduced sulfur content in ship fuels: This voluntary measure incurred a short-term additional cost for AssiDomän, but was more environmentally cost-effective than alternative measures to reduce sulfur emissions. It influenced the decision of authorities regarding environmental charges for shipping.

Research and development on cleaner and more energy-efficient processes

Research and development investments in the group amounted to around MSEK 210 in 1997. It is estimated that approximately MSEK 65 of these have a direct or indirect link to the environment. These investments include the projects concerning bleaching, air and water pollution control, and "ecocycles" that are for the most part being pursued on a joint sectoral basis.

A unique AssiDomän project in cooperation with suppliers is black liquor gasification, where a demonstration plant is planned at AssiDomän Kraftliner in Piteå. This new technology is expected to become a breakthrough that will provide higher energy efficiency and twice the production of electrical energy from biofuel. The project has been granted state energy subsidies amounting to half of the construction and experimentation costs totaling MSEK 475.

Development of resource-efficient packaging

Development of more resource-efficient, lighter-weight, and more transport-efficient packaging is not just environmentally, but also often economically desirable. (See Fig. E-42.) Life-cycle assessments (LCAs)* as a basis for the development work are an important tool here. The potential for commercial exploitation of the environmental benefits is greater if these benefits can be promoted as part of a new product or packaging solution.

Examples of such projects within AssiDomän are:

- A stronger sack paper that enables the density of the paper to be reduced by more than 10 percent for certain applications. See Fig. E-42.
- The launch of the board Frövi Light. An example: 25 percent reduced board weight for a frozen food pack in Germany resulted in fewer and lighter-weight shipments plus MSEK 0.7 in reduced packaging charges for the customer.
- New barrier-coated papers that replace aluminum-foiled papers, which are difficult to recycle.
- Continued promotion of "eco-white" kraftliner, based on a new process for significantly more efficient wood and fiber utilization plus totally chlorine-free bleaching. Thanks to the improved environmental profile, it has been possible to speed up the market introduction, resulting in additional revenues of several tens of millions of Swedish kronor.

Within the corrugated packaging business, a design and material optimization program has been developed and has resulted in the launch of several new types of packaging with both reduced material consumption and lower costs. One example is a tray for detergent. The improved environmental profile convinced the German manufacturer Henkel to choose this new packaging solution, permitting a 28 percent weight reduction.

Environmental management systems (EMS)

The total costs of work with EMS in the Group during 1997 is estimated to be about MSEK 30. EMAS registration or ISO certification provides a strategic advantage for customer relations, but is not expected to yield direct additional revenues. However, it is noted that setting environmental objectives and plans entailed by certification/registration can in the long run yield significant efficiency gains that clearly outweigh the costs. See Table E-10.

Analysis of potential environmental debts is a part of ongoing EMS work. The costs of rectifying environmental debts known today are estimated to be less than MSEK 10.

Consequences for Long-Term Profitability

Specific environmental issues in AssiDomän are:

Conservation of the world's forests

Widespread demands for increased protection of the world's forests, including old-growth forest in the northern coniferous forest belt, have led to restrictions and

---

* The term *life-cycle assessment* is used here with a different connotation from that of the entry "LCA," which concerns LCA of turbomachinery components.

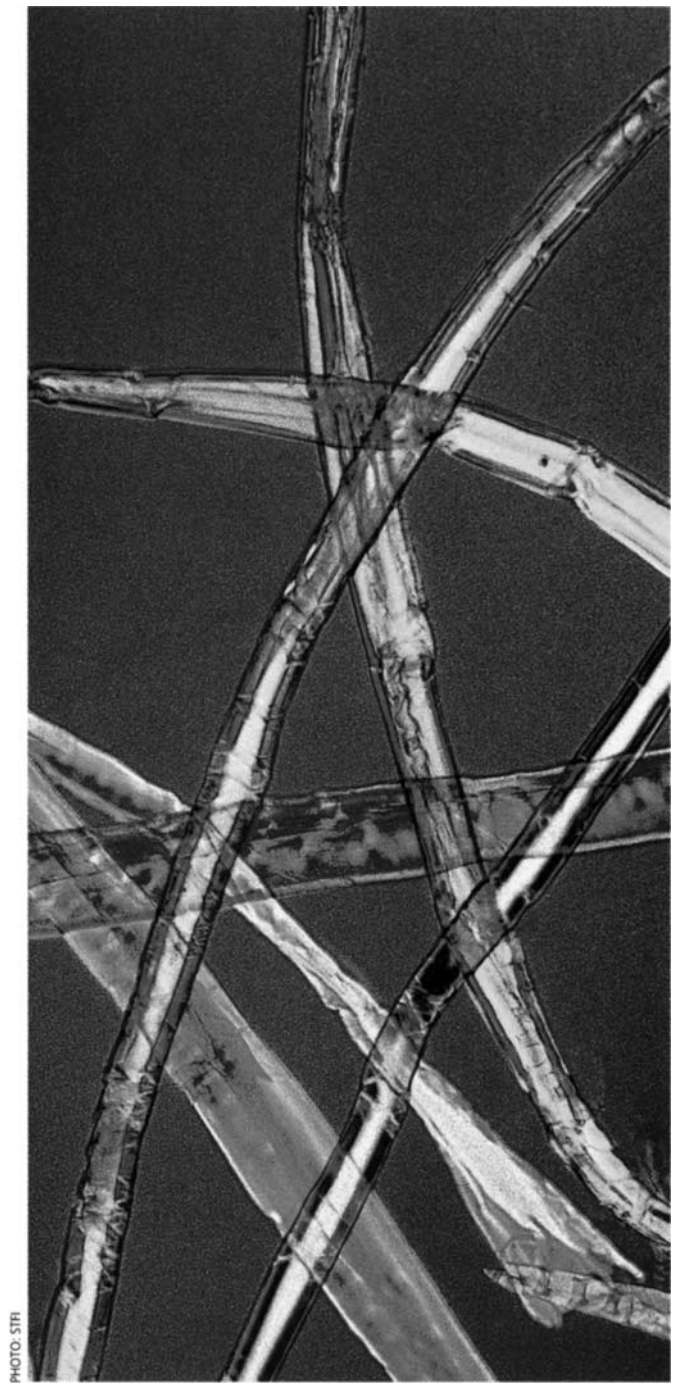


FIG. E-42 Long, strong fibers are a prerequisite for resource-efficient manufacture of strong packaging. The photo shows spruce fibers at a magnification of 100 times. (Source: AssiDomän.)

thereby some scarcity of high-quality softwood timber. AssiDomän's vast holdings of forest land are one of the group's most important assets in this context, and certification of these holdings further enhances their value.

Emission requirements and environmental charges

Completed environmental investments have given AssiDomän and most other Nordic pulp and paper mills a lead over many international competitors—a lead which can often amount to several hundred million Swedish kronor in investments per mill. In Sweden, environmental charges and taxes on, for example, acidifying sulfur and nitrogen oxides, as well as on climate-warming carbon dioxide, are used as one of the policy instruments for bringing about environmental improvements. Such economic instruments are now being adopted to an increasing extent internationally—a trend that will benefit the Swedish forest products industry.

Legislation concerning packaging, waste paper, and waste

A few years ago, packaging based on new fiber appeared to be seriously threatened by planned legislation within the EU in particular. Waste paper and various waste paper systems have been favored legislatively. The negative consequences have not materialized, however, since the legislation has become less radical and adjustments have been made in the marketplace. AssiDomän has increased its usage of waste paper where this has proved economically and environmentally feasible. The drive toward increased standardization both in Europe and globally entails both advantages and disadvantages. All things considered, however, this trend is viewed more as an opportunity than as a threat.

Ecolabeling

The ecolabeling of sawn timber is viewed by AssiDomän as economically advantageous since most of AssiDomän's forestry operations have been FSC-certified. There is still very little ecolabeling of packaging, compared with consumer goods. Ecolabeling is not expected to affect profitability.

Additional Costs, Additional Revenues, and Goodwill

The annual additional costs for environmental investments and other environmental measures that AssiDomän undertakes as strategic, voluntary investments are estimated at MSEK 120–130. The life of these investments can vary widely, however, since the additional values to which they give rise sometimes have a short duration (environmental promotion, ecolabeling) of one or two years. Forest certification, corporate image, trademarks, and customer alliances are examples of goodwill investments with considerably longer lives.

Industry Comparisons

AssiDomän compares favorably with its Nordic competitors in environmental concerns, particularly with regard to forestry practices and the introduction of EMS. The generally high level of environmental compatibility in Nordic companies gives the Swedish forest product industry a lead over many international competitors in,

for example, North America, who face considerable environmental investments in the years to come.

Two important aspects of the economics of process engineering are environmental ecolabeling and environmental LCA, which will be discussed below. People in these industries frequently shorten these terms to "ecolabeling" and "LCA." LCA is a significant term in the operation of machinery as well; in that sense it will be discussed later in this book (under the "L" entries), as the technical definition is different when dealing with high-speed rotating machinery (versus forestry). The two usages of the term should not be confused.

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.

Environmental Ecolabeling*

The purpose of ecolabeling is to distinguish products that have the least environmental impact. The criteria for ecolabeling are regularly tightened to stimulate the development of increasingly environmentally sound products. Criteria are based on the total environmental impact of the product throughout its life cycle. Requirements are also made on the function and performance of the product.

There are several types of ecolabeling, all of which are the object of standardization within the ISO, the International Organization for Standardization. AssiDomän is participating in the Swedish standardization work by offering viewpoints on the formulation of standards.

Within the Nordic Swan ecolabeling system, criteria are developed for packaging paper and sawn timber products. Representatives of AssiDomän are participating in this work. This type of labeling consists of a symbol intended to guide the consumer in making environmentally informed purchases.

Environmental product declaration is another type of ecolabeling where a number of environmental parameters are declared, but without being evaluated. The primary target group is large purchasers within industry and government. The need for environmental product declarations has increased with the introduction of EMS, which require environmental information from suppliers and contractors. See Table F-4.

Environmental Life-Cycle Assessment*

LCA is a method for describing the total environmental impact caused by a product "from cradle to grave," i.e., from extraction of the raw materials to final waste disposal. See Fig. E-43.

After the scope of the assessment has been defined, environmental data are gathered from all links in the production chain. This phase is called the inventory and results in a body of data regarding resource and energy use, emissions to air and water, and waste. Based on the collected data, an evaluation can then be made of total environmental impact. This is called *environmental impact assessment*. LCA

*Source: AssiDomän, Sweden. Adapted with permission.

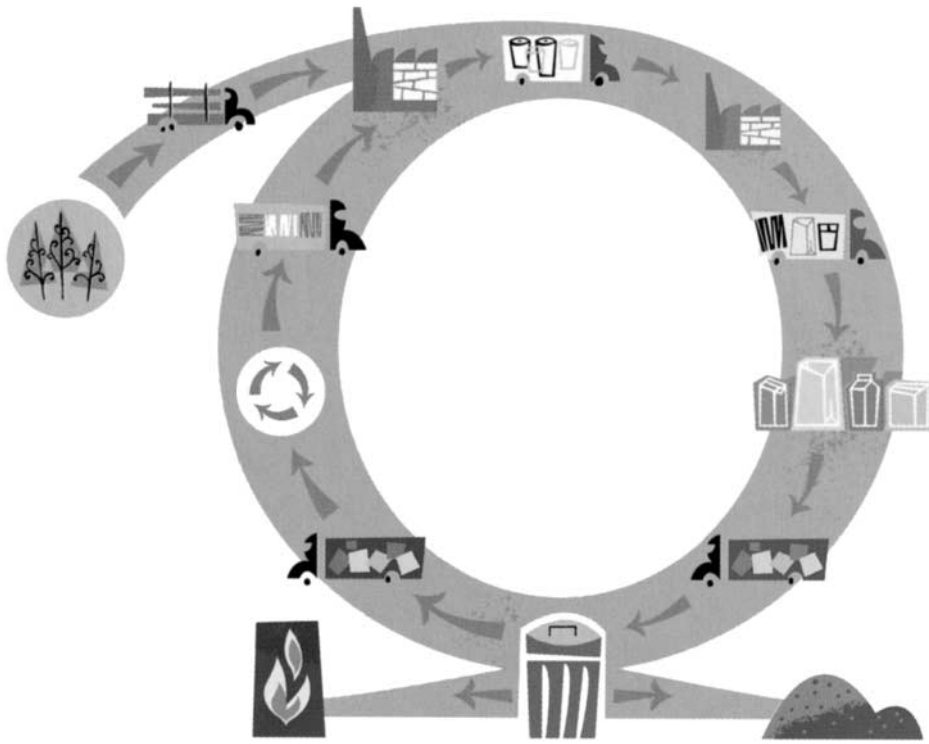


FIG. E-43 Environmental life-cycle assessment. (Source: AssiDomän.)

is used as a guide to the prioritization of remedial measures. The method is also used in product development for comparing alternative materials from an environmental point of view.

Evaporative Coolers (*see* Chillers; Coolers, Dairy)

Exhaust Stacks (*see* Stacks)

Exhausters, Centrifugal Gas*

Gas exhausters of the geared type have reduced overall dimensions, weights, and high-performance specifications. (See Fig. E-44.)

Machines of this type are used in geothermal plants where the turbine inlet steam has a high noncondensable gas content. See Table E-10 for a list of example applications.

The choice of gas exhauster is dictated both by the noncondensable gas content and the steam flow rate in the turbogenerator set.

This means that installing a centrifugal gas exhauster may prove economically advantageous even for low percentages of gas if the steam flow rate is sufficiently high. See Table E-11.

The main advantages offered by gas exhausters over alternative solutions are excellent noncondensable gas extraction process efficiency and the high levels of

* Source: Ansaldo, Italy. Adapted with permission.



FIG. E-44 Compressor set for Castelnuovo geothermal power plant (Italy). (Source: Ansaldo.)

vacuum that can be achieved. This accounts for power savings of between 40 and 60 percent with respect to other technical solutions such as steam driven ejectors. Centrifugal gas exhausters are designed to offer long term in-service reliability, high performance, and simplicity of design.

These design goals have led to the development of centrifugal impellers milled from a single piece, or welded together to form a single block, thus improving resistance to stress and corrosion. Advanced three-dimensional wheels are used to achieve efficiency figures close to the values that characterize axial compressors, but with a reduced risk of fouling. Particular attention has been given to heat exchange during interstage cooling, achieved by direct contact and dehumidification of the cooled gas.

Expansion Joints*

Expansion joints have been installed on the outlet flange of gas turbine exhausts since turbines were first used in applications other than to fly planes. An expansion joint is required to isolate the delicately balanced turbine from the thermally expanding and vibrating ductwork system.

Recent developments in turbine technology have put ever-increasing demands on the expansion joint supplier to accommodate more and more movements at higher temperatures.

Though financial consideration is always a factor, reliability and trouble-

* Source: Townson Expansion Joints, UK.

TABLE E-11 One OEM's Sample Applications List—Significant Installations

Customer Plant Country	No. of Units	Year	Compr. Type Fluid Handl	Capacity	Inlet Press (bar)	Press Ratio	Speed (rpm)	Power (kW) Drive (1)
Acrylonitrile Plant								
Tecnimont—Milano	1	1991	Centrifugal	67,000 Nm3/h	0.98	3.34	4,700	4,360
Saratov (URSS)			Air					ST, EMG
Geothermal Power Station								
ENEL—Roma	5	1989	Gas exhauster	20,000 kg/h	0.07	15.71	5,300	2,430
Pisa (Italy)			End. gas (2)					ST
Bacon Manito	2	1991	Gas exhauster	10,500 kg/h	0.14	7.47	6,800	870
Philippines			End. gas (2)					ST
ENEL—Roma	3	1993	Gas exhauster	4,000 kg/h	0.07	15.71	9,070/	565
Pisa (Italy)			End. gas (2)				16,440/	ST
							16,440	
Nitric Acid Plant								
UHDE-Dortmund for Quimigal	2	1980	Axial-centrifugal	63,400 Nm3/h	0.99	6.06	6,950	5,660
Alverca & Lavradio (Portugal)			Air					ST, GE
Kemira Helsinki	1	1989	Gas expander	47,755 Nm3/h	3.20		7,500	1,950
Refinery Service								
Mannesmann-Germany for	1	1989	Centrifugal	16,952 kg/h	6.48	1.59	11,252	1,100
Szazhalombatta (Hungary)			Hydrogen					EMG
Sugar Mill								
I.S.I.—Padova	2	1986	Centrifugal	106,630 kg/h	2.32	1.20	5,840	1,370
Finale E.-Pontelongo (Italy)			Steam					ST
I.S.I.—Padova	2	1987	Centrifugal	116,000 kg/h	2.87	1.20	5,400	1,700
Argelato (Italy)			Steam					ST
Sadam S.p.A.—Fermo	1	1988	Centrifugal	116,000 kg/h	2.35	1.41	6,800	2,660
Jesi (Italy)			Steam					ST
Terephtalic Acid Process								
Snam P. for Enichem Fibre	1	1986	Gas expander	30,500 Nm3/h	21.50		7,500	2,450
Ottana (Italy)								

(1) EM = electric motor. (2) End. gas = mixture of CO_2, H_2S, steam.
EMG = electric motor with gear system; GE = gas expander; ST = steam turbine.
SOURCE: Ansaldo.

free service are the ultimate goals of both the end user and expansion joint supplier.

Unfortunately the most competitively priced product does not guarantee these goals. Expansion joints are an integral part of the exhaust system whose reliability is every bit as important as the turbine itself. If the expansion joint fails then the whole system must be shut down.

Basic Definitions and Configurations

Expansion joint suppliers have individual designs that they have developed and tested over many years. These designs may vary from one supplier to another but generally there are three basic configurations of expansion joints as seen in Figs. E-45A, E-45B, and E-45C:

- Hot to hot
- Hot to cold
- Cold to cold

Although designs may vary slightly there are basic principles that must be followed to ensure a trouble-free life for the expansion joint. For example, to avoid differential thermal expansion at the connecting flanges, both flanges must be made

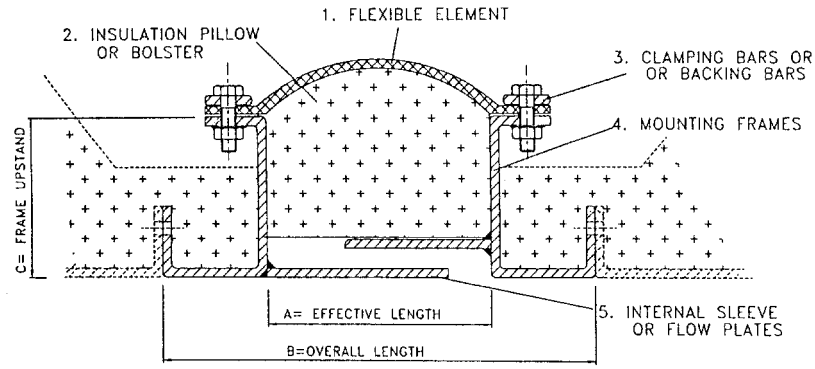


FIG. E-45A Expansion joint: hot to hot. (Source: Townson Expansion Joints.)

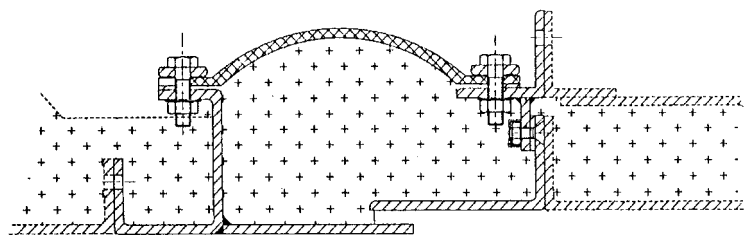


FIG. E-45B Expansion joint: hot to cold. (Source: Townson Expansion Joints.)

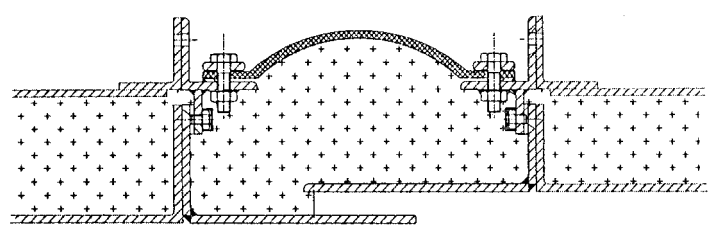


FIG. E-45C Expansion joint: cold to cold. (Source: Townson Expansion Joints.)

of the same material and have internal and external insulation designed to allow both flanges to operate at the same temperature.

Design Principles

Composition

Fabric expansion joints are designed to accommodate the thermal movements and vibration of turbines and adjacent ductwork without imposing any loads on the

turbine outlet flange. This is achieved by utilizing a flexible element capable of withstanding the high temperature and pressure fluctuations yet able to be compressed or stretched without allowing the exhaust gas to escape.

The composition of a flexible element varies according to the individual supplier's recommendation but each will contain an impervious gas barrier at some point within the structure. This gas barrier is usually made from a layer of PTFE or similiar that would have an operational temperature limitation of around 250°C. The inner layers of the joint are designed to reduce the exhaust gas temperature, which can be as high as 650°C to within the temperature limitations of the gas barrier.

Generally expansion joint flexible elements have a buildup of:

- *Outer cover*—Weather and mechanical damage barrier
- *Gas barrier*—Impervious layer of usually PTFE
- *Thermal barrier*—Layers of fabric or insulation. The thickness and composition of these will depend on the individual supplier and thermal drop required.

The flexible element of GT expansion joints should be designed to withstand the full gas temperature without the aide of an insulation pillow. However it is common practice to install a full-cavity insulation pillow for the following reasons:

- Prevents fluttering of flexible element
- Reduces noise emission through joint
- Provides additional thermal insulation to expansion joint.

Advantages of fabric expansion joints over metallic joints

- Accommodate more thermal movement within a shorter length.
- Easy to replace for future maintenance.
- Absorb radial growth without damage.
- Accommodate lateral movements within a single joint.
- Reduce imposed loads on turbine nozzle.

Problems and Solutions

Support frame damage

Cracking. Gas turbine exhaust systems are subject to severe temperature fluctuations ranging from ambient to full exhaust temperature within seconds. This feature creates problems for the steel work of the exhaust system and the mounting frame of the expansion joint. See Fig. E-45A. The inside surface of the frame sees the full gas temperature almost instantly yet the outer flange that the joint is mounted on is at this instant cold. This particular frame configuration has been used successfully for many years on conventional coal-fired stations where there is a startup period that allows all steel parts to reach optimum gas temperature over a period of hours rather than seconds.

The upstand of the frame in Fig. E-45A is designed to position the flexible element away from the full gas temperature so that it can survive without burning out. Obviously the higher the upstand the cooler the temperature and the safer the joint becomes, but in doing this you create greater temperature differentials between the hot inner surface and mounting surface of the expansion joint. This is a delicate balance that is one of the main secrets to a successful expansion joint, keep the

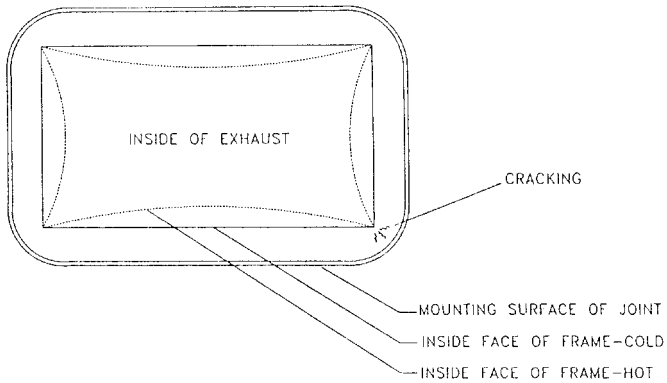


FIG. E-46 Cracking in an expansion joint. (Source: Townson Expansion Joints.)

joint cold and risk cracking to the frame or keep the frame hot and risk burning the joint out. The cracking of frames is more of a problem in rectangular expansion joints than circular ones and becomes more of a problem as the size increases.

As a result of the temperature differentials the frame distorts by bowing inward. (See Fig. E-46.) The result is flexing of the frames at the corners, which over a short period of time causes cracking. Upstands vary depending on the designer but are frequently seen to be between 100–150 mm, which is sufficient to cause cracking if precautions are not taken. With finite-element calculations and on-site temperature experience we have developed features within the corner design that eliminates the cracking.

For example, all welds that are a potential source of cracking have been removed, internal surfaces radiused, and corners stiffened to produce a design that can withstand several thousand cycles.

Distortion. The deflecting of the straight lengths of the expansion joint frame coupled with temperature differentials and induced stresses as a result of welding can cause the internal sleeves to distort, which as a result may cause them to clash or even fall off.

The internal sleeves are a vital component for the successful operation of the joint, the loss of which could result in premature failure of the joint. The simplest internal sleeve design is to weld a flat plate to the side wall of the expansion joint frame. See Fig. E-45A. This is a design that has been operating successfully for many years on conventional coal-fired power plants and smaller GT joints. However as turbines increased in size it became apparent that this basic design needed to be improved. To allow the sleeve to flex with the frame it should be supplied in short lengths of approximately 1 to 1.5 m with a gap between each piece. This gap must have a cover strip over it to prevent loss of pillow. To eliminate distortion as a result of welding and to stiffen the sleeve, it is suggested that the sleeve be flanged and bolted to the side wall. See Fig. E-47.

Tests and field experience have shown that internal sleeves should not be manufactured too long as this would allow them to vibrate which could cause them to fall off. Sleeves over 250 mm must be given special consideration.

Heat damage

Lateral movements. As turbines, boilers, and associated ductwork get ever bigger, the thermal movements increase proportionally. However, due to financial

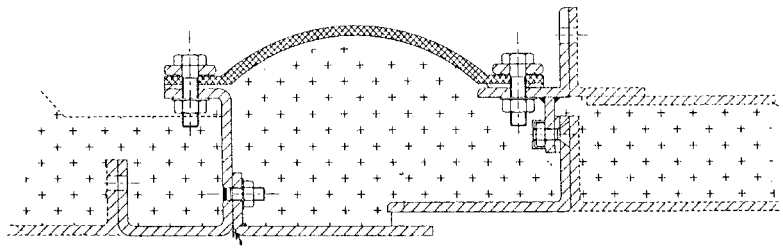


FIG. E-47 Support frame distortion. (Source: Townson Expansion Joints.)

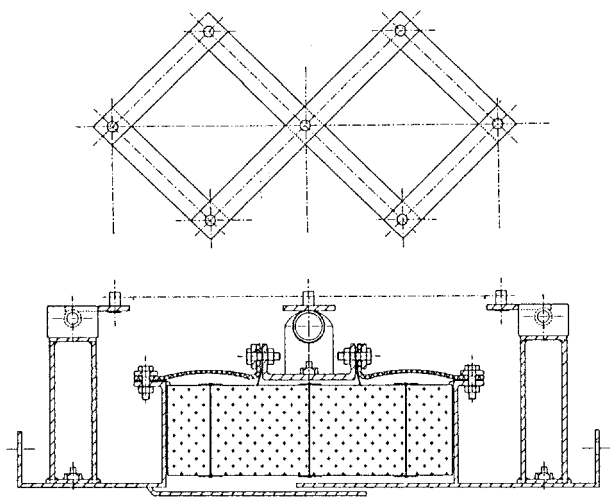


FIG. E-48 Pantograph design. (Source: Townson Expansion Joints.)

restraints the number of expansion joints in each system has not increased; it is merely expected that the same number of joints will accommodate the increased movements. There comes a time, though, when a simple expansion joint cannot absorb any more movement individually. This is especially true when one considers lateral movements.

Typically at the HRSG inlet the expansion joint sees the vertical growth of the boiler, which can be as much as 150 to 200 mm. Inexperienced suppliers would try to use one single expansion joint for this application but in fact no individual flexible element should be asked at these elevated temperatures to absorb more than 60–75-mm lateral movement. The materials of the expansion joint start to crease or fold in the vertical side walls as the joint absorbs the lateral movement. As the folds become more and more pronounced the external materials of the joint are prevented from being cooled by the ambient air, heat builds up, and the joint fails prematurely.

The solution to the problem is to simply limit the lateral movement on the individual joint by:

- Presetting the joint by half the lateral movement.
- Installing a double joint with a pantograph control mechanism to equally divide the movement between both joints. See Fig. E-48.

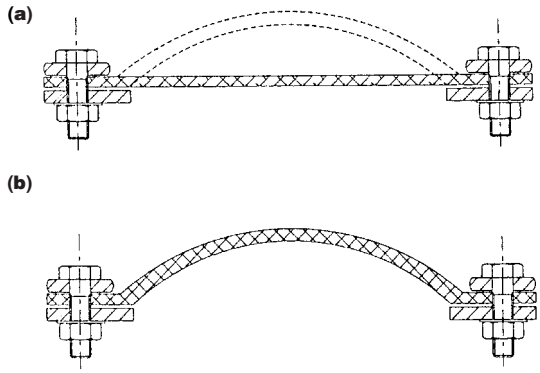


FIG. E-49 Flat belt (*a*) and preformed arch (*b*) construction. (Source: Townson Expansion Joints.)

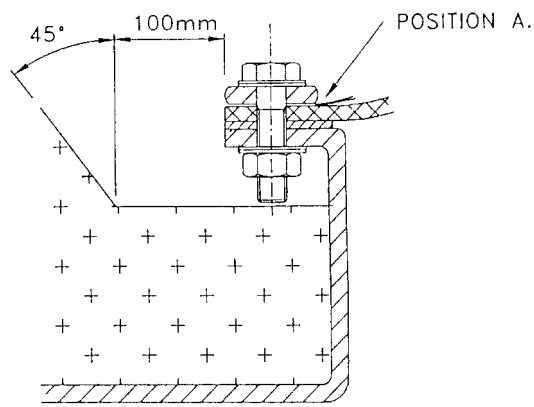


FIG. E-50 Heat damage at mounting flange. (Source: Townson Expansion Joints.)

Axial movement. Axial thermal movement can create different problems for the expansion joint than those seen with lateral movements. Conventional flat belt expansion joints when installed have small creases or folds in the corner areas. Under normal operational and small axial movements these creases do not cause a problem but as movements increase the creases become so pronounced that the outside materials are not allowed to cool and therefore burn out. To prevent this from happening the joint should be constructed with a preformed arch in the corners. See Fig. E-49. This configuration will increase the movement capacity of the fabric joint to approximately 150 mm; anything above this would require a double joint using a pantograph mechanism. Excessive axial movements can also result in the material of the joint in the straight lengths folding either back on itself or over the backing bar. Either situation would cause premature failure of the joint.

Mounting flanges. The flange frame on which the joint is mounted is designed to operate at very high temperatures as described previously to reduce stress levels. Unfortunately these temperatures frequently cause failure of the expansion joint if they have not been considered by the expansion joint designer. See Fig. E-50.

The temperature of the frame is transferred through the mounting bolt to the backing bar that burns the joint from the outside or allows heat to build up at point A resulting in premature failure of the joint.

Generally at gas temperatures of 400 to 500°C conventional expansion joints operate successfully without the need for any special precautions. For temperatures above these the overall design configuration should be reviewed by finite-element analysis to establish the temperature of the backing bars.

Should the backing bar temperature be a problem it can be reduced by installing a gasket under the joint and using thermal barrier washers.

Proximity of buildings. Fabric expansion joints are designed to withstand the full gas temperature but for them to achieve this there must be a relatively cool ambient temperature of less than 100°C. The design of the fabric element is such that the gas temperature is lowered through the different layers of the joint to an acceptable level at the point where the gas barrier is located within the joint. Frequently joints are located within the acoustic enclosure where the ambient temperature can be higher than expected, which can result in an abnormal temperature rise through the joint and premature failure.

High ambient temperatures are a real problem for expansion joint designers. Materials to overcome the problem by providing a gas barrier that will operate at elevated temperatures are limited to the likes of stainless steel foils. Unfortunately foils do not absorb movements easily and are prone to cracking from vibration. The recommended way to solve this problem, providing the ambient temperature cannot be reduced, is to blow cooling air over the joint surface. The cooler the outside surface of the joint the greater the life expectancy. Problems of this nature also occur when joints located close to buildings or the floor do not allow adequate room around the joint for air circulation. The closer a structure gets to the outer surface of the joint the greater the insulating effect it has on the joint surface.

Water washing

Expansion joints are designed to withstand hot gases and thermal movements; they are generally not designed to withstand the moisture that would build up in the joint from water washing the ductwork or turbine. The effect of this is twofold. First, the moisture attacks the fiberglass or high temperature fabric layers to a point that when they are dry they will have lost a considerable amount of their tensile strength and the integrity of the joint will be reduced. Second, the water wash often contains solvents to enhance the washing effect. These solvents settle in the base of the joint and once the water has vaporized a film remains within the fabrics or on the PTFE gas barrier. These solvents have in the past caught fire at startup, which completely destroys the joint. Various materials are available to line the inside face of the joint to prevent moisture attack, but unfortunately water will eventually get through. The ideal solution to this problem is to prevent the water access to the joint rather than trying to accommodate the moisture within the joint. See Fig. E-51.

Radial growth of steel parts

The expansion joint designer has many factors to consider when selecting materials, configurations, and methods of manufacture. One condition that is often overlooked is the fact that the steel frame that the joint is mounted on expands radially at the same rate as it does axially. For example a stainless steel frame with internal duct dimensions of 5 m × 5 m operating at 550°C would expand radially 50 mm, which in effect is stretching the materials of the joint by this amount. To

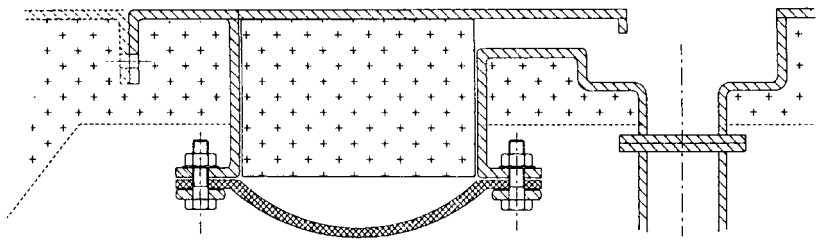


FIG. E-51 Water washing. (Source: Townson Expansion Joints.)

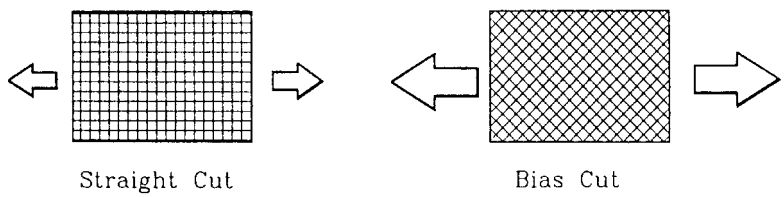


FIG. E-52 Straight and bias cuts. (Source: Townson Expansion Joints.)

overcome this problem reputable suppliers manufacture the joints using materials cut on a bias (see Fig. E-52), which allows the individual layers to stretch by as much as 15 percent more than materials cut the conventional way.

Insulation pillows

Pillow loss. Turbine exhaust gases operate at such a high velocity that they can create a negative pressure or sucking effect within the expansion joint. The joint and steel work are designed to withstand this but the insulation pillows need special attention. The velocity of the gas attacks the very fibers of the pillow and sucks them downstream in what is known as a picking effect. This is a slow process that gradually works at the pillow until the whole pillow has disappeared. There have been instances when the pillow as a complete unit has been sucked downstream through the gap in the sleeves. To prevent this the pillows must be mechanically fixed to either the side wall of the frames or the surface of the sleeves. Care must be taken to allow adequate material between the fixings to accommodate the thermal movement. Second, the pillow should be encased in a fine wire mesh to prevent the picking effect. High nickel alloy meshes are available that will prevent small particles passing through. Though not always possible due to movement limitations, utilizing a floating sleeve design (see Fig. E-53) helps to overcome the picking effect.

Pillow settlement. Insulation pillows are produced from insulation blankets, which in turn are manufactured from small strands of either ceramic fiber or fiberglass.

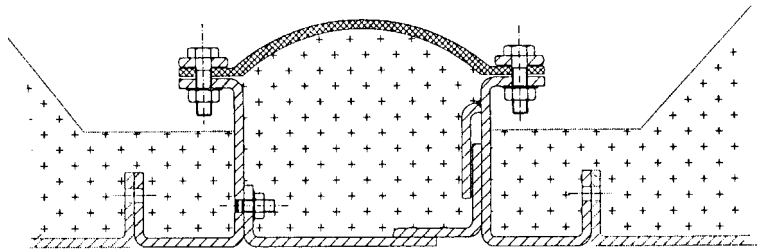


FIG. E-53 Floating sleeve. (Source: Townson Expansion Joints.)

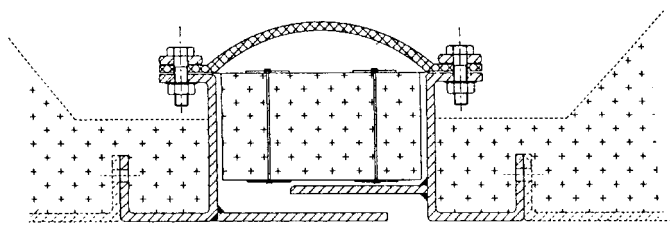


FIG. E-54 External insulation. (Source: Townson Expansion Joints.)

Though the density of the material is of the order of more than 150 kg/m, the material itself is still quite loose. Expansion joints vertically mounted are always at risk of having the insulation pillows in the vertical legs settle down to the bottom of the leg exposing the top corners of the joint.

Pillows must be manufactured incorporating cushion pins equally spaced across the pillow to keep the layers of insulation bound together. In addition pillows should be either pinned to the side walls or sleeves as described above or flanged so they can be mounted between the expansion joint and mounting steelworks. See Fig. E-54.

External insulation

Expansion joint manufacturers worldwide insist that the outside of gas turbine expansion joints must not be insulated over. As described previously, covering the joint would significantly raise the ambient air temperature and cause premature failure of the joint. The external insulation should be terminated at the joint as shown in Fig. E-54. Failure to do this could jeopardize the finite-element calculation that guarantees the required life cycle of the frames or could cause premature failure of the joint as a result of heat buildup from exposure of the steel casing.

Noise

Noise requirements must be specified at the inquiry stage as they can significantly affect the design of the fabric element and steelwork. To achieve optimum noise reduction the joint and pillow must be manufactured from the highest density material available that will operate at the stated design conditions. The internal

flow plates should be designed slightly different to reduce the gaps allowed for thermal movement. Floating sleeves are often recommended in this situation. See Fig. E-53. Noise calculations should be made available by the expansion joint supplier.

Nonmetallic expansion joints, if designed and installed correctly, will provide years of trouble-free service. One of the most common reasons for failure in expansion joints are problems developing within the expansion joint as a result of incorrect installation or damage caused during installation. Emphasis must be placed on strictly following the guidelines as laid down in the installation manual supplied with the joints. These instructions have been developed over many years of experience of fitting joints and solving problems. To ignore them or deviate is risking failure of the joint.

In conclusion, nonmetallic expansion joints are the most economical method of accommodating thermal expansion in gas turbine and HRSG applications. However, to achieve this the expansion joint supplier must have a proven track record as most of the above have been developed over many years using finite-element calculations, on-site experience, and exposure to problems and providing expansion joints that have stood the test of time.

Explosion; Explosion Hazard Analysis; Explosion Hazards

Explosion hazards are a major consideration in most operating plants. Hazards include leaks of flammable liquid and/or a gas. One potential culprit is gas-turbine fuel. Others may be hydrocarbon product leaks. Because of space considerations, analyzing gas-turbine fuel leaks as a hazard will be discussed as a typical case. Although the extracts that follow are from powerplant operators, their methodology with their gas turbine or combined cycle applications are not restricted to power production applications.

Case Study 1: Assessing Explosion Risk Potential at Teesside Power Station, UK*

The potential for an explosion inside a gas turbine combined heat and power (CHP) plant or a combined cycle gas turbine (CCGT) plant enclosure from a fuel leak has been widely addressed in the industry and reviewed at a number of seminars. Enron, the Teesside Power Station operators, contracted a study into the explosion potential inside the gas turbine (GT) enclosure and to recommend solutions to achieve an acceptable level of risk.

A structured approach was applied to identify the hazard, evaluate the risk, and identify effective, practical, risk reduction measures.

The study consisted of:

- A comprehensive air movement study inside the enclosure to define the total air balance and to create a 3D air velocity grid.
- A process hazard review (PHR) to identify credible leak events for the CCGT fuels.
- Assessment of the potential for formation of a flammable mixture for each case.
- Evaluation of potential ignition sources to review causes and to determine suitable probability values to be used in the risk assessment.

---

*Source: Adapted from extracts from Hunt and Beanland, "A Risk Based Approach to the Potential for CCGT Enclosure Explosions: A Study on Teesside Power Station," Eutech Engineering Solutions Ltd., UK, ASME paper 98-GT-446.

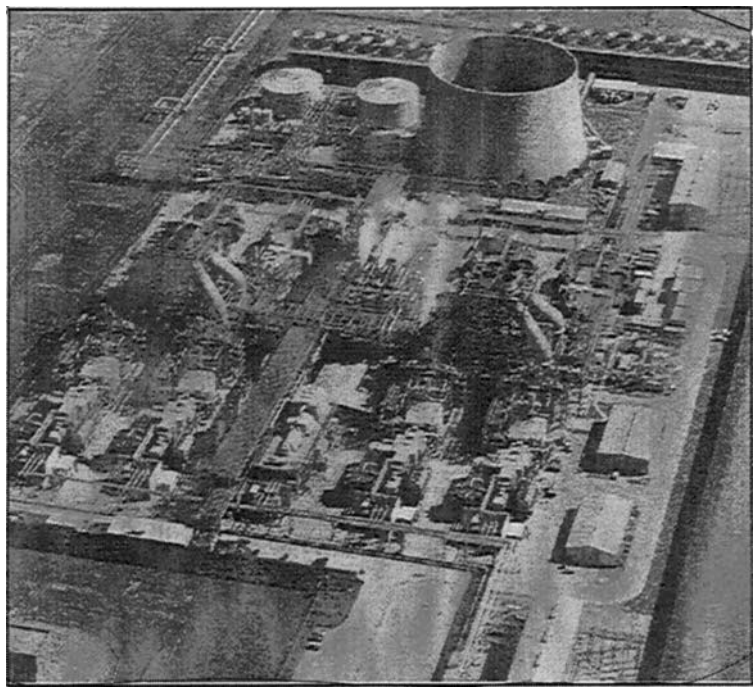


FIG. E-55 Teesside Power Station. (Source: Hunt and Beanland.)

- A hazard analysis (HAZAN) to quantify the risk from an explosion to the operator at most risk.

From the resulting fault trees the most effective options to achieve a measured reduction in risk were identified. These included improved gas detection, air circulation measures to reduce the incidence of leaks, revised CCGT enclosure access, and operating procedures.

The structured assessment of the risk and the selection of effective improvements as applied to the study on Teesside Power Station is covered below.

The plant

Teesside Power Station, with a 1875 MW output, is the world's largest gas-fired combined cycle heat and powerplant (see Fig. E-55). It has a total of eight Westinghouse MW 701 DA gas turbines with associated heat recovery generation and two Westinghouse 800 te/hr steam turbine generators. The combination provides for low environmental emissions and high thermal efficiency. All eight CCGT units are identical having acoustic enclosures for noise suppression with ventilation for gas turbine cooling. See Table E-12.

Figure E-56 shows the station's basic process flow diagram. From this diagram it can be seen that there are a number of fuel sources available to ensure continuity of process steam supply to the adjacent chemical companies. The primary fuel source is natural gas supplied from either the Central Area Transmission System (CATS) or Transco's National Transmission System. Backup fuel is available in the form of vaporized propane, which may be used to increase the natural gas Btu value as a startup fuel prior to introduction of liquid fuel. Naphtha is also available for online transfer to provide security of steam supply in the event of loss of primary fuel.

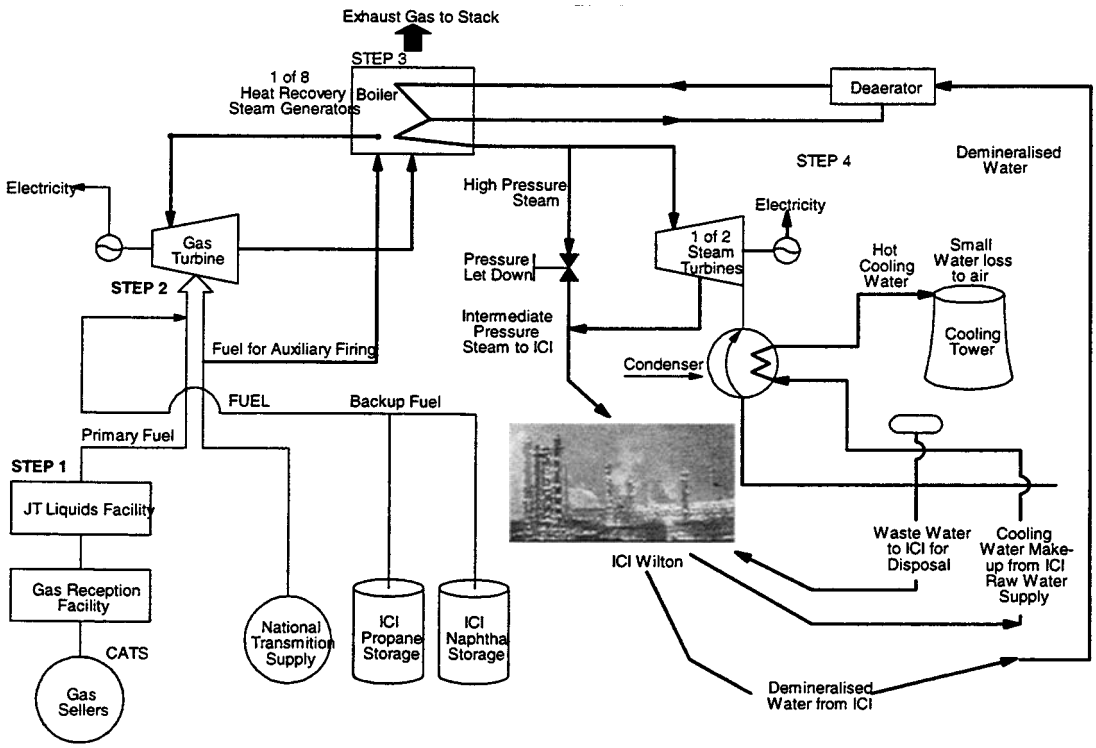


FIG. E-56 Teesside Power Station process flow diagram. (Source: Hunt and Beanland.)

TABLE E-12 Teesside Power Station CCGT Leak Scenarios

Leak Event Cases	Orifice Size (mm)	Mass Rates (kg/sec)	Mixture Formed
Small (<1.5 mm dia)			
1 Leak through valve stem	1.1	0.005	Nonflammable
Medium (1.5 to 6 mm dia)			
2 Joint with no gasket	2.8	0.013	Flammable
Large (>6 mm dia)			
3 0.25″ pipe failure	9.2	0.14	Flammable
4 Untested flange	10.2	0.18	Flammable
5 0.50″ pipe failure	15.8	0.42	Flammable
6 Misalligned joint	17.1	0.52	Flammable

The potential for an explosion from a fuel leak inside gas turbine enclosures for CHP and CCGT plants had been identified, and during early 1996 Enron, the operator, contracted Eutech to undertake a study into the risk to operators inside the enclosure from the potential for an explosion from the gas fuel supply.

The study used a structured approach based on experience with carrying out similar studies within the chemical/petrochemical industry to identify the hazards, evaluate the risk, and to identify practical risk reduction measures.

Unfortunately at 00:25 hr on July 17, 1996, a fire and explosion occurred in CCGT 106 enclosure. The incident was due to ignition of a leak of naphtha from a joint during fuel changeover. The explosion injured an operator who had entered the GT enclosure to confirm satisfactory operation of equipment and it caused serious

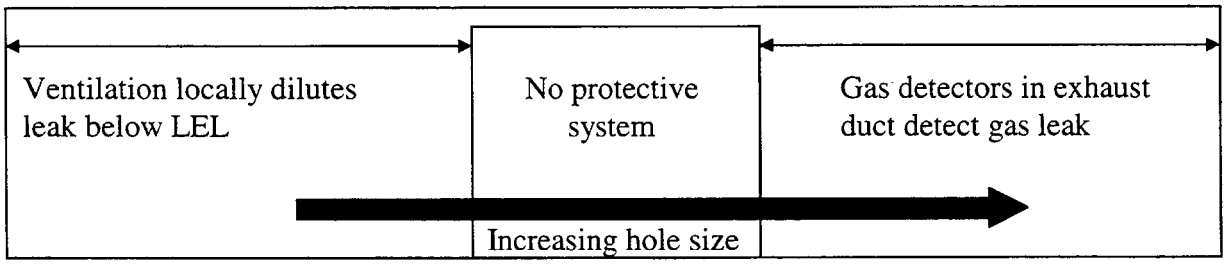


FIG. E-57 Basis of safety. (Source: Hunt and Beanland.)

damage to the enclosure, including dislodging equipment, walkways, pipework, roof panels, and doors. The fire was extinguished by the automatic activation of the gaseous fire protection and fuel emergency shutdown systems. This section covers a study into the risk to operators inside the enclosure from the potential for a gas explosion; however, it does not cover the incident investigation and redesign of the naphtha fuel system as carried out by Eutech.

The explosion potential study was completed and the recommendations from the study were implemented.

CHP plants and CCGT plants have become more numerous in recent years for local power and heat generation. Many are based on gas turbines located within acoustic chambers for noise reduction and fire-suppressant-system retention. The primary fuel is typically natural gas with a liquid fuel backup. They have very complex fuel supply pipework to the turbines with multiple flange joints, which operate at high pressure and can give rise to explosion hazards within the enclosure in the event of fuel leaks.

For a 40-MW machine the pipework may include 30 flanges or flexibles, whereas a 250-MW machine may have over 200 flanges, 90 flexibles, etc. The explosion hazards are relevant to smaller gas turbine–based CHP schemes (3–50 MW) up to large CCGT installations.

The HSE have become increasingly concerned about this issue, having identified situations of poor ventilation, ineffective gas detection and uncontrolled access procedures in enclosures, which would result in the potential for an explosion and injury.

Safety in most enclosures relies on a combination of dilution of leaks by ventilation and gas detection, with the gas detectors located in the exhaust ventilation stream.

However most ventilation systems were designed for cooling rather than diluting a flammable mixture; modifications can create a dilemma. Increasing the ventilation flow rate alone makes the gas detection system less likely to detect a leak according to the particular installation. Depending upon flow rates and gas detector alarm settings, there may be a range of leak sizes that are too large to be diluted below the lower explosion limit (LEL) in the vicinity of the leak, but are too small to register an alarm on the gas detector when mixed with the total airflow through the enclosure. This can lead to a range of leak events that have no "basis" of safety—this is represented in Fig. E-57. This concept is important when considering appropriate solutions to reduce the risk of enclosure explosions.

Risk-based approach

The application of a risk-based approach to CCGT enclosure explosion potential is developed from the need to identify the hazards, to evaluate the risks of current

operation and then to identify the most effective improvements to achieve a basis of safety for operation.

The approach falls into three main steps for explosion potential in gas turbine plant enclosures:

- Step 1: Hazard identification
- Step 2: Risk evaluation
- Step 3: Risk reduction

This approach leads to the identification of hardware or software improvements that address the specific requirements of the installation under consideration. Software improvements cover such issues as safety management systems, incident response procedures, training, quality, maintenance practices, alarm and trip testing, etc. Hardware improvements cover such issues as ventilation, gas detection, interlocks, control system, fuel system, protection systems, etc.

The methodology is outlined in this section and then its application to Teesside Power Station is described as follows.

Step 1: hazard identification. This step involves considerable data collection and input by plant personnel. This ensures that an accurate picture of current operations, incidents, and hence potential risks is achieved. See Fig. E-58.

Step 2: risk evaluation. A hazard analysis is undertaken to quantify the risk to operators both internal and external to the enclosure.

It is based on combining the quantified frequency of an explosion with the probability that an operator is either inside the enclosure, or is external to the enclosure but within range of the debris.

This establishes a base level of risk associated with current operating practices and hardware. See Fig. E-59.

Step 3: risk reduction. During this step the fault trees developed in step 2 are used as guidance to assist identification of software and hardware improvements to reduce the risk to an acceptable level.

The most effective improvement options to give the maximum benefit are identified by consideration of the fault trees. Finally, the fault trees are updated in order to assess achievement of an acceptable level of risk. See Fig. E-60.

Hazard identification

Air movement study. The purpose of the air movement study was to assess the effectiveness of the ventilation around potential leak points. The air movement study consisted of a series of measurements, as described below, leading to the preparation of a 3D flow map. This gave a comprehensive set of accurate measured ventilation data from inside the enclosure.

At Teesside Power Station, each GT enclosure was approximately 635 m^3 with 14 air inlet points (Fig. E-61). Ventilation was provided by three roof fans with a design airflow rate of 4.8 m^3/s each.

For all tests a full method statement was issued to ensure the gas turbine enclosure was in a normal operating mode. Checks included—fan direction, doors closed, air dampers at normal point, any inlet meshes on vents clean, and the external wind speed was noted.

Full airflow balance. Airflow readings using a vane anometer were carried out on all 14 air inlet points with the number of readings defined by the air inlet area. For each inlet a mean velocity and hence total flow was established.

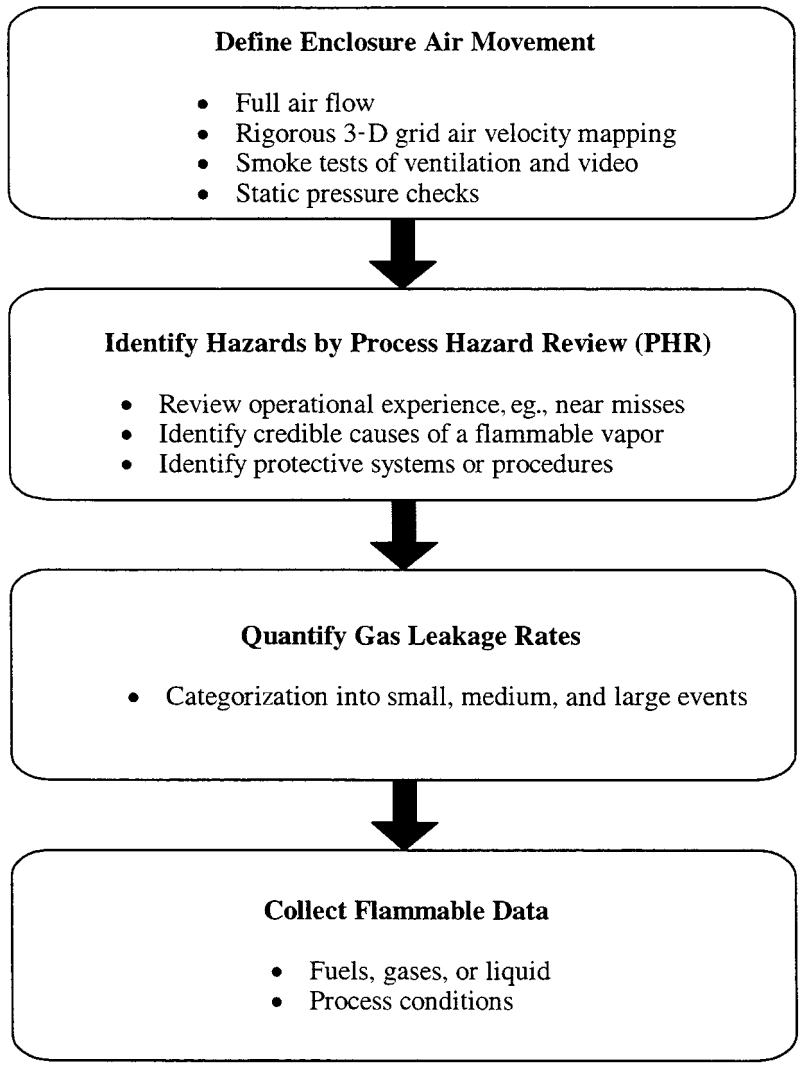


FIG. E-58 Hazard identification. (Source: Hunt and Beanland.)

The total measured airflow into the enclosure at 20°C was 13 m^3/s, which includes a leakage allowance. This equated to a total flow of 15 m^3/s at 70°C in the fan exhaust.

Static pressure checks. To cross-check the total airflow the internal static pressure was measured at 50 Pa, with the calculated loss through fan casing of 44 Pa. This gave a total static pressure developed by the fans of 94 Pa. This was then checked against the fan performance curve, which correlates to approximately 5 m^3/s per fan, the same as the measured air rate per fan.

3D velocity grid. A 3D velocity grid was prepared by measuring air movement velocities at each point and using tracer smoke to give directional data. This was used later to evaluate the dilution ventilation rates at potential fuel leak points.

(This was considered the most accurate and practical approach for this installation as opposed to using available computational fluid dynamics (CFD) software.)

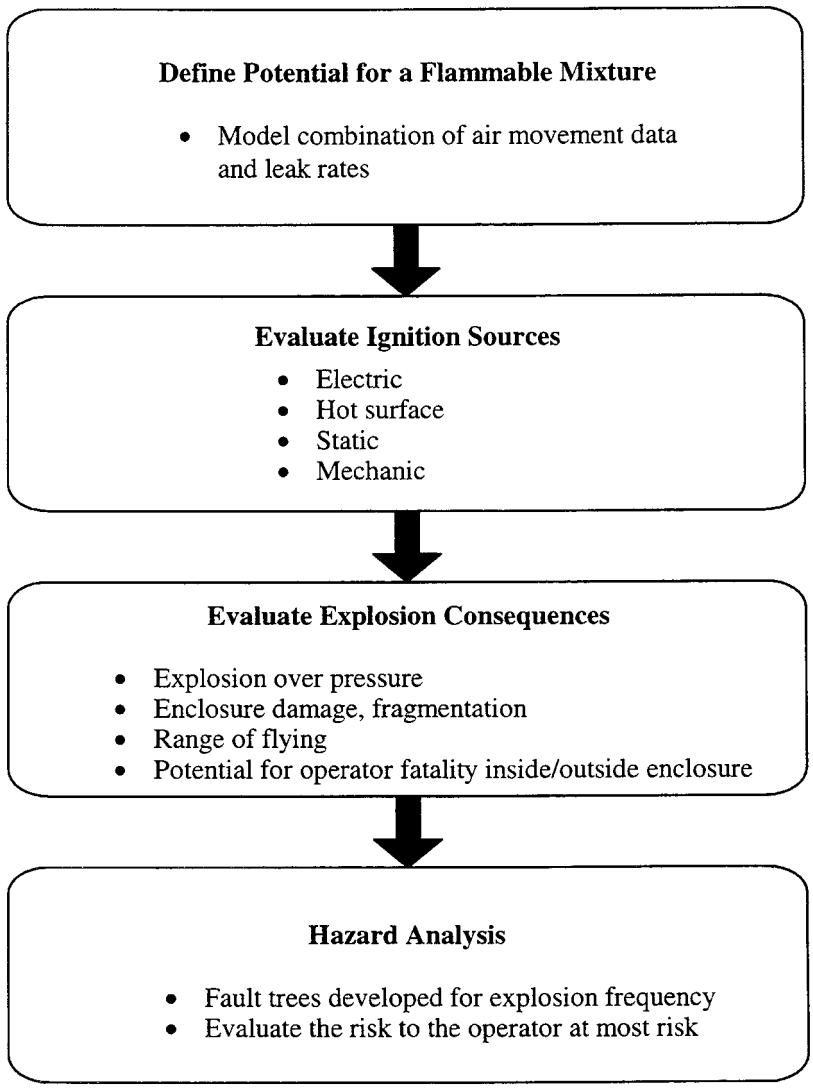


FIG. E-59 Risk evaluation. (Source: Hunt and Beanland.)

The fuel ring main was used as the reference point and a 3D grid established at 0.5 m intervals for planes A–F, 1–8, and a–m (on end elevation); see Fig. E-62.

From this a complete velocity and thus volumetric flow picture was produced inside the enclosure consisting of approximately 500 data points. The measured velocities ranged from 0.25 to 2.0 m/s throughout the enclosure.

Smoke tests. In order to attain a qualitative flow assessment inside the CCGT enclosure, tests were carried out using a smoke generator and recorded on video. The smoke used had a similar density to air and had the effect of making the airflow patterns visible inside the enclosure. The solid flooring resulted in a reduction in effective air movement and replacement with open flooring grid was recommended.

From the smoke tests and 3D grid it was determined that there were no dead spots (i.e., areas of zero flow) although there was considerable flow variation inside the enclosure.

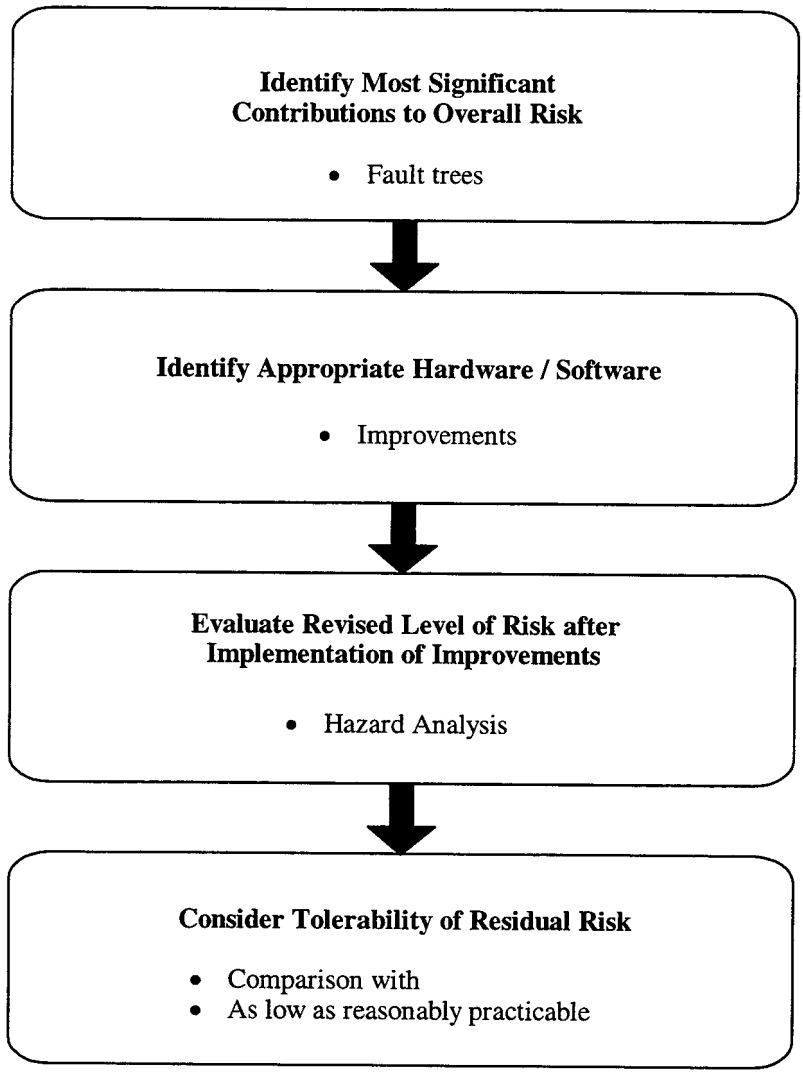


FIG. E-60 Risk reduction. (Source: Hunt and Beanland.)

CCGT operating conditions. The initial air movement measurements were undertaken on an operating gas turbine where the temperatures of the air at the exhaust fan were up to 70°C. In order to assess the effects of a cold CCGT, which would represent a startup situation, the smoke tests and velocity measurements were repeated with the CCGT offline. Due to the majority of the gas turbine being lagged no significant effects on flow pattern or velocity were noted for this installation. However from experience with other studies, the temperature difference of the circulating air between a gas turbine at startup (i.e., cold) and a running unit can lead to considerable variation in airflow patterns and velocities, which has a subsequent effect on the potential for the formation of a flammable mixture.

Process hazard review. The technique of process hazard review (PHR) was originally developed by ICI to assist in the identification of possible causes of accidents on existing chemical plants. The review has many similarities with

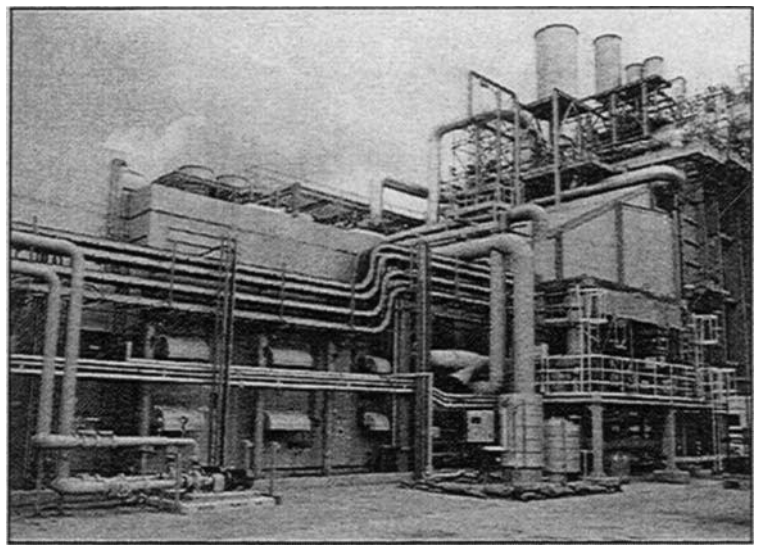


FIG. E-61 Teesside Power Station CCGT enclosure showing air inlets. (Source: Hunt and Beanland.)

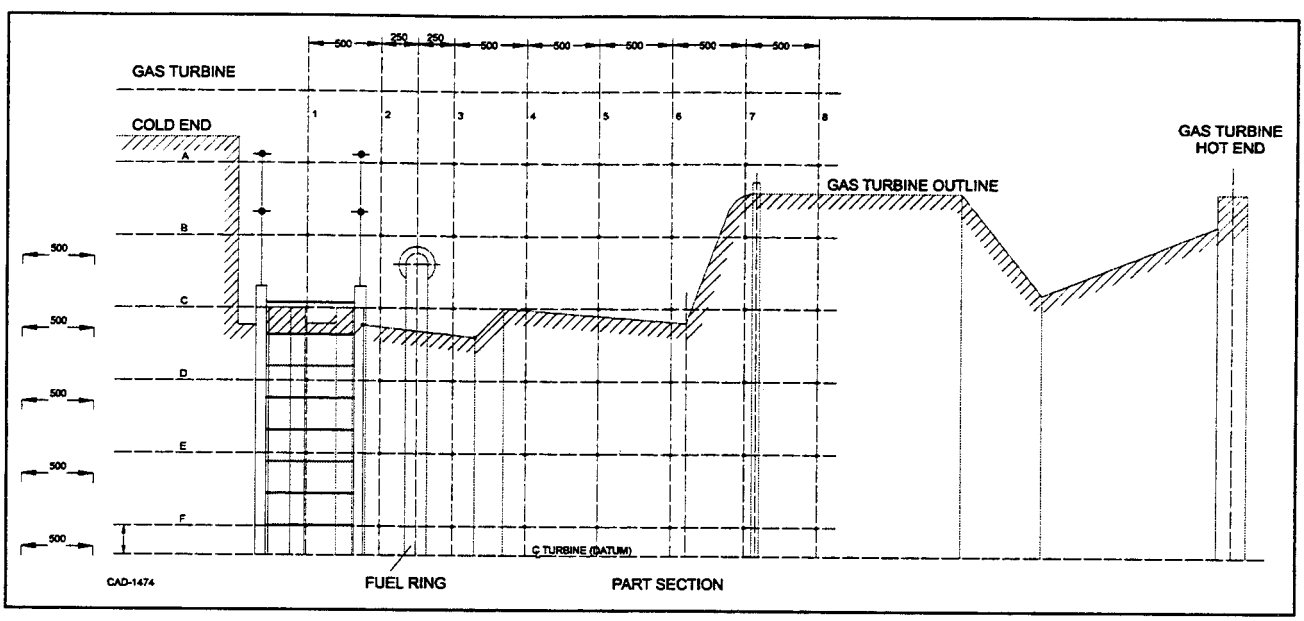


FIG. E-62 Teeside Power Station CCGT 3D velocity grid. (Source: Hunt and Beanland.)

HAZOPs or hazard studies practiced in the chemical industry since the late 1960s, but whereas HAZOP was primarily developed to assist hazard identification at the design stage of new plants, PHR exploits the experience already gained of operating and maintaining the actual plant that is under review.

PHR is normally used to identify a wide range of potential accident types from fire and explosion to toxic gas releases. However, the common theme is the identification of credible mechanisms that can lead to a loss of containment. The PHR team, prompted by a set of guidewords, considers whether such issues as over-

pressure, failure of vulnerable components, maloperation, maintenance activities, etc., can lead to a loss of containment. The team also identified the safety systems that either reduce the likelihood of the incident occurring or mitigate the consequences. The PHR for the gas turbine enclosure at Teesside Power Station was carried out by a team of engineering and operations personnel from the station and was chaired by a Eutech PHR leader.

The PHR at Teesside Power Station concentrated on the risk to operators inside the enclosure. Consequently the study team was interested in three events: a loss of containment of fuel, the presence of a source of ignition, and an operator present inside the enclosure. The team used the well-established PHR methodology, but with a modified set of guidewords to assist in the identification of causes of these three events.

Having qualitatively identified potential hazards and associated protective systems in this way, the team identified a series of minor but significant changes that reduce the likelihood (or frequency) of a hazardous event occurring, or that mitigate the consequences. These changes were related to either minor hardware modifications or to changes in operating or maintenance practices.

Also, the PHR study assisted the team to develop a clearer, but qualitative, understanding of the residual hazards that it had not been possible to eliminate. The hazard analysis used this understanding to assist business management in deciding if this was a tolerable level of risk.

Fuel leak quantification. Three fuels are used to supply the CCGTs at Teesside Power Station.

Natural gas is the primary fuel. It is supplied at 25 barg and then reduced to a normal operating pressure of 15 barg supply to the nozzles. The gas enters the CCGT enclosure via a 4-in nb pipe that feeds a circular manifold, which is fed to the gas turbine via $18 \times 1^1/_2$-in hard-piped lines.

Naphtha and propane are used as the backup or secondary fuel. Figure E-63 shows the fuel system.

The PHR, as detailed previously, identified credible fuel leak events. For each event, gas leakage rates were calculated.

- Small leak events correspond to a hole size <1.5 mm
- Medium leak events correspond to a hole size 1.5–6 mm
- Large leak events correspond to a hole size >6 mm

A summary of events and flammable assessment is given next.

Risk evaluation

Flammable mixture potential. In order to assess the ventilation inside the CCGT enclosure, the measured ventilation volume rates at each point on the grid were quantified from the 3D velocity grid data. The flammable mixture potential was then modeled by a combination of the air volumetric rates with the calculated fuel leak rates and compared to the flammability criteria of the lower explosive limit (LEL). This approach assumed instantaneous ideal mixing of any gas leak within the air to give a homogeneous mixture. (This was supported by additional jet mixing calculations.)

For the study on Teesside Power Station a summary of results is given in Table E-12 with a flammability diagram (Fig. E-64) showing two leak event cases.

The results show that for small leak scenarios, ventilation reduces the

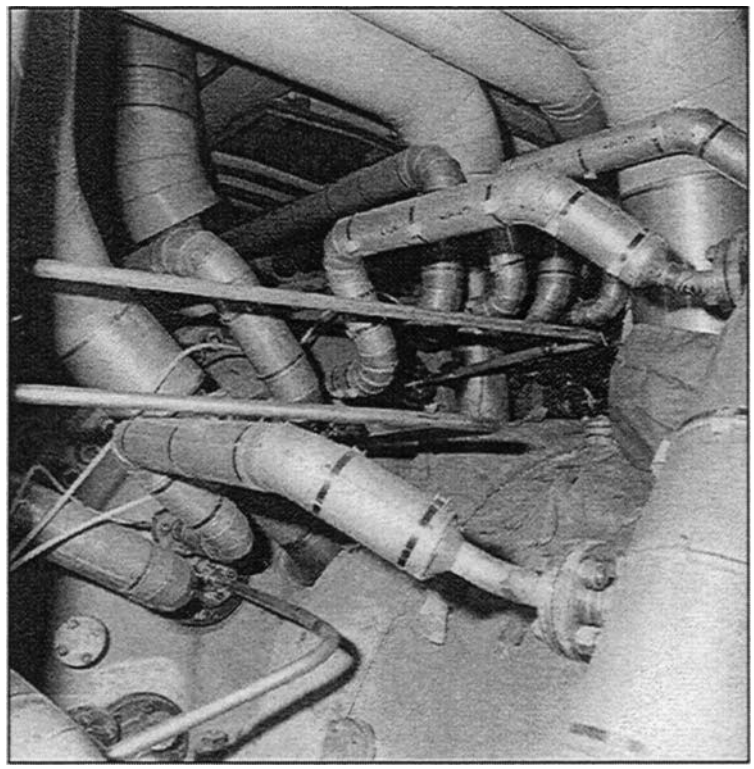


FIG. E-63 Teesside Power Station CCGT fuel system pipework. (Source: Hunt and Beanland.)

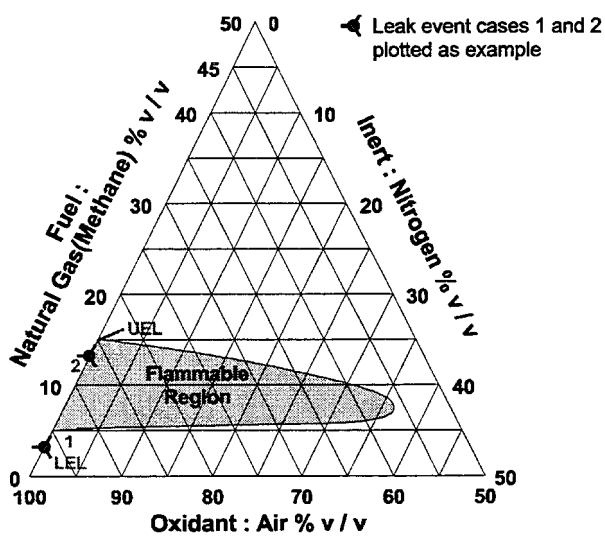


FIG. E-64 Methane flammability diagram. (Source: Hunt and Beanland.)

concentration below the LEL and therefore can be considered to provide a "basis of safety" for such events.

In medium leak scenarios 70 percent of mixtures formed exceeded the LEL, and in large leak scenarios all mixtures formed exceeded the LEL.

These results were used later in the hazard analysis as the basis to establish the probability of a flammable mixture being present caused by different hole sizes.

From these results it can be seen that for small leaks dilution ventilation provides a basis of safety by eliminating the formation of a flammable mixture. Hence the reliability of the ventilation system was assessed in the hazard analysis as being a critical safety system.

However, for medium and large leak scenarios the enclosure ventilation rates would need to be increased by up to 50 times in order to achieve a basis of safety of dilution ventilation. This was not considered a viable option and the basis of safety needed to be provided by other measures.

Sources of ignition. Conventional wisdom presents the "fire triangle" with three corners of fuel, oxidant, and source of ignition. Although sources of ignition should be eliminated wherever possible, avoidance of all ignition sources is not generally practical and would not be considered an acceptable approach as a basis of safety. Most of the protective systems inside gas turbine enclosures are intended to prevent the combination of fuel and air becoming flammable. However steps can also be taken to reduce the probability that there is a source of ignition present should a flammable fuel/air mixture accumulate.

At Teesside Power Station, the following potential sources of ignition were identified:

1. Faults in electrical equipment. Electrical equipment designed and maintained for hazardous areas is less likely to be a source of ignition.
2. Gas turbine hot spots which may cause ignition of a flammable mixture. Published data show specific autoignition temperatures for individual fuels; however, the potential for autoignition exists over a wide range of temperatures and is dependent on such factors as flow rates, surface roughness, geometry, and fuel concentration. Within the enclosure at Teesside Power Station several exposed surfaces at temperatures up to 400°C were measured as part of the study. This is below the published autoignition temperature of natural gas, but for the above reasons it is not possible to discount autoignition as a potential source of ignition.
3. Discharges of fuel from leaks may generate static that can ignite the ensuing flammable vapor.
4. Mechanical failure of rotating equipment that may generate sparks or local hot spots. It is usually not possible to eliminate these potential sources of ignition but attention to design details reduced the risk of them being a source of ignition.

The hazard analysis, discussed below, requires quantification of the probability of a source of ignition being present. From other investigation reports of explosions there are two situations worth mention:

- Flammable mixtures in environments where there is almost always a source of ignition (e.g., busy roads, steel plants). The probability of a source of ignition can be considered to be 1 with some confidence (i.e., every time there is a flammable mixture it will be ignited).
- Flammable mixtures in environments that are specifically designed and managed to be free of sources of ignition (e.g., refineries, flammable chemical plants). For a small leak in such an environment, we might consider the probability of a source of ignition to be 0.1 (i.e., only 1 in 10 flammable mixtures will be ignited).

The latter might seem pessimistic, given that the environment is designed to be ignition free, but ignition can be caused due to a variety of sources, many of which are neither clearly understood nor quantifiable.

Hazard analysis. The concept of hazard analysis (or HAZAN) is widely used in the UK chemical industry and is a recognized part of the approach to the Control of Industrial Major Accident Hazard Sites or CIMAH legislation.

The chemical industry has long recognized that the operation of plants with large inventories of toxic or flammable materials will always constitute a hazard. These hazards cannot be completely eliminated other than by abandoning the industry and losing the associated social and financial benefits. A balanced decision is required based on the acceptability or tolerability of these residual risks compared with the economic and material benefits of being an industrialized nation. Quantification of this residual risk assists managers in making this decision.

The second benefit of hazard analysis is the guidance it provides in selecting appropriate improvement measures. If there are a variety of possible causes of a hazardous event, coupled with a number of protective measures, some of which are only applicable to specific causes, the HAZAN assists the Management in clarifying which factors make the most significant contribution to the overall risk level.

In carrying out a hazard analysis, the event being considered needs to be specified. The HAZAN provides a measure of risk of that specified event occurring. The units of this measure are per year (i.e., it is a frequency of occurrence). The event considered for Teesside Power Station was "ignition of a flammable vapor resulting in an explosion whilst an operator is present inside the enclosure."

A number of logic trees were developed to identify the possible causes of the top event based on current operation. Each logic tree was developed to a level of detail consistent with supporting operating data. Possible causes, or "demands," identified for Teesside Power Station included failure of a pipe flange joint, failure of an instrument connection, operator error during online maintenance, etc. A combination of the experience of Teesside Power Station personnel and typical failure rates for standard components or rates for human errors from Eutech Process SHE Guides were assigned to these possible causes.

Associated with each of these "demands" that may cause the hazardous event were various protective systems. These were either hardware (e.g., ventilation system, flammable gas detectors) or procedural (e.g., instructions on allowable maintenance activities). For any particular demand to lead to the hazardous event, all the protective systems designed to protect against that demand must have failed to perform. Again, this failure may be a hardware failure (e.g., the gas detector has drifted out of calibration) or a human error issue (e.g., the flammable gas detector alarm warned of a flammable leak but the operator failed to take appropriate action).

Also based on the air measurement study, an assessment was made of the probability of the gas detector being in the correct location to detect a leak. This took into account the gas detector alarm set point.

The reliability of these protective systems was quantified as the "probability that the protective system is in a failed state when a demand is placed upon it." This was a dimensionless number between 0 and 1 also known as the fractional dead time. The reliability of the protective systems was calculated using component failure data, Teesside Power Station proof testing procedures, and generic human error rates for plant operators.

The logic diagram combined the frequencies of the various "demands" that can lead to the top event with the probabilities that the protective systems designed

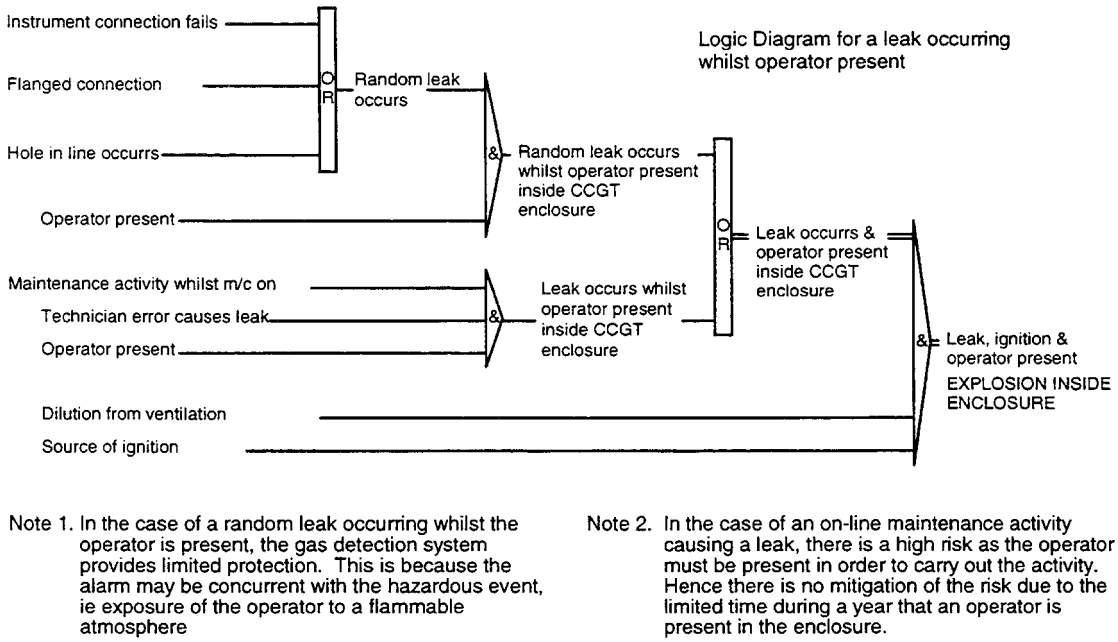


FIG. E-65 Typical fault tree logic diagram. (Source: Hunt and Beanland.)

to prevent those demands leading to the top event, be in a failed state. (See Fig. E-65.)

The output was a frequency at which the event under consideration (i.e., explosion and operator present) will occur. This was a measure of the risk of the incident that assisted Teesside Power Station management in deciding if it was a tolerable risk.

The risk from in-plant hazards may be considered in the context of the individual operator who is at most risk from the hazardous event being assessed. For Teesside Power Station, this was the operator who carries out the periodic inspections of the enclosure. The risk to him from the specified event was compared to the risk he is routinely exposed to in the course of working in the industry. (Local management may wish to set other targets that become the safety criteria for that plant.)

The criteria adopted at Teesside Power Station was that the risk of a fatality to the operator at most risk from this specific hazard should not be greater than 7×10^{-6}/year. This is equivalent to a fatal accident rate (FAR) of 0.4. This is a criteria used by many companies in the chemical industry and is based on the philosophy that the level of risk should be no greater than the risk of a fatal accident occurring to a fit adult of working age at home in the UK. In practice, the application of this criteria over the recent past has resulted in high standards and procedures that have contributed to significant reductions in fatal accident rates in the chemical industry.

Although this was a quantitative process, HAZAN should not be seen as an exact science. It applies statistically derived average component and human failure data to specific and often individual components. However, HAZAN provides a quantitative input into the assessment of the tolerability of a risk. Similarly, the criteria tolerability of risk should only be seen as indicative, and should be considered against a background that risks are to be as low as reasonably practicable (ALARP).

Risk reduction

The initial hazard analysis at Teesside Power Station indicated that the risk of an explosion and fatality of an operator inside the enclosure was too high. It also indicated the major contributors to this risk. Some risk-reduction measures had already been recognized (e.g., reducing the incidence of leaks and installing reliable gas detection equipment). In order to achieve an acceptable level of risk the hazard analysis assisted in defining the required reliability of the new gas detection system. The proposed multichannel infrared gas detection system was studied to consider failure modes, common mode effects, and improvements to achieve a higher level of reliability. The significance of other contributors has been less obvious, for example, the specific risks associated with:

1. Permitting maintenance activity inside the enclosure with the turbine online
2. Operator error in responding to a gas detection alarm
3. Failure of the ventilation system

Teesside Power Station was able to identify and implement a range of changes to both hardware and operating practices to address the significant contributors to the overall risk of the hazardous event under consideration.

A final hazard analysis was then carried out to quantify the residual risk with the range of improvements in place and shown to be acceptable.

Case Study 2: Explosion Hazards at Gas Turbine–Driven Power Plants in the UK*

There are currently over 1050 on-shore UK CHP plants, of which about 10 percent are gas turbine based, with a total installed generating capacity of 2800 MW. A number of new large gas turbine–based plants and CCGT-based power stations are under construction or recently completed and not included in these figures. As part of its carbon dioxide emission reduction strategy, the UK government has a CHP goal of 5000 MW by the year 2000. Over 500 of the existing land-based units are rated at less than 100 kW, and are installed in schools, hotels, hospitals, and leisure centers. Any site with a large and fairly steady consumption of heat and electricity is likely to be suitable for CHP. Thus large industrial units, including over 150 plants rated at over 1 MW, are located in the oil and chemicals, food and drink, paper and board, and iron and steel sectors. This section is concerned mainly with larger gas turbine driven plant, but many of the matters discussed are relevant to smaller plants, and to plants driven by internal combustion engines.

Hazards

The fuel supply to a gas turbine is required at high pressure. While typically a 6 MW unit requires gas at 8 to 20 barg, 40-MW units operate at up to 30 barg, and some larger machines operate at up to 50 barg. Liquid fuel supplies, if present, may operate at up to 60 barg. The pipework supplying the fuel to the turbine combustion chambers is often highly complex, due to the need to supply from annular distribution manifolds to numerous individual combustion chambers. Each chamber may require several main fuel supplies to accommodate multistage burning for reduced emissions and alternative fuel supplies. For a 40-MW machine

---

* Source: Adapted from extracts from Santon, "Explosion Hazards at Gas Turbine Driven Power Plants," ASME paper 98-GT-215.

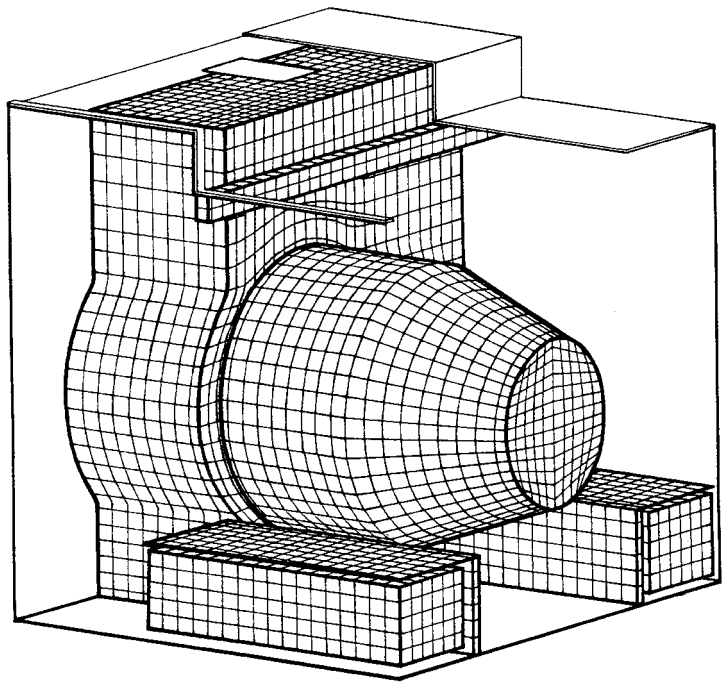


FIG. E-66 100 m^3 enclosure, computational grid.

the pipework may include 30 flanges or flexible pipes; for a 250-MW machine it may include over 200 flanges, 90 flexible hoses, 18 valves, and 8 bellows, all operating at 20 to 30 barg.

A particular problem associated with gas turbines arises from the absence of isolating valves on the fuel supply systems. The pipework is invariably connected to the combustion chambers without an isolating valve so that, while a blank can be inserted at this point for pressure testing of all upstream pipework, this final connection cannot be tested. In practice, while pipework is preassembled and tested by suppliers, it may not be tested when reassembled on site, in particular following maintenance, because of the difficulty of access. This is contrary to normal gas supply industry, chemical, or petrochemical plant practice, where any such complex pipework carrying hydrocarbon at such pressure would be rigorously tested whenever disturbed.

A fuel leak from the fuel supply pipework is foreseeable. It may arise following assembly, either when new or following maintenance, since the fuel pipework is routinely dismantled for turbine maintenance at intervals of one to three years. Gas turbines should operate without excessive vibration, and vibration detectors are often, but not always, fitted to larger units to detect bearing failure. Such vibration could also cause fuel pipe joint failure. Catastrophic, sudden guillotine, pipe failure is very improbable, but a fuel leak from a control valve, flexible pipe, or flanged, screwed or welded pipe joint is a hazard against which appropriate precautions should be taken.

Gas turbines are a significant noise source and environmental pressures often dictate their installation within acoustic enclosures. The release of fuel within the enclosure is potentially hazardous if the release results in a volume of flammable fuel/air mixture within a significant fraction of the chamber volume. The release may be of gas, such as natural gas, flashing liquid such as propane, or liquid such

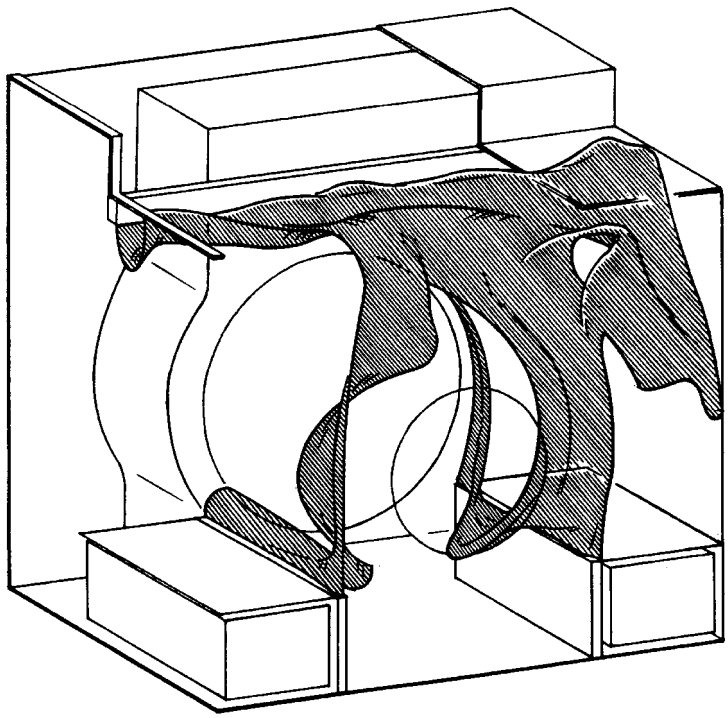


FIG. E-67 100 m^3 enclosure, 50 percent LEL ISO—surface before modification.

as naphtha or fuel oil. Even the release of liquids at temperatures below their flash points can create an explosion hazard because releases from high pressures can generate fine mist that behaves like a flammable gas. The ignition of such a gas or liquid mist fuel release would result, depending upon the size of the release and the construction of the chamber and its location, in explosion blast, fire ball, and missile effects that could seriously injure or kill persons nearby. The commercial implications of such an event are also likely to be very significant.

The ignition of a fuel release would require the presence of a source of ignition. Control over electrical sources of ignition in the flammable zone in the vicinity of foreseeable sources of release is the basis of area classification, but this concept may provide incomplete protection if, as in the case of turbine enclosures, other sources of ignition may be present. Ignition may result from static discharges, from mechanical means such as moving parts in the event of a major pipe failure (pipewhip) or blade enclosure failure, from mechanical disruption of electrical equipment, from back-flowing combustion gases in the event of a fuel pipe failure close to a combustion chamber, or from hot surfaces. Autoignition from hot surfaces is the most probable source of ignition of a fuel release. The fuel is being burned in turbine combustion chambers and the exhausts of some units based on aircraft engines glow. The exhaust diffusers of larger units do not reach such high temperatures but are nominally at 450 to 500°C. Thermal imaging techniques have been used to measure such surface temperatures and have identified hot spots of over 520°C. Under fault conditions higher temperatures may arise. Autoignition temperature is not an absolute property of a fuel, and is a function of surface roughness, orientation, contamination and size, and of fuel purity, stoichiometry, velocity, and turbulence. The literature values quoted for relatively common materials vary widely even for results obtained under standard conditions. Such values are unreliable for use with small margins in this application.

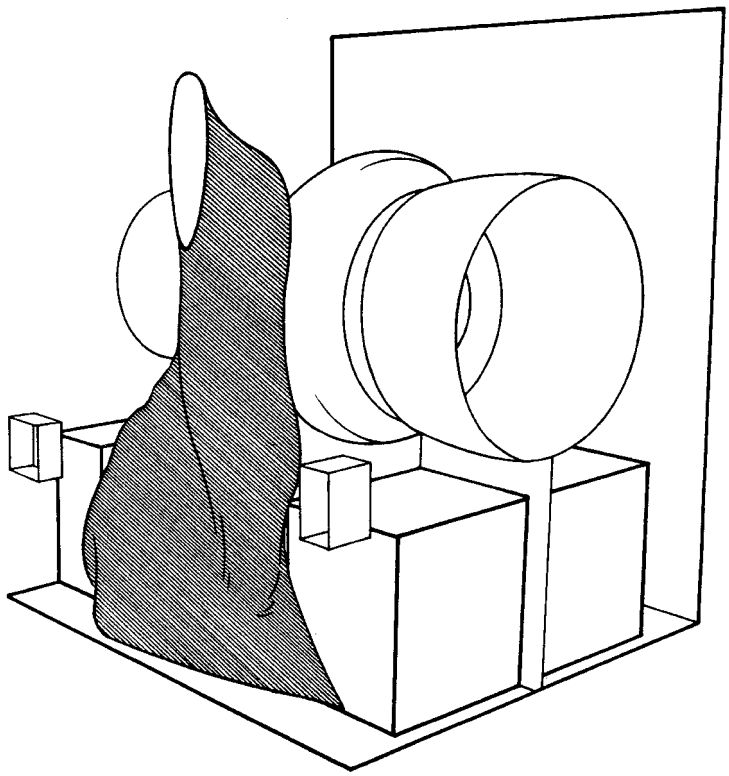


FIG. E-68 600 m^3 enclosure, 50 percent LEL ISO—surface before modification.

While it is clearly appropriate to take relevant precautions to minimize the presence of sources of ignition, it is not possible to eliminate them in these circumstances, and in the event of a fuel release within an acoustic enclosure the probability of ignition should be assumed to be high.

Apart from the hazard of an explosion within the acoustic chambers, there are other explosion hazards, characteristic of any gas-fired plant. In particular there is the possibility of the accumulation of a flammable gas/air mixture within the turbine and associated inlet and exhaust systems, and its ignition by the combustion process itself, e.g., at startup. This hazard is relatively easier to mitigate, with adequate purging and reliable gas safety shutoff arrangements.

Incidents

Following the Piper Alpha incident in the North Sea in 1988, and the recommendations of an inquiry held by Lord Cullen, an offshore hydrocarbon release database has been set up in the UK by the Offshore Safety Division of the Health and Safety Executive (HSE). Reports are mandatory and, as a result, this database represents an accurate picture. Most offshore installations include gas turbines and the database thus records incidents relating to them. Data from 5.5 years of operation has been analyzed, showing that there were 85 incidents within this period associated with fuel at gas turbines in general. Of these, 52 were ignitions and 4 resulted in explosions. One incident in particular was classified as of major potential.

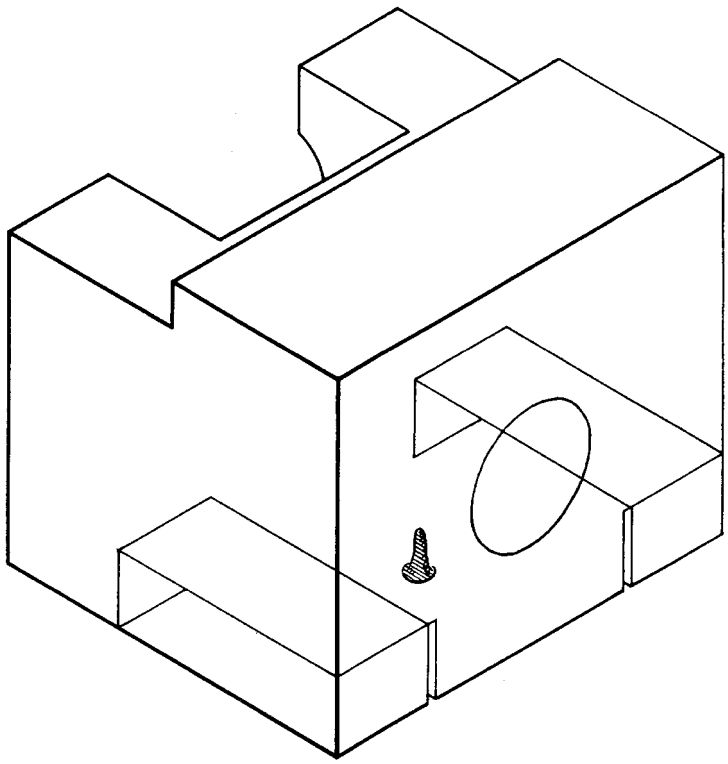


FIG. E-69 100 m^3 enclosure, 50 percent LEL ISO—surface after modification.

By comparison, taking into account the larger onshore population, the number of onshore incidents has been relatively low, perhaps reflecting the less arduous environment. There was a major explosion at a large CCGT plant in North East England in mid-1996.* Naphtha released from a pipe joint ignited at fuel changeover, and one man was seriously injured by the explosion that lifted the 600 m^3 acoustic chamber from its foundations. There have been a number of less serious incidents. Blade failure, resulting in casing rupture and a small fire, was reported at one site and a major fire at another following fuel line rupture. A fuel release on first start-up at another was traced to a loose flange. Other releases have occurred at fuel changeover. Many fires have occurred within compartments, and a survey by insurers reported 64 worldwide over 20 years. The discrepancy between the onshore and offshore incident frequency suggests that other nonreportable incidents may have arisen onshore. It is possible that there is a level of underreporting as many companies are unaware of the importance of reporting dangerous occurrences and the value of publicizing near misses.

There was a gas explosion in Holland in 1996 in which an acoustic chamber was damaged following a fuel leak. Other incidents in the U.S., France, Argentina, and Indonesia have been reported, but with little detail, and may be less relevant.

It is concluded, however, that the potential for explosions as a consequence of the identified hazards has been realized often enough in practice to make these hazards foreseeable and their mitigation necessary.

---

*See Case Study 1.

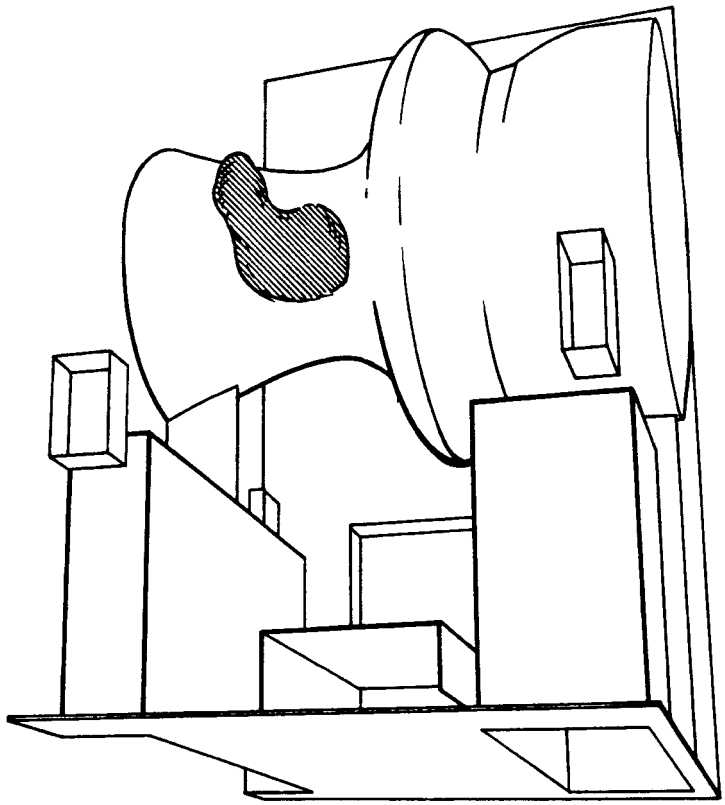


FIG. E-70 600 m^3 enclosure, 50 percent LEL ISO—surface after modification.

Standards

There are a number of standards that refer to gas turbines specifically, as well as others of general relevance. The following review only summarizes the codes insofar as they deal with the acoustic chamber and fuel supply explosion hazards.

National Fire Protection Association (NFPA) Code 37 is a fire protection standard but recognizes the explosion hazard and recommends the provision of explosion relief for turbine enclosures, or the provision of "ventilation adequate to prevent a hazardous accumulation of flammable vapours or gases . . ." Adequate ventilation is not defined further. In the case of an engine handling hazardous material other than its own fuel supply, i.e., a gas turbine–driven gas compressor, there is no alternative to the recommendation of explosion relief. The scope of this code is limited to engines and turbines not exceeding 7500 hp, i.e., 5.6 MW.

American Petroleum Institute (API) Code 616 is essentially a purchasing specification but includes some relevant recommendations. It requires exhaust system purging, an automatic vent on any gas fuel supply, and the minimum of flanges and flexible pipework. It makes no specific reference to ventilation or other explosion mitigation means. A later API code is directed specifically at packaged plant. It refers to acoustic enclosure ventilation as having a purging duty but gives no safety specific guidance on it, and it extends the fuel gas supply shutoff requirement to two valves and an automatic vent.

British Gas Code IM/24 is a broad code covering the whole installation from fuel supply to instrumentation. It refers to the need to provide adequate ventilation, and quotes 1 m^3/s as the minimum for gas leakage ventilation. It refers to gas

detection for some circumstances, but stresses that it should not be regarded as a substitute for good ventilation.

Institute of Petroleum (IP) Code 15 is a general area classification code for petroleum installations. It contains extensive discussion on the adequacy of ventilation, and a specific section on the ventilation and classification of turbine enclosures, or hoods, and the relevance of this ventilation to the prevention of the accumulation of flammable mixtures. Significantly it permits the enclosure to be classified as safe (unclassified) during normal operation if *dilution ventilation* is present, recognizing the effectiveness of properly designed ventilation in preventing accumulation.

American National Standards for Gas Turbines are published in 13 parts as ANSI B133. These are largely procurement standards but some contain relevant safety references. B133.3 (Procurement Standard for Gas Turbine Auxiliary Equipment) in particular recommends that ventilation should minimize the accumulation of hazardous atmospheres, while B133.7M (Gas Turbine Fuels) recommends stringent standards for the handling of low flash fuels.

UK legal position

An interim advice note has been produced by HSE covering all aspects of health and safety at these plants. Manufacturers and suppliers are subject to The Health and Safety at Work Act (HSWA) 1974, section 6(1)(a), which places a duty on them to ensure, so far as is reasonably practicable, that equipment is designed and constructed so that it will be safe and without risks to health at all times when it is in use or being cleaned or maintained by a person at work. They are also likely to be subject to the Supply of Machinery (Safety) Regulations 1992 (as amended), which implement The Machinery Directive, and require safety by design and manufacture, and to the Equipment and Protective Systems Intended for Use in Potentially Explosive Atmospheres Regulations 1996, which implement the Article 100A ATEX Directive and also require safety by design and manufacture.

Owners and operators are subject to HSWA 1974, sections 2 and 3 in particular, The Provision and Use of Work Equipment Regulations 1992 and The Management of Health and Safety at Work Regulations 1992, and The Confined Spaces Regulations 1997, all of which impose relevant duties.

Extraction, Liquid-Solid (*see also* Oil Sands)

This kind of extraction is a process where two or more solid constituents are separated, potentially even phase converted, by addition of a solvent or solvents, and the solvents later recovered or neutralized before disposal.

F

Fans, Centrifugal

Besides being used in process plants, centrifugal fans are common in building heating and air-conditioning systems. Vendors for these systems should be consulted for required options.

Types of Fans*

Three types of centrifugal fans are available: (1) forward curved blade fans, (2) radial blade fans, and (3) backward curved blade fans. The characteristics of the forward curved blade fan make it the most appropriate type of cooling tower service. By virtue of the direction and velocity of the air leaving the fan wheel, the fan can be equipped with a comparatively small size housing, which is desirable from a structural standpoint. Also, because the required velocity is generated at a comparatively low speed, forward curved blade fans tend to operate quieter than other centrifugal types. Table F-1 lists some typical fan installations. Note the relatively small rpm and head values. See also Figs. F-1 through F-4. The envelopes on the head versus volume curves each indicate a separate fan model is available.

Centrifugal fans are usually of sheet metal construction, with the most popular protective coating being hot-dip galvanization. Damper mechanisms are also available to facilitate capacity control of the cooling tower.

Fan laws

All propeller-type fans operate in accordance with common laws. For a given fan and cooling tower system, the following is true:

1. The capacity (cfm) varies directly as the speed (rpm) ratio, and directly as the pitch angle of the blades relative to the plane of rotation.
2. The static pressure (h_s) varies as the square of the capacity ratio.
3. The fan horsepower varies as the cube of the capacity ratio.
4. At constant cfm, the fan horsepower and static pressure vary directly with the air density.

If, for example, the capacity (cfm) of a given fan were decreased by 50 percent (either by a reduction to half of design rpm, or by a reduction in blade pitch angle at constant speed), the capacity ratio would be 0.5. Concurrently, the static pressure would become 25 percent of before, and the fan horsepower would become 12.5 percent of before. These characteristics afford unique opportunities to combine cold water temperature control with significant energy savings.

Selected formulas, derived from these basic laws, may be utilized to determine the efficacy of any particular fan application:

*Source: Garden City Fan & Blower Company, USA.

TABLE F-1 Centrifugal Fan Examples of Applications* (Source: Ansaldo)

Customer	Plant	Year	Application	Model Number Size Width	Wheel dia (mm)	Flow (m^3/sec)	Head (mm wg)	BHP (kW)	rpm
ENEL	Termini Imerese 3–4	1971	GR	4-1013 SWSI	2,946	106	447	658	735
TAVANIR	Esfahan 1–2 (Iran)	1977	FD	4-23120 DWDI	3,230	200	1,428	3,176	990
ENEL	Porto Tolle 1–4 (Italy)	1977	GR	8-1012 DWDI	2,964	187	385	1,765	990
TAVANIR	Esfahan 1–2 (Iran)	1977	GR	2-1012 DWDI	2,800	259	592	2,191	990
ENEL	Porto Tolle 1–4 (Italy)	1978	FD	8-4105 DWDI	2,827	313	1,320	4,670	990
A.Y.E.E.	San Nicolas (Argentina)	1981	ID	4-4109 DWDI	2,965	290.2	520	1,735	735
AGSM-ASM (VR)	Ponti S/Mincio (Italy)	1981	FD	2-23101 DWDI	2,712	93.6	1,164	1,235	963
ENEL	La Spezia 3–4 (Italy)	1983	ID	4-4126 DWDI	3,488	551	660	4,321	740
ENEL	Brindisi 1–2 (Italy)	1986	ID	4-4154 DWDI	4,083	610	656	4,600	590
ENEL	Fiume Santo 3–4 (Italy)	1988	PA	4-2375 DWDI	2,030	70.45	1,483	1,180	1,490
TECHIMPORT	Dagang 3–4 (China)	1988	ID	4-4109 DWDI	2,965	290.2	520	1,735	735
ENEL	Brindisi 3–4 (Italy)	1988	ID	4-4154 DWDI	4,083	610	656	4,600	590
INECEL	Esmeraldas (Ecuador)	1990	FD	1-4081 DWDI	2,092	145	988	1,600	1,180
SNAM	Enichem-P. Torres (Italy)	1990	ID	1-126/3 DWDI	3,280	147.65	800	1,550	740
ENEL	Tavazzano 3–4 (Italy)	1990	GR	4-114/4 DWDI	3,040	142.55	665	1,435	980
S.W.C.C.	Al Jobail 1–6 (S. Arabia)	1990	FD	1-22385 DWDI	2,238	105.7	924	1,085	1,175
AGSM	Brescia (Italy)	1990	FD	1-2395 SWSI	2,510	64.3	852	614	980
TAVANIR	Bisotoun 1–2 (Iran)	1991	FD	4-23120 DWDI	3,170	197.6	1,360	3,155	988
ENEL	Sulcis 1–3 (Italy)	1991	DESOX*	6-23143 DWDI	3,778	271.1	957	2,960	730
ENEL	Fiume Santo 3–4 (Italy)	1991	DENOX	4-23135 DWDI	3,566	260.8	732	2,214	738
TAVANIR	Bisotoun 1–2 (Iran)	1991	GR	2-1012 DWDI	2,714	276	516	2,170	980
ENEL	Torrevaldaliga 2 (Italy)	1992	GR	1-1008 DWDI	2,180	116.6	430	765	990
E.E.A.	Assyut (Egypt)	1992	FD	2-23109 DWDI	2,762	178.2	1,081	2,190	980
ENEL	Brindisi (Italy)	1992	DESOX*	8-4154 DWDI	3,970	568	773	5,100	584
ENEL	Fusina 5 (Italy)	1992	GR	1-109/3 DWDI	2,900	148.6	675	1,400	284
ENEL	Fusina 3–4 (Italy)	1992	DESOX*	4-4120 DWDI	3,230	291	825	2,860	735
ENEL	Turbigo (Italy)	1993	ID	4-4105 DWDI	2,707	245.2	400	1,160	745
ENEL	Brindisi 12 (Italy)	1993	DENOX	4-4146 DWDI	3,840	605	910	6,330	745
ENEL	Montalto 1 (Italy)	1993	ID	2-4154 DWDI	4,083	610	656	4,600	590
T.E.K.	Tuncbilek 45 (Turkey)	1993	ID	4-1014 DWDI	3,036	278	500	2,040	730

* "C" position (wet–low-temperature position).

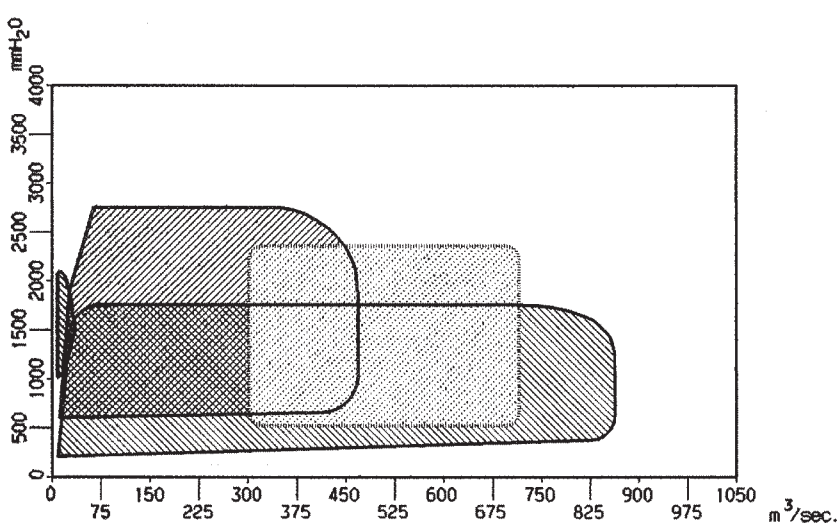


FIG. F-1 Typical fan blade–type performance range. (Source: Ansaldo.)

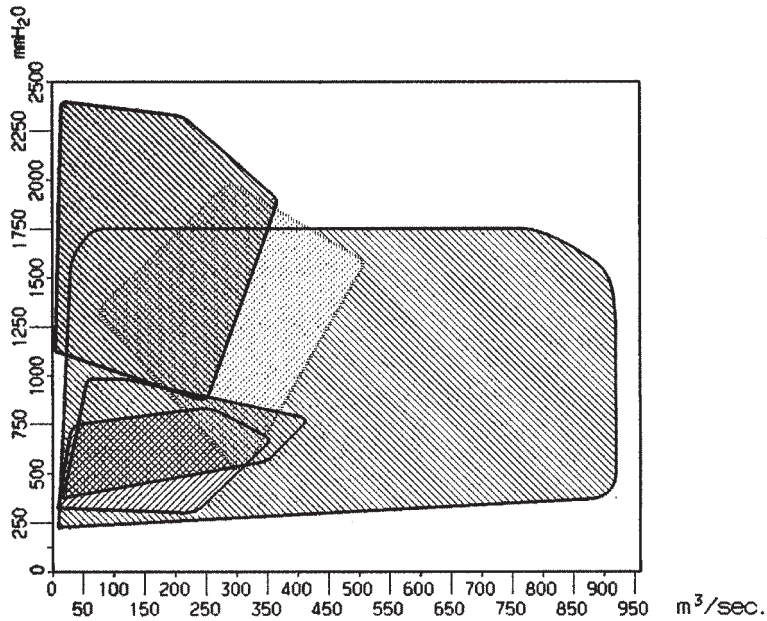


FIG. F-2 Typical single-thickness fan blade–type performance range. (Source: Ansaldo.)

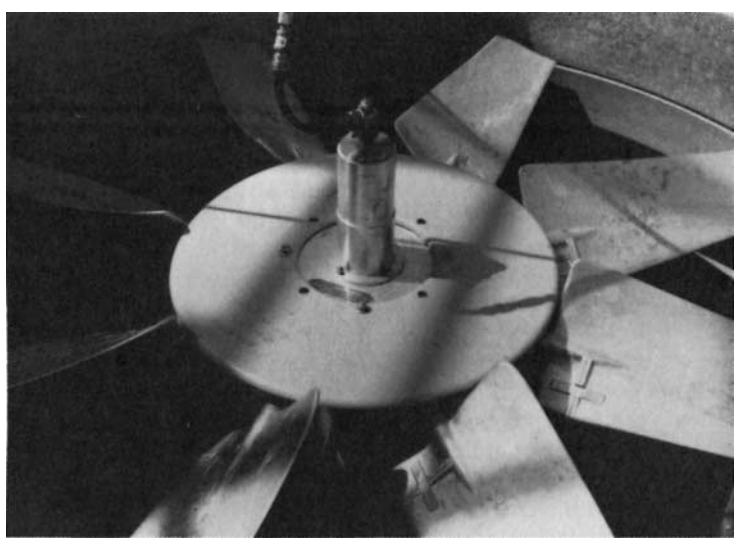


FIG. F-3 Automatic, variable-pitch fan used to adjust airflow and fan horsepower. (Source: The Marley Cooling Tower Company.)

Symbols

Q = Volume of air handled (cfm). Unit: ft^3/min.
A = Net flow area. Unit: ft^2.
V = Average air velocity at plane of measurement. Unit: ft/s.
g = Acceleration due to gravity. Unit: 32.17 ft/s^2.
D = Density of water at gauge fluid temperature. Unit: lb/ft^3.
d = Air density at point of flow. Unit: lb/ft^3.
h_s = Static pressure drop through system. Unit: inches of water.

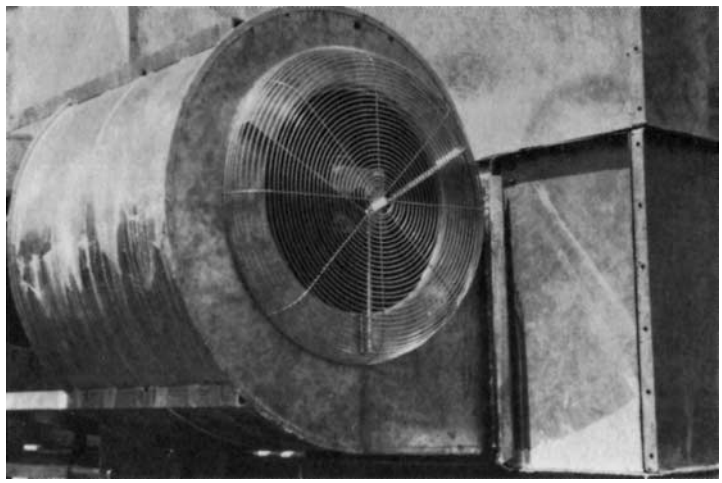


FIG. F-4 Blower-type cooling tower fan. (Source: The Marley Cooling Tower Company.)

h_v = Velocity pressure at point of measurement. Unit: inches of water.
h_t = Total pressure differential (= $h_s + H_v$). Unit: inches of water.
v_r = Fan cylinder velocity recovery capability. Unit: percent.

Thermal performance of a cooling tower depends on a specific mass flow rate of air through the fill (pounds of dry air per minute), whereas the fan does its job purely in terms of volume (cubic feet per minute). Since the specific volume of air (cubic feet per pound) increases with temperature, it can be seen that a larger volume of air leaves the tower than enters it. The actual cfm handled by the fan is the product of mass flow rate times the specific volume of dry air corresponding to the temperature at which the air leaves the tower. This volumetric flow rate is the Q used in the following formulas, and it *must* be sufficient to produce the correct mass flow rate or the tower will be short of thermal capacity.

Utilizing appropriate cross-sectional flow areas, velocity through the fan and fan cylinder can be calculated as follows:

$$V = \frac{Q}{A \times 60}$$

It must be understood that A will change with the plane at which velocity is being calculated. Downstream of the fan, A is the gross cross-sectional area of the fan cylinder. *At* the fan, A is the area of the fan *less* the area of the hub or hub cover.

Velocity pressure is calculated as follows:

$$h_v = \frac{V^2 \times 12 \times d}{2 \times g \times D}$$

If V represents the velocity through the fan, then h_v represents the velocity pressure for the fan itself (h_{vf}). Moreover, if the fan is operating within a nonflared-discharge fan cylinder, this effectively represents the total velocity pressure because of no recovery having taken place.

However, if the fan is operating within a flared, velocity-recovery-type fan cylinder (Fig. F-5), h_v must be recalculated for the fan cylinder exit (h_{ve}), at the appropriate velocity, and applied in the following formula to determine total velocity pressure:

$$h_v = h_{vf} - [(h_{vf} - h_{ve}) \times v_r]$$

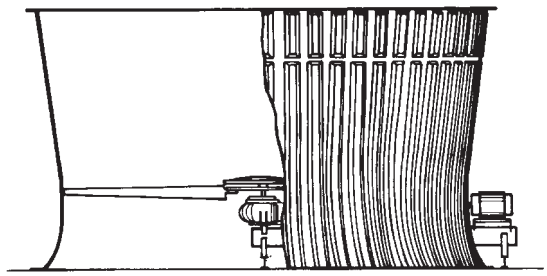


FIG. F-5 Cutaway view of a velocity-recovery-type fan cylinder. (Source: The Marley Cooling Tower Company.)

Although the value of v_r will vary with design expertise and is empirically established, a value of 0.75 (75 percent recovery) is normally assigned for purposes of anticipating fan performance within a reasonably well-designed velocity-recovery cylinder.

The power output of a fan is expressed in terms of air horsepower (ahp) and represents work done by the fan:

$$\text{ahp} = \frac{Q \times h_t \times D}{33{,}000 \times 12}$$

Static ahp is obtained by substituting static pressure (h_s) for total pressure (h_t) in the formula.

A great deal of research and development goes into the improvement of fan efficiencies, and those manufacturers that have taken a systems approach to this research and development effort have achieved results that, although incrementally small, are highly significant in light of current energy costs. Static efficiencies and overall mechanical (total) efficiencies are considered in the selection of a particular fan in a specific situation, with the choice usually going to the fan that delivers the required volume of air at the least input horsepower:

$$\text{Static efficiency} = \frac{\text{static ahp}}{\text{input hp}}$$

$$\text{Total efficiency} = \frac{\text{ahp}}{\text{input hp}}$$

It must be understood that input hp is measured at the fan shaft and does not include the drivetrain losses reflected in actual motor brake horsepower (bhp). Input up will normally average approximately 95 percent of motor bhp on larger fan applications.

Reference and Additional Reading

1. Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Filters (*see also* Air Filtration; Separators)

Filters are common in agricultural and pharmaceutical industries. A filter manufacturer should be consulted for catalogs on the filter element or medium

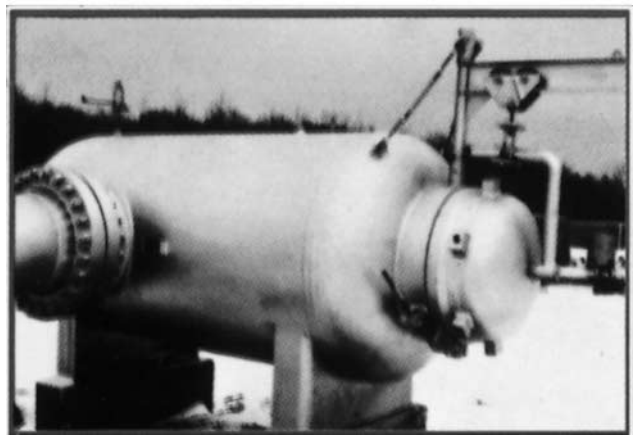


FIG. F-6 Small closure of filter opens quickly. (Source: Peerless.)

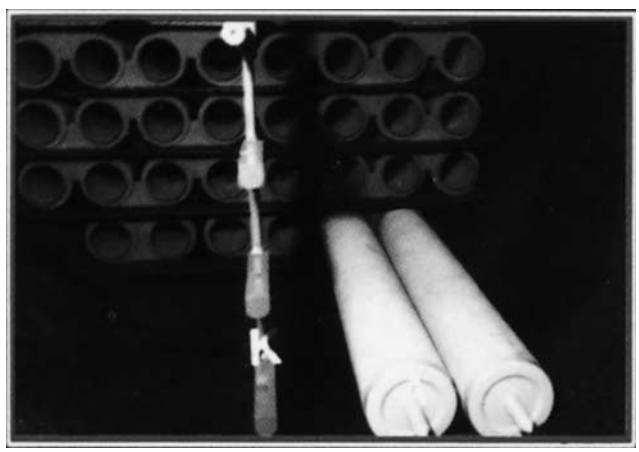


FIG. F-7 Filter cartridges are placed on temporary shelf and rolled into place. (Source: Peerless.)

options available for any application. A vast array of possibilities exist, depending on whether filtration requirements are intermittent or continuous, partial or absolute, and whether toxic, corrosive, or erosive materials are handled.

One of the most common varieties of filter to be used by the process engineer is the gas filter. Other common applications of filtration include separating liquid and gas (*see* Separators) and oil system filters in turbomachinery (see machinery operating manuals). A description of an in-line process gas filter follows.

In-line Process Gas Filter*

Filter cartridge replacement is simple. In gas filters of 30-in shell diameter and larger, the cartridges are generally locked in place by means of a patented retainer assembly. See Figs. F-6 through F-12.

This assembly is constructed of large, rugged, easily handled parts. It eliminates nuts, washers, and other small parts that are time consuming to handle and that

*Source: Peerless, USA. Adapted with permission.

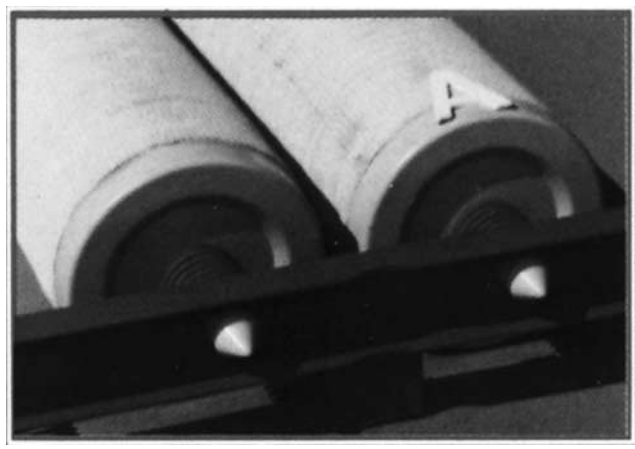


FIG. F-8 Support bars with attached pressure springs are fitted over bullet noses of cartridges. (Source: Peerless.)



FIG. F-9 Lower tier of filter cartridges is completed. (Source: Peerless.)

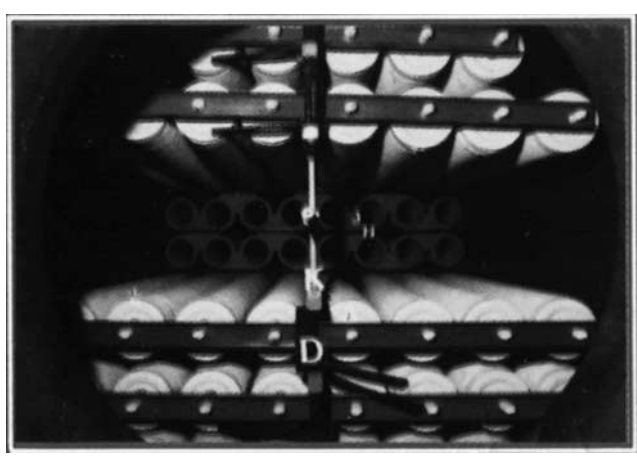


FIG. F-10 Additional tiers of filter cartridges are completed. (Source: Peerless.)

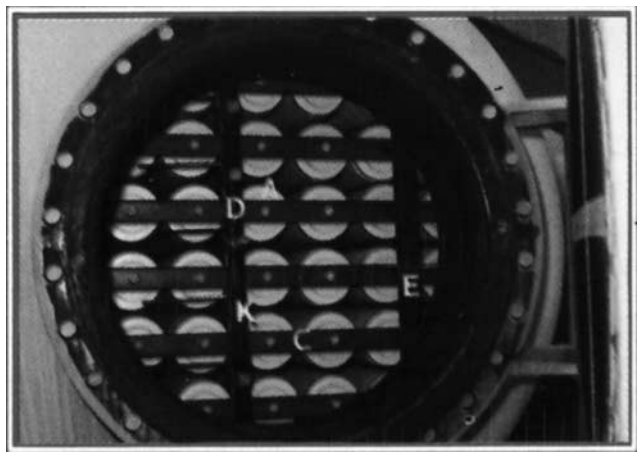


FIG. F-11 Vertical support bars are in place and large wing nut (shell diameters 42 in and larger) is tightened. (Source: Peerless.)

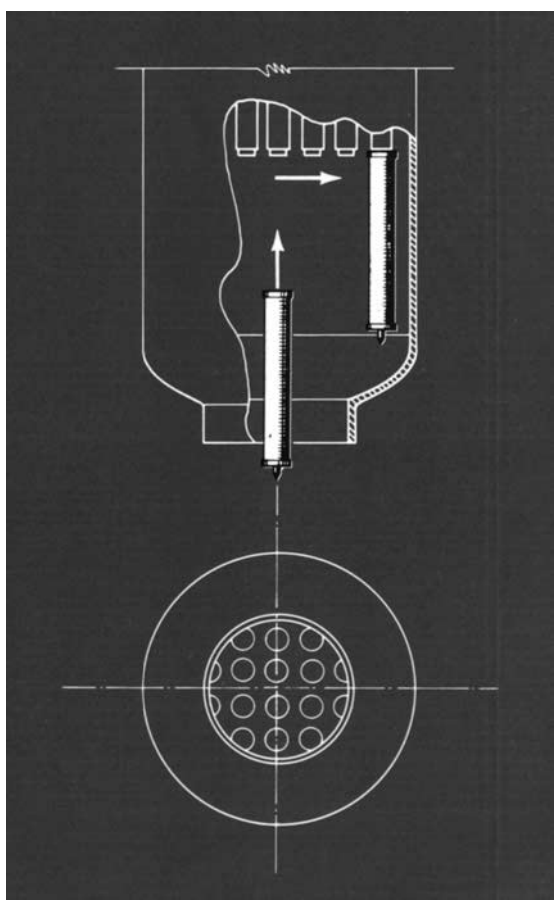


FIG. F-12 Cartridge removal. (Source: Peerless.)

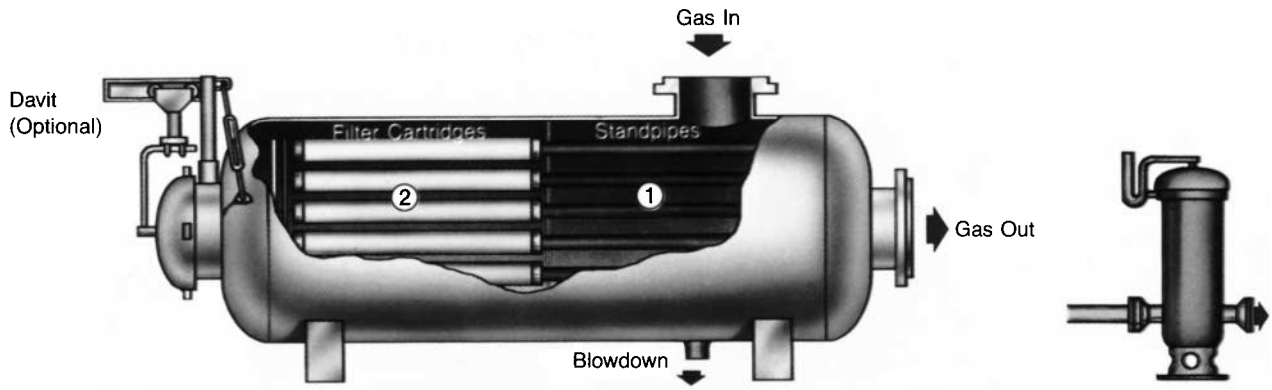


FIG. F-13 Series MT horizontal multitube filter. (Source: Peerless.)

are easily dropped and lost. For example, a 60-in ID vessel made by this information source has only 20 removable parts (not counting the filter cartridges) as opposed to 235 parts in a conventional design.

There are no support arm threads to become stripped, and no support arms. This avoids the difficult job of removing a collapsed filter cartridge from a support arm.

The design of the retainer assembly allows all maintenance work to be done through a small closure (below), with substantial savings in first cost as well as in man-hours in removing and resealing the closure.

Gas filters are designed for both operating efficiency and clean-out efficiency. To make the cartridge replacing operation as quick and simple as possible, a quick-demountable filter cartridge retainer assembly is used in all vessels of 30-in shell diameter and larger. The result is a substantial saving in both man-hours and down-time whenever cartridges are replaced.

Not all foreign matter from the gas stream is deposited on the filter cartridges. In any gas filter there is a gradual buildup of rust, scale, and other gas line dirt on the inner surfaces of the shell, especially at the bottom.

This phenomenon is used by directing the incoming gas into a generously sized inlet chamber (① in Fig. F-13), where the abrupt drop in velocity allows the heavier particles to fall out of the gas stream by gravity before they reach the filter cartridges. Thus the filter cartridges' dirt-holding capacity is used to best advantage in trapping the hard-to-catch fine particles.

The material collected in the shell is removed from gas filters. There are no longitudinal support arms for the filter cartridges, so that when the cartridges are removed, the filter chamber (② in Fig. F-13) is empty of all structural parts. The standpipes that connect to the cartridge outlets are arranged in a square rather than a diagonal pattern, with plenty of room between them for dirt to fall through to the bottom of the shell. In addition, the cartridges and standpipes are positioned well above the bottom, clear of accumulated dirt.

Because of the design of the cartridge retainer assembly and location of the cartridges themselves, all of this servicing can be done by one person, standing *outside* the vessel.

Sizing and performance

For this original equipment manufacturer's (OEM) product range, for instance, series MT multitube filters are built in sizes from 6-in through 144-in shell diameter.

They are usually selected for a pressure drop of $^1/_4$ to 2 psi (clean), although they can be selected or designed for other pressure drops.

Multitube filters remove 100 percent of all particles larger than 3 microns in diameter and 99 1/2 percent of all particles 1/2 to 3 microns.

Construction

Each filter cartridge consists of a small-diameter tube made up of closely packed glass fibers and covered with a knitted cotton sock. A perforated metal liner stiffens the cartridge and maintains its inside diameter against the inward-acting pressure of the gas.

The shell is constructed in accordance with the latest edition of the ASME Code—with stamp. A resident National Board licensed inspector is on hand at all times to check each phase of construction.

Optional features

Customized construction can be provided to meet various state codes and customer specifications. An external bypass with rupture disc can be added. A davit can be installed to facilitate removing the closure head. Alternative metals can be used instead of the standard carbon steel.

In-line Gas Filters*

In-line gas filters (see Fig. F-14) generally provide the same gas-handling performance as (series MT for this information source) multitube filters. They are built in two configurations—vertical (VGF) and horizontal (HGF). (VGF and HGF are both model designations specific to this manufacturer.)

The inlet and outlet nozzles are opposite each other, on a common centerline, near one end of the vessel. This arrangement simplifies connections and allows installation in piping runs that are too short for other filter designs.

Sizing and performance

VGF and HGF filters are built in sizes from 6-in through 24-in shell diameter. They are usually selected for a pressure drop of 1/4 to 2 psi (clean). Like the MT filters, they are guaranteed to remove 100 percent of all particles larger than 3 microns diameter and 99 1/2 percent of all particles 1/2 to 3 microns.

Construction

Each filter cartridge is made up of closely packed glass fibers covered with a knitted cotton sock. The cartridge is stiffened by a perforated metal liner. It is held in place by a longitudinal rod and wingnut.

The shell is ASME Code stamped, as in the series MT. The O-ring closure allows quick access to the interior.

Optional features

A davit can be installed to facilitate removing the closure head. A mounting base (cylindrical in the VGF, angle iron in the HGF) can be added.

---

*Source: Peerless, USA. Adapted with permission.

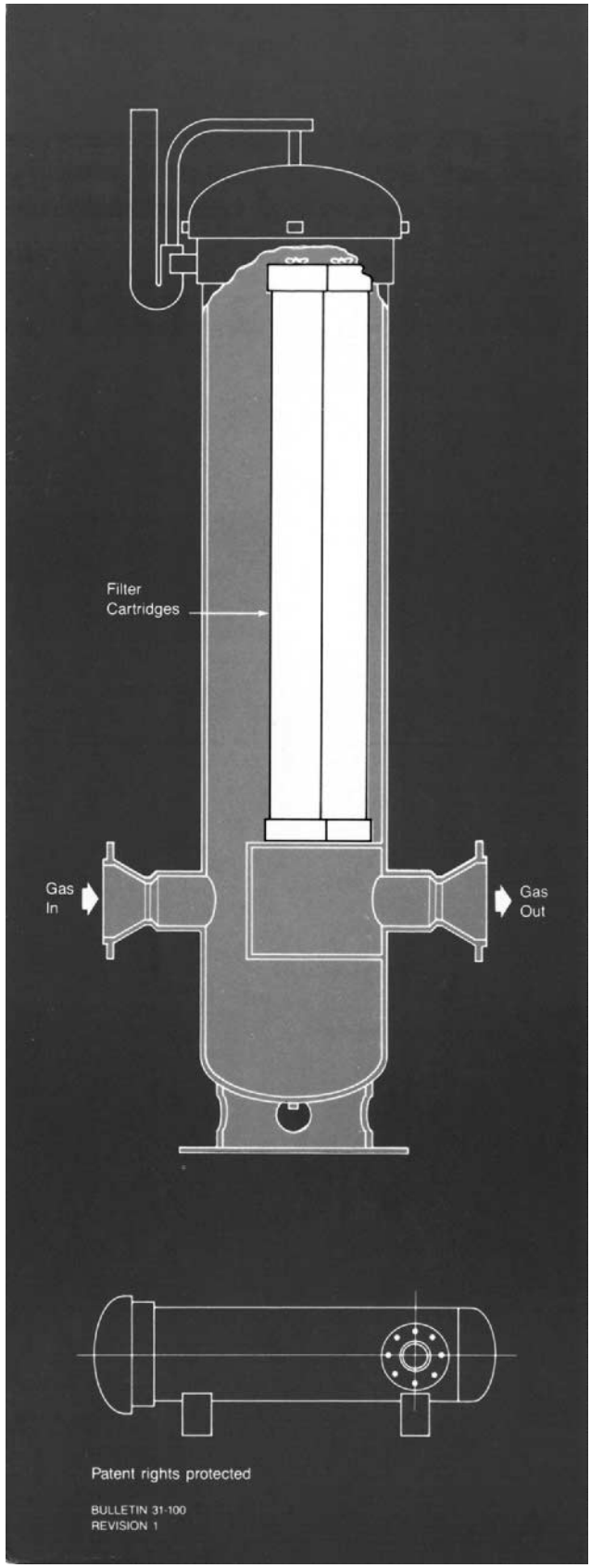


FIG. F-14 In-line gas filter. (Source: Peerless.)

Flare Stacks (*see also* Stacks)

A *flare stack* is a stack that conducts vented gases that are then lit at its top exit by a burner and ignition system (which has its own fuel supply line). The stack has to be a minimum distance (worked out as a function of substances being flared and their heat-release rates) from the rest of the plant for the purpose of preventing fires, ignition, and secondary damage to other equipment such as storage tanks.

Floating Covers

In the petrochemical and chemical industries, evaporation losses from stored liquid product can be sizable and costly. When these products are stored in tanks, they can be covered with a movable floating cover that will eliminate evaporation. Tank manufacturers can supply suitable covers for the application in question (*see* Tanks).

Fluidized Bed

In a large-scale reaction process, a catalyst is often used to assist the reaction. To maximize catalyst surface area, the catalyst may be pulverized and then held in suspension (fluid suspension or fluidized bed). *See* Reactors.

Forest Products

These products form part of the agricultural sector in that they require that good agricultural practices, such as selective log felling and timely seedling planting, be followed so that the raw material resource can be sustained. The forest products industry is also in a position to inadvertently or otherwise affect many other industries and quality-of-life factors. For instance, it has been determined that the damage done to much of the Canadian west coast salmon stock was a result of high levels of lumber mill sawdust that clogged salmon spawning grounds.

Deforestation that occurs if logging is not done selectively has severe consequences, including loss of wildlife habitat (*see* Ecosystem) with consequential loss of revenue for personnel who work in these areas and soil erosion. It is widely accepted that had severe deforestation occurred in the rain forest around the Panama Canal, the canal would have been irretrievably clogged with eroded soil soon after.

The forest products industry, therefore, is an industry that could be thoroughly interwoven into the community in which it is located. As an illustration, the following extracts adapted from the 1997 AssiDomän company report regarding its corporate environmental objectives are included here. Further understanding of this material is evident by reading the sections on Environmental Economics and Pulp and Paper; and the total material summaries, the material on the contemporary forest and timber industry, and the packaging materials production data in this section.

Contemporary Forest and Timber Industry Practices*

In many countries of the world, the forest products industries carry on as before, with developers hastily cutting, clearing, and burning down forest to make their

*Source: AssiDomän, Sweden. Adapted with permission.

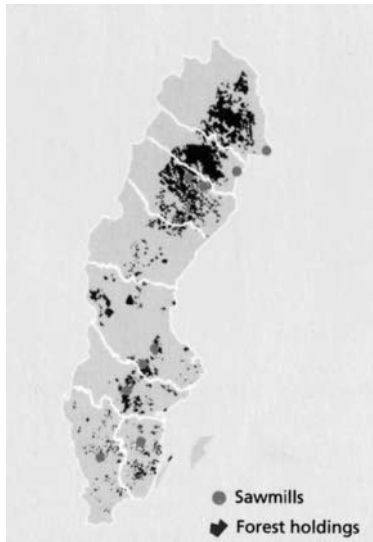


FIG. F-15 AssiDomän operations regions in Sweden. (Source: AssiDomän.)

products. This form of production is not sustainable. In the United States and Canada, consumption of forest materials in industry is still high compared to the western European countries that aim at maximizing the output from forest products, thereby minimizing their use and maintaining other practices that help ensure their raw material supply will be sustainable.

Case study 1: Forestry management at AssiDomän

This case describes typical, sophisticated, state-of-the-art western European management of the raw materials required for forest products. See also Tables F-2 and F-4 through F-6 and Figs. F-15 through F-17. Note that measures involving emissions reduction take on considerably more significance in a country with NO_x and SO_x taxes (like Sweden).

Forestry work

- All forest management regions work in accordance with the Environmental Management System (EMS) standard ISO 14001. This means that they have established a local environmental policy as well as goals and programs for their environmental work. EMS registration was not possible for forestry until 1997, so work is currently focused on ISO 14001.
- The forest management regions in Kalix, Älvsbyn, Lycksele, Örebro, and Värnamo were approved for certification in accordance with ISO 14001. See Fig. F-16.
- The two timber units started work with the introduction of EMS.
- More than half of AssiDomän's productive forest land area, 1.7 million hectares, was approved during 1997 in accordance with the Forest Stewardship Council's (FSC) Swedish standard for certification of forestry.
- Ecological landscape plans will be drafted for the entire forest holdings by 2002.

TABLE F-2 Activities on Corporate Environmental Objectives

Corporate Environmental Objectives	1997 Outcome	Further Work
To certify the Group's forest holdings in accordance with FSC's standard during 1997.	More than half of the forest holdings were certified during the year.	The remaining forest holdings are expected to obtain certification during the first half of 1998.
That all units will have begun the work on introducing EMS by year-end 1997.	Approx. 90% of the units have begun this work. Some 15 units were approved for certification according to ISO 14001 and/or EMAS registration.	Some 15 units are expected to be certified in 1998 and the remainder in 1999 and 2000.
To implement cost-effective environmental measures based on life-cycle assessments.	Life-cycle assessments are an important part of the environmental work and have been used in product development and in choice of transport mode.	The environmental objective remains. The Group's environmental work will be pursued in a holistic perspective.
To achieve ecocycle compatibility in the Group's operations. Ash restoration in large-scale tests.	An "ash restoration group" has been formed to promote the recycling of ashes and residual products to the forest. A field study has been started.	Field studies and development projects continue.
To reduce the use of fossil fuels, mainly at the Group's pulp and paper mills.	The environmental objective was not achieved at most mills in 1997. At some mills use has increased as a result of production increases and higher prices for biofuels than oil (due to national energy tax policy) and rebuilding work in the plants. The biofuel boiler at AssiDomän Frövi was rebuilt, permitting increased biofuel use. Energy-saving measures have been implemented at AssiDomän Skärblacka so that oil use will be reduced by 9000 m^3 per year.	The environmental objective remains. As a result of modernizations and other strategic investments in energy and recovery systems in the mills, the use of fossil fuels is expected to decline progressively up to the turn of the century. A black liquor gasification plant is being designed at AssiDomän Kraftliner. This will yield higher energy efficiency and increase electricity production from biofuel.
To collaborate with carriers in projects aimed at greater environmental compatibility and shorter transport distances.	Collaboration with carriers in transport of both timber and finished products. Reduced specific transport volume has been achieved by scheduled route transport. Environmental questionnaires are used to evaluate carriers. Environmental calculations comprise part of the basis for choice of transport mode.	The environmental objective remains. Work is being pursued in prioritized areas.
To promote the development of lighter weight and more resource-efficient packaging materials and packaging.	Several new packaging solutions with common environmental profiles were developed during the year. Reduced packaging weight is an important factor in this work.	Working toward the environmental objective is integrated in the daily work of developing new products.
To participate in and carry to completion the joint sectoral environmental research project within SSVL.	AssiDomän has been represented in the management of the project, which will be concluded in 1998.	AssiDomän will actively participate in continued activities within the sector's environmental research.

SOURCE: AssiDomän.

- New directions for precommercial thinning were put into use during the year. Regeneration planning prior to final felling was finished in time for the 1998 field season.
- This source imported 11 percent of the pulp and paper industry's wood needs in the form of roundwood and chips during 1997. The imported wood comes mainly from the Baltic states and otherwise from Finland, Russia, Germany, and Brazil.

TABLE F-3 Production Data on Barrier Coating

	1997		1996	
	Total	Per Tonne End Product	Total	Per Tonne End Product
Paper raw material, tonnes/kg	131,700	881	110,400	870
Plastic, tonnes/kg	26,800	179	25,500	200
Fossil fuels, TJ/GJ	108	0.7	120	0.9
Purchased electricity, GWh/MWh	44	0.30	32	0.25
SO_2, tonnes/kg[1]	1	0.007	1	0.006
NO_x, tonnes/kg[1]	19	0.13	9	0.07
Production of plastic-coated paper, tonnes	149,500		127,000	
By-products to recycling, tonnes	5,450		8,850	

1. Estimated values.

TABLE F-4 Total Material Summary of AssiDomän Operations: Forest Management Regions (8 Units)

	1997 Total	1996 Total
Raw material use		
Harvested own forest, excl. standing timber, 1000 sm^3ub	6,190	5,930
of which harvested with own machinery, 1000 sm^3ub	3,600	3,760
of which harvested by contractors, 1000 sm^3ub	2,590	2,170
Purchased forest, 1000 sm^3ub	1,160	980
of which harvested with own machinery, 1000 sm^3ub	120	50
of which harvested by contractors, 1000 sm^3ub	1,040	930
Total wood raw material, 1000 sm^3ub	7,350	6,910
Oils, m^3	730	625
of which vegetable, m^3	240	ND
Fertilizer, tonnes N	360	350
Energy use		
Total fossil fuels, own machinery, TJ	245	445[3]
Emissions to air[1]		
SO_2, tonnes	0.2	0.7
NO_x, tonnes	320	615
CO_2 fossil, tonnes	18,100	33,200
Waste		
Hazardous waste, m^3 [2]	156	145
Recovered spent oil, %	32	23
Land		
Total land area, 1000 ha	4,49[4]	4,430
Productive forestland, 1000 ha	3,280	3,320
Cultivated forestland, 1000 ha	3,030[5]	3,090
Finally felled area, ha	26,800	28,700
Products		
Wood to AssiDomän, 1000 m^3fub	5,020	3,690
Wood, sold externally, 1000 m^3fub	2,330	3,220
Total wood delivered, 1000 m^3fub	7,350	6,910

ND = No data available.

1. From own machines. 2. Collected oil. 3. Over available estimated values. 4. Incl. 123,000 ha not inventoried previously. 5. Certain areas have been redefined in conjunction with ecological landscape planning.

TABLE F-5 Total Material Summary of AssiDomän Operations: Sawmills and Wood Processing (11 Units[1])

	1997		1996	
	Total	Per m^3 Sawn Timber	Total	Per m^3 Sawn Timber
Raw material use				
Wood, 1000 sm^3ub/sm^3ub	2,380	2.2	2,090	2.1
Oils, m^3/dm^3 [2]	233	0.21	200	0.20
Externally supplied energy				
Fossil fuels, TJ/GJ	90	0.082	73	10.073
Purchased thermal energy, TJ/GJ[3]	194	0.18	234	0.23
Purchased electricity, GWh/MWh	96	0.088	92	0.092
Emissions to air[4]				
SO_2, tonnes/kg	33	0.030	37	0.037
NO_x, tonnes/kg	179	0.16	142	0.14
Particles, tonnes/kg	78	0.071	95	0.095
CO_2 fossil, tonnes/kg	6,810	6.2	4,850	4.9
CO_2 biogenic, tonnes/kg	97,500	89	82,600	83
Residual products and waste				
Hazardous waste, m^3/dm^3 [5]	45	0.04	60	0.06
Landfilled material, m^3/dm^3	16,000	15	24,000	24
Products				
Sawn timber, 1000 m^3	1,100		1,000	
By-products for external delivery				
Sawdust, bark, chips, 1000 m^3/m^3	1,210	1.1	1,020	1.0
Pulp chips, 1000 m^3/m^3 [6]	2,050	1.9	1,820	1.8
Heat, GJ/MJ	7,100	6.4	1,200	1.2

1. Of which one unit was sold as per 31 March 1997 and one as per 31 December 1997. 2. Lubricating and hydraulic oils. 3. Based on biofuels. 4. Only from own energy generation. 5. Collected oil and chemical residues. 6. Raw material for the pulp industry.

Continued environmental work. All forest management regions and timber units were certified according to ISO 14001 during the first half of 1998. At the same time, AssiDomän's entire holdings of productive forest land were FSC-certified.

With the introduction of EMS, each unit set up environmental objectives. Examples of areas covered by the objectives are: reduced usage of fossil fuels, improved waste management and resource management, and programs for the preservation of biodiversity in the forest landscape.

The FSC certification affects AssiDomän's forestry practices in a number of ways. Some of the more important changes are the requirements on an increased hardwood fraction and controlled burning on forest land, and the fact that more deadwood should be left. Besides consideration for the environment, the FSC's criteria also include economic and social consideration. One example of the latter is that consideration be given to reindeer herding.

Ecological balance sheet. For the purpose of evaluating how well the forest-related objectives have been achieved, internal follow-up has been done since 1993 in the

TABLE F-6 Total Material Summary of AssiDomän Operations: Transport Activities

	1997		1996[3]	
	Total	Per tonne-km	Total	Per tonne-km
Transport volume[1]				
Road, mill. tonne-km	2,860		1,770	
Sea, mill. tonne-km	5,230		4,800	
Rail, mill. tonne-km	1,670		1,240	
Total, mill. tonne-km	9,760		7,810	
Energy use				
Fossil fuels, TJ/MJ	3,420	0.35	2,330	0.30
Electricity, GWh/kWh	68	0.007	57	0.007
Emissions to air				
SO_2, tonnes/g	1,670	0.17	1,710	0.22
NO_x, tonnes/g	3,630	0.37	3,610	0.47
CH, tonnes/g[2]	225	0.023	185	0.024
CO_2 fossil, tonnes/g	254,000	26	181,000	23

1. Return trips only included for system traffic at sea. 2. Hydrocarbons. Common key figure for transport emissions. 3. Excl. Central and Eastern Europe.

form of an ecological balance sheet. The balance sheet is a good aid in the further training of forestry workers. See also Figs. F-18 through F-22.

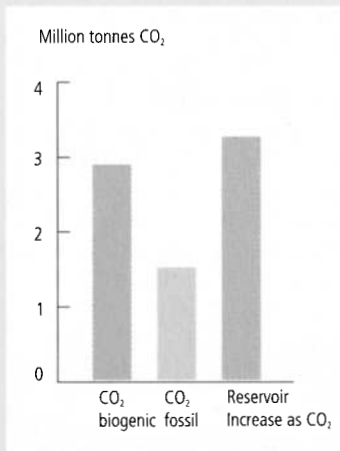
The grades for final felling show that the former trend of improvements from year to year has been improved. This is true for site-adapted forestry as well as for nature and water conservation. The number of evaluated categories, as well as the requirements for getting the highest grade for the individual site, have gradually increased as knowledge has grown and the criteria for FSC certification have taken firmer shape. An example of a new category for the year 1997 was evaluation of whether the nature conservation measures were concentrated and targeted correctly within the felled site. Only 44 percent of the final felling area received the highest grade in this respect. This is an important, but difficult, category that will require further training.

On more than 16 percent of the evaluated sites, so many trees have been left on the productive land that an appreciable portion of these trees are not judged to contribute to better functional nature conservation. Here the production target was disregarded, resulting in a lower yield.

The 1997 balance sheet showed that on 88 percent of the final-felled area, the best possible combination of felling method, tree species, and regeneration method were chosen.

The thinnings in the 1997 balance sheet showed no great changes in grades compared with 1996. Water conservation improved slightly, however. The removal of hardwood trees in the thinnings decreased since the introduction of the new thinning instruction in 1996. However removals were still too high to meet the long-range target of a 10 percent hardwood fraction of standing timber.

Precommercial thinning was included in the ecological balance sheet starting in 1998: It is important for tying together the follow-up of stand management during the entire rotation period.



AssiDomän's carbon dioxide emissions derive from biofuels and fossil fuels. The carbon dioxide from biofuels, CO_2 biogenic, which constitutes the greater part, is bound in the growing forest and therefore does not influence the concentration of CO_2 in the atmosphere. The carbon dioxide that is emitted by combustion of fossil fuels in forestry, pulp and paper mills, and transport does, however, contribute to the greenhouse effect.

Growth during 1997 in AssiDomän's forests, including forest reserves, has been estimated at 10.6 million m^3fo. Since felling amounted to 7.6 million m^3fo, the net reservoir increase in the forest biomass was 3.0 million m^3fo.

The above chart shows that the reservoir increase, counted as bound quantity of CO_2, far exceeds the quantity of CO_2 fossil which AssiDomän emitted during 1997. Overall, the Group's operations thereby contribute towards reducing the concentration of CO_2 in the atmosphere.

FIG. F-16 Carbon dioxide balance. (Source: AssiDomän.)

Approximately 14 percent of the productive forest land area is given special treatment for nature conservation purposes in connection with both final felling and thinning. This is a reasonable level for preserving and strengthening biodiversity.

Timber

The business area's nine sawmills and wood-processing units are required to obtain permits under the Environmental Protection Act.

The plants impact the environment mainly through emissions from biofuel combustion, surface runoff from timber storage, noise from production and transport, and waste management.

The sawmills produce by-products in the form of chips, sawdust, and bark. These materials are used as raw materials in pulp and particleboard manufacture or as biofuels, either at the sawmills or externally.

Environmental work list for a sample year (1997)

- Environmental management systems were introduced at six of the nine units during the year. The main reason why EMS have not yet been introduced at three of the units is extensive restructuring of the operations.

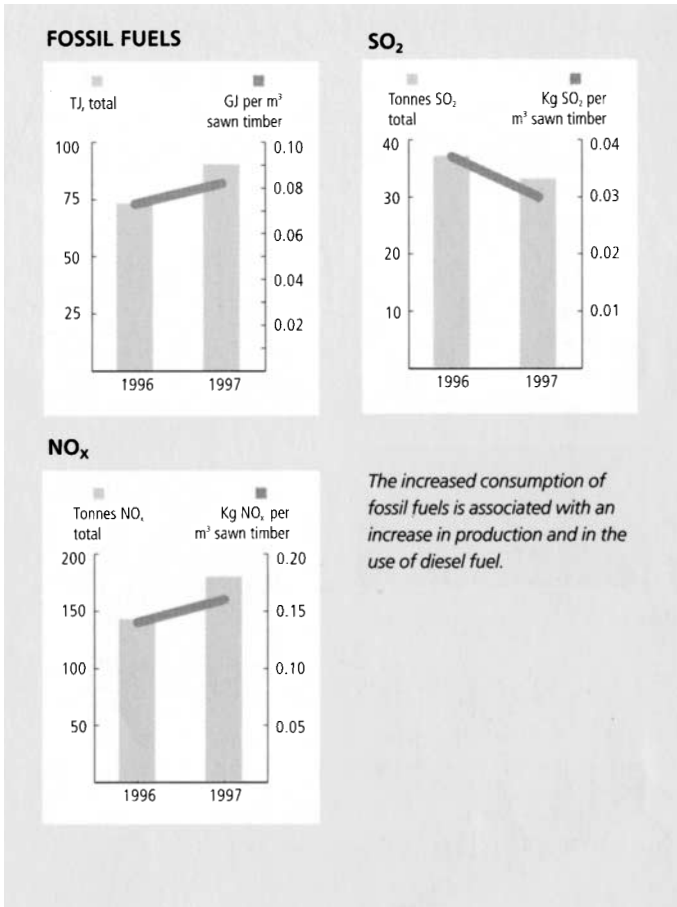


FIG. F-17 Sawmills and wood processing. (Source: AssiDomän.)

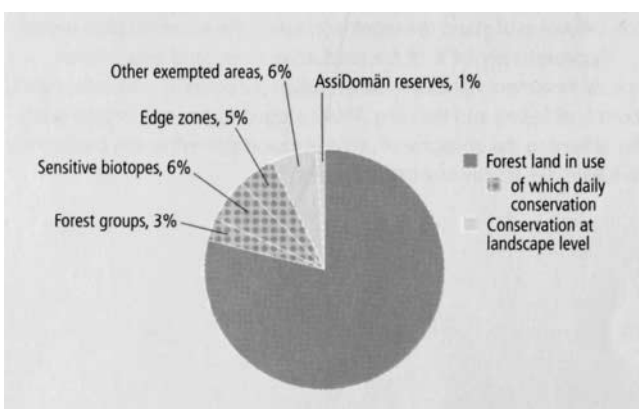


FIG. F-18 Nature conservation measures on AssiDomän's productive forest land. (Source: AssiDomän.)

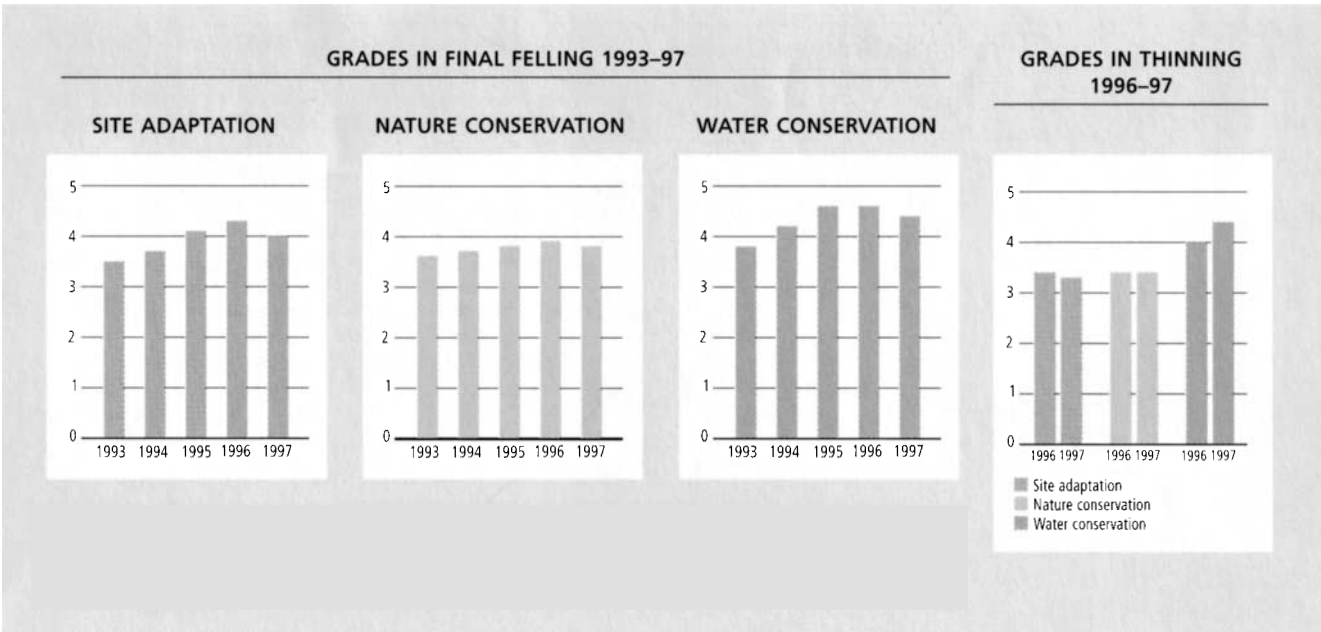


FIG. F-19 The evaluation of each final felling and thinning results in three different grades: site-adapted forestry (how well the wood-producing capacity of the different growing sites is used), nature conservation, and water conservation. Grades are awarded on a five-point scale, where 3 is satisfactory and 5 is the highest grade. (Source: AssiDomän.)

- Niab Hestra and Hasselfors Timber were certified in accordance with ISO 14001.
- Environmental training was carried out for all employees at six units as a step in the introduction of EMS.
- Two units, Hasselfors Timber and Lövholmen, were certified during the year with regard to "chain of custody" for production of sawn timber from raw material from FSC-certified forestry. This means that the origin of the timber can be guaranteed through the whole production chain.
- Environmental incidents occurred at some of the units during the year. Examples include exceeding prescribed noise level and improper handling of contaminated industrial sites. The incidents were reported to the appropriate authority and remedial plans were prepared and initiated.
- Noise-suppression measures were adopted at three units.
- Climate-controlled timber watering was installed at three units, leading to reduced water consumption and thereby reduced surface water runoff.
- Measures were adopted at four units to reduce emissions of nitrogen oxides and particulates. For example, a new solid fuel boiler with cyclone ejector and electrostatic filter was installed at one sawmill and a wet scrubber at another.
- Waste management was improved by increased source separation at five of the units.

Environmental work scheduled for a typical year (1998). In accordance with ISO 14001, the units adopted environmental objectives for their operations. Examples of such objectives are to reduce the quantity of spillage and waste, to reduce emissions from combustion, to improve oil and fuel management, to adopt noise suppression measures, and to reduce effluent discharges.



The special design of the Starpot modules with air gaps permits natural growth of the plant's root system.

Seed and Plants

AssiDomän's three nurseries – Kilåmon in Ångermanland, Lugnet in Uppland, and Hillet in Västergötland – produced a total of 41 million plants in 1997.

Work on the introduction of EMS was pursued at the nurseries during 1997. Certification according to ISO 14001 is expected to be obtained in the first half of 1998. Environmental policies, objectives, and action programs were prepared for the units' environmental work. Minimizing the consumption of input materials in production, source separation of waste, and reducing the transport volume by means of more efficient packaging technology are examples of environmental objectives at the nurseries.

A new plant growing system, Starpot, began to be introduced in 1997 at the nursery in Kilåmon, which accounts for two-thirds of the Group's production of plants. The new system will lead to reduced root deformation, more vital plants, and more efficient packing and transport of plants out to the forest. The new system was introduced at all nurseries by year-end 1998.

FIG. F-20 Seed and plants management. (Source: AssiDomän.)

Packaging Materials

A closer look at the manufacture of two common packaging materials enables a better understanding of the need for environmental policy to develop as an integral part of increased design sophistication.

Barrier coating

There is a great need for fiber-based packaging with some kind of barrier against moisture or grease. The environmental work within the business area is being focused on the development of packaging solutions where new material

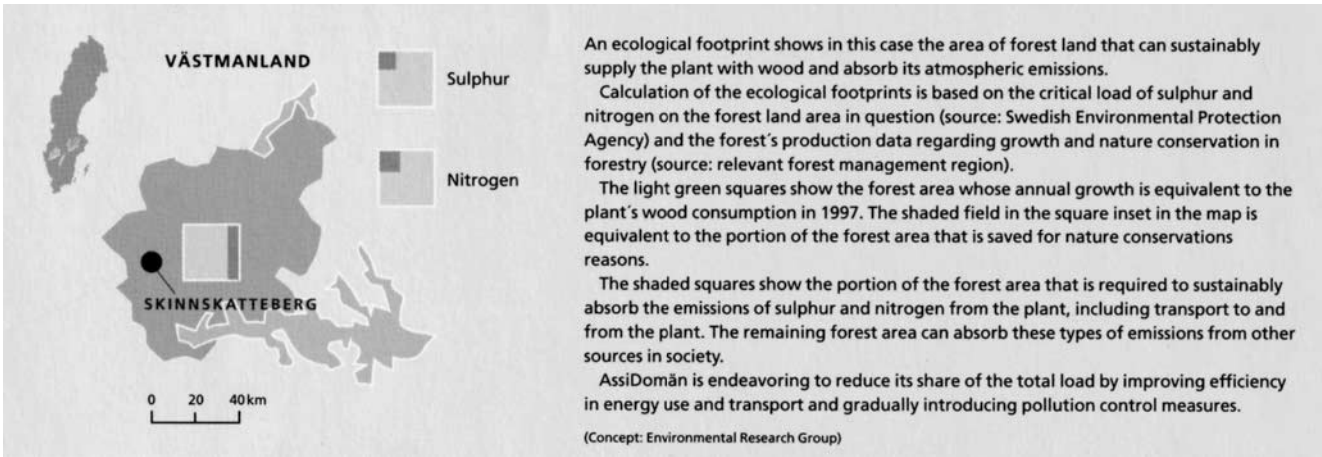


FIG. F-21 An ecological model "footprint." (Source: AssiDomän.)



FIG. F-22 AssiDomän's first delivery of FSC-labeled sawn timber products went to England in November 1997. The FSC trademark indicates that the products come from well-managed forests, independently certified in accordance with the rules of the Forest Stewardship Council (SGS-C-0081). (Source: AssiDomän.)

combinations permit more efficient resource utilization and where the barrier coating method is environmentally compatible. See Table F-3.

The environmental issues given top priority are:

- *Solvent-based inks, barrier chemicals, and adhesives.* The possibilities of replacing these with water-based products are being explored.
- *Waste and residual product management.* The possibilities of utilizing reject and production waste to a greater extent for energy recovery and material recycling are being explored. For example, the plant in Timrå has a well-functioning system for conversion of combustible process waste to marketable

TABLE F-7 Production Data on Sacks

	1997	
	Total	Per 1,000 Sacks
Paper raw material, tonnes/kg	156,000	209
Fossil fuels, TJ/GJ	177	0.24
Purchased electricity, GWh/MWh	38	0.05
SO_2, tonnes/kg*	9	0.012
NO_x, tonnes/kg*	18	0.024
Landfilled waste, tonnes/kg	3,190	4.3
Sack production, tonnes	745	
By-products to recycling, tonnes	5,660	

* Estimated values.

energy raw material, and several plants have installed systems for collection or incineration of solvents.

- *Noise from plants and transport.* Noise abatement measures are being implemented at several plants.
- *Energy use.* All mills are using exclusively fossil fuels as a thermal energy source. Most of the fuels consist of fossil gas. Oil will be replaced wherever possible. Environmental targets for reduced energy use are established annually.
- *Transport.* A large portion of the business area's environmental impact comes from transport operations. Ways to improve transport are being investigated.

Sack manufacture

Sacks are made from paper based on unbleached or bleached sulfate pulp. The sacks are mainly used for packaging building materials, food, chemicals, and animal feed. (See Table F-7.)

The trend is toward lower paper weights per unit volume and fewer layers in the finished sack, at the same time as the strength requirements on the paper have risen. A mixture of recycled fiber, which is weaker, is rare. A new grade with approximately 10 percent better strength properties was launched in 1997.

Lower packaging weights mean lower raw material, energy, and transport needs. This also means reduced emissions to air and water.

The introduction of environmental issues systems (EMS) is under way and has come farthest at the German plants, which achieved environmental certification/registration. See Fig. F-23.

Prioritized environmental issues in connection with sack manufacture are:

- *Inks and adhesives.* The introduction of ink rooms will reduce consumption of ink and waste as well as associated transport. Water-soluble inks and adhesives will gradually be introduced.
- *Wastewater, polluted with inks and adhesives.* Existing treatment systems will be optimized and new ones introduced.
- *Energy use.* Reducing energy consumption is an environmental objective at many mills. Fossil gas is the principal source of thermal energy. At Sepap, steam from the nearby pulp and paper mill is used as a source of heat.
- *Waste management.* Training of personnel and good control has led to a reduction of process waste. Waste plans exist at many plants.

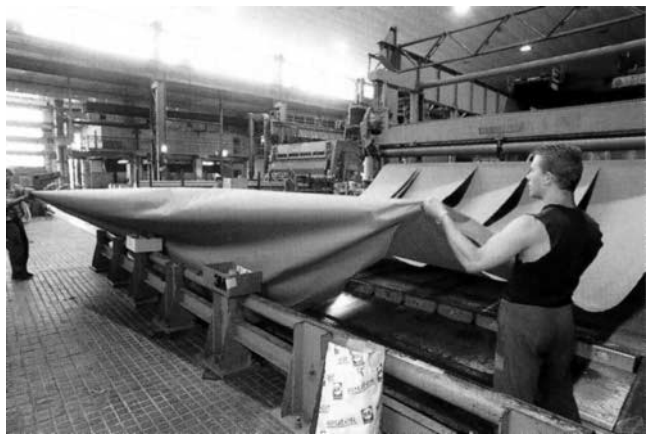


FIG. F-23 At AssiDomän Sepap, one of Europe's biggest manufacturers of sack paper and sacks, pollution abatement measures have been approved for a total of MSEK 200. (Source: AssiDomän.)

- *Noise*. Abatement measures have been introduced at plants in Italy and Germany.

Carton material production data (production site: AssiDomän Frövi)

- The aerated lagoon was converted into a long-term aerated activated sludge plant. Treatment efficiency with regard to both phosphorus and COD is expected to increase considerably compared with the previously used treatment plant. When the converted treatment plant was put into operation, a large quantity of sludge was released from the old treatment plant. The discharges exceeded the permit guidelines.
- A number of internal process measures were introduced. This will lead to reduced discharges of COD to water, reduced emissions of particulates and sulfur dioxide to air, and reduced discharges of odorous substances. See also Fig. F-24.
- Production of TCF-bleached pulp, in which bleaching takes place without chlorine-containing chemicals, began in 1997.
- A study was made of the animal life on the bottom of Lake Väringen. The results show that bottom conditions have improved steadily over the past few decades.
- In connection with the development of the EMS, contractors who work at the mill have received training in environmental practices.

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.

Fuel Gas Conditioning System(s)

One of the most important features of a fuel system is that it be acceptably free from moisture or liquid droplets (if the fuel is gaseous) and other impurities (if the fuel is gaseous or liquid). A typical system is outlined here.*

*Source: Peerless, USA. Adapted with permission.





● Pulp and paper mill

Production:

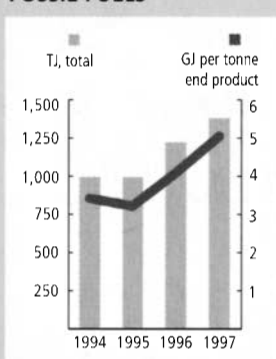
market pulp, tonnes	56,900
cartonboard, tonnes	215,000
Net turnover, MSEK	1,498
Average number of employees	792

Production unit:
AssiDomän Frövi

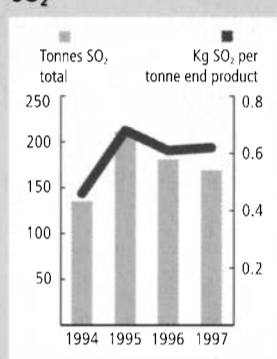
The business area manufactures coated cartonboard at the mill in Frövi, Sweden. More than half of the production consists of liquid packaging board, the remaining volume being packaging board for food.

KEY ENVIRONMENTAL DATA FOR FRÖVI 1994–1997

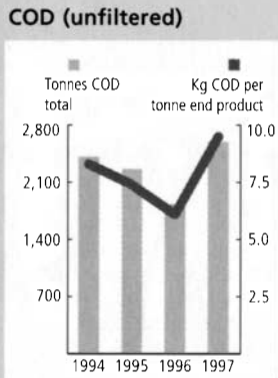
FOSSIL FUELS



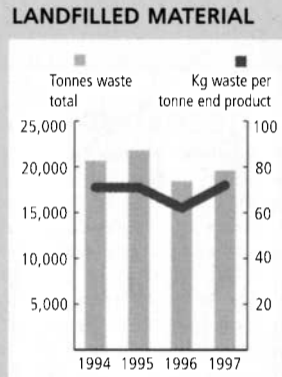
SO_2



COD (unfiltered)



LANDFILLED MATERIAL



The use of fossil fuels has increased due to the fact that oil replaced bark during the rebuild of the solid fuel boiler. The increased discharge of COD is associated with the incident at start-up of the mill and treatment plant.

FIG. F-24 Key environmental data for AssiDomän Frövi. (Source: AssiDomän.)

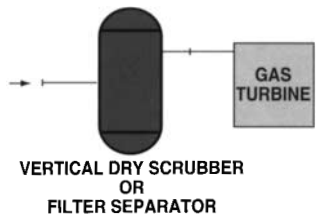


FIG. F-25 Position of vertical dry scrubber upstream of gas turbine. (Source: Peerless.)

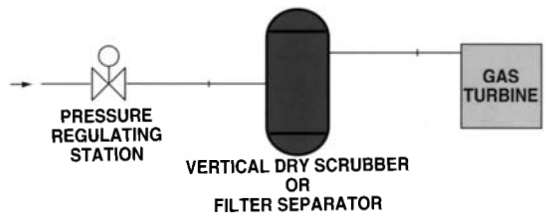


FIG. F-26 Pressure regulating station (PRS) upstream of vertical dry scrubber. (Source: Peerless.)

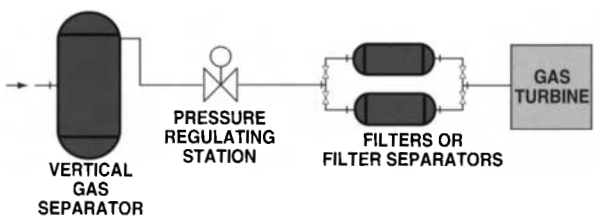


FIG. F-27 Vertical gas scrubber upstream of pressure regulating station. (Source: Peerless.)

When gas pressure regulation and heating is not required, a filter separator or vertical dry scrubber is recommended to remove the entrained solid and liquid contaminants from the gas upstream of the turbine (see Fig. F-25). The vertical dry scrubber is a high-efficiency multicyclone separator with constant pressure drop and no replaceable internals. Due to a limited turndown ratio, it is recommended that a separate scrubber be installed upstream of each gas turbine. When specified, or necessitated by variable flow conditions, a filter separator is often used. Since the elements must be periodically replaced, a duplex configuration consisting of two 100 percent units is recommended when protecting two or more gas turbines. As with all of our fuel gas conditioning systems, full instrumentation for local and/or remote monitoring can be supplied for either manual or fully automatic system operation.

When gas is received from a transmission line at a much higher pressure than required by the turbine, pressure regulating stations are included in the system design as shown in Figs. F-26 through F-30. The simplest package consists of a single skid with one or more 100 percent capacity pressure regulating valves (PRV) placed upstream of the separator(s). This is shown in Fig. F-26.

In most cases, the PRVs must be protected from liquid slugs, which could be present in the gas supply line. A vertical gas separator is the primary choice to remove both liquid slugs and mist. Duplex filters or filter separators are located

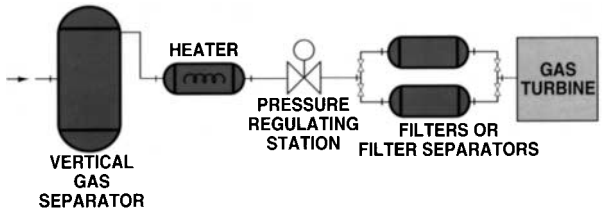


FIG. F-28 Heater downstream of vertical gas scrubber. (Source: Peerless.)

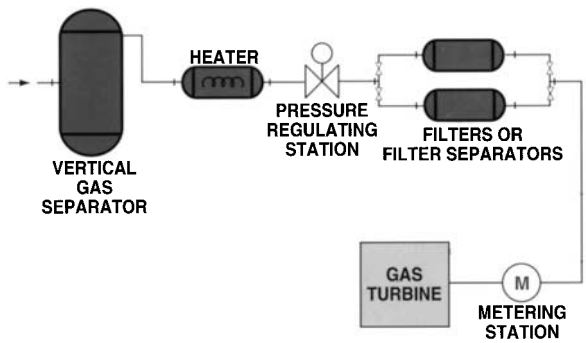


FIG. F-29 Metering station upstream of gas turbine. (Source: Peerless.)

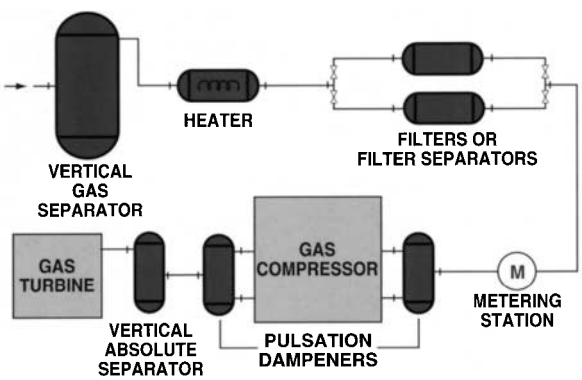


FIG. F-30 Pulsation dampeners upstream and downstream of compression station. (Source: Peerless.)

downstream of the pressure regulating station to complete the system as shown in Fig. F-27.

Figures F-28 and F-29 show the addition of heating and metering. When practical, all equipment is mounted on a single skid to facilitate ease of erection at the site. If individual turbine metering is a requisite, manufacturers supply each metering station on a separate skid that is easily installed in the field.

Figure F-30 is an example of a system requiring compression of the gas upstream of the turbine. Some manufacturers offer a precompression gas conditioning package, as well as pulsation dampeners and a high-efficiency vertical absolute separator downstream of the compressor. Some manufacturers also offer both analog and digital simulations to solve dampening and vibration difficulties.

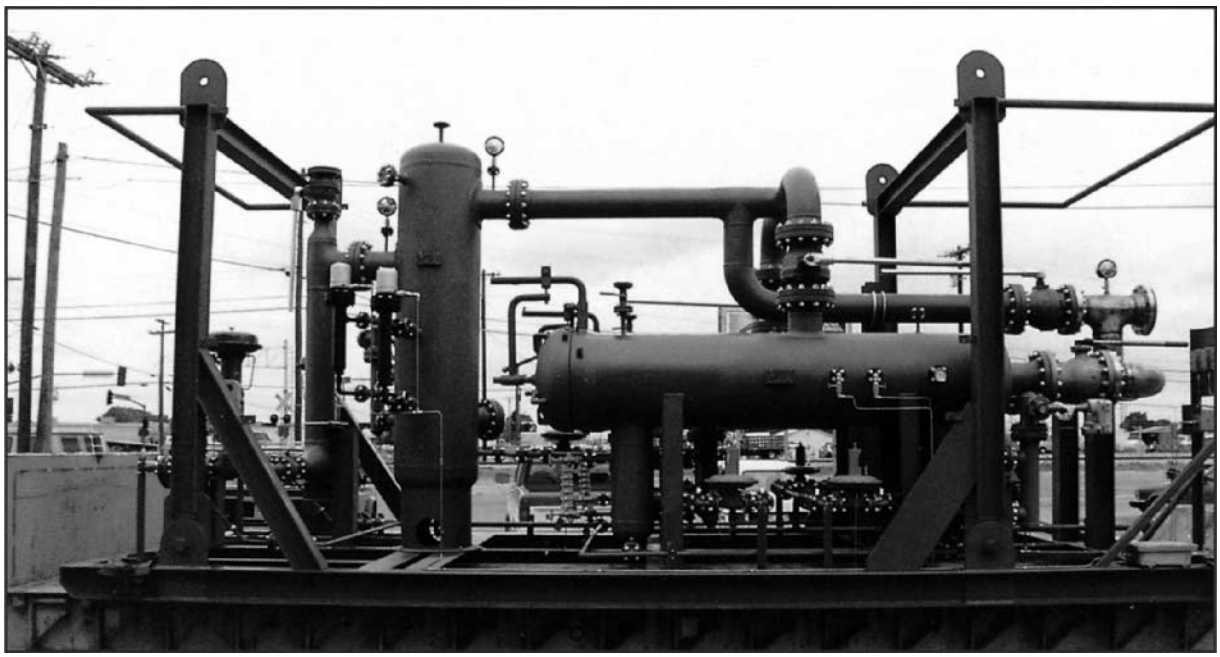


FIG. F-31 A vertical gas separator followed by two duplexed horizontal multitube filters is used to protect two gas turbines in Nigeria. Piping and valves downstream of filters are stainless. (Source: Peerless.)

Typical Range of Gas Conditioning Separators*

Vertical gas separators

Definition: Vertical vane-type separator with inlet baffle for high liquid loads with liquid removal efficiency of 100 percent for particles of 10 microns and larger. See Fig. F-31.

Filter separators

Definition: Vertical or horizontal two-stage separators for removal of solids and liquids with liquid removal efficiency of 100 percent for particles of 8 microns and larger and 99.5 percent for particles of 0.5 to 8 microns.

Solids removal efficiency: 100 percent for particles that are 3 microns and larger and 99.5 percent for particles that are 0.5 to 3 microns.

Multitube filters

Definition: Vertical or horizontal single-stage filters for removal of solids.

Solids removal efficiency: 100 percent for particles that are 3 microns and larger in size and 99.5 percent for particles of 0.5 to 3 microns in size.

Vertical dry scrubbers

Definition: Vertical multicyclone separator for removal of solids and liquids with liquid removal efficiency of 100 percent for particles of 10 microns and larger.

---

*Source: Peerless, USA. Adapted with permission.



FIG. F-32 System installed in Saudi Arabia includes one vertical dry scrubber followed by two pressure regulating valves and a line separator. A condensate drain tank is mounted alongside. (Source: Peerless.)

Solids removal efficiency: 100 percent for particles of 8 microns and larger, 99 percent for particles of 6 to 8 microns, 90 percent for particles of 4 to 6 microns, and 85 percent for particles of 2 to 4 microns. See Fig. F-32.

Vertical absolute separators

Definition: Vertical single or two-stage separator for removal of solids and very fine mists with liquid removal efficiency of 100 percent for particles 3 microns and larger, and 99.98 percent for particles less than 3 microns.

Solids removal efficiency: 100 percent for particles of 3 microns and larger, and 99.5 percent for particles of 0.5 to 3 microns.

Line separators

Definition: Vertical vane-type separator with liquid removal efficiency of 100 percent for removal of particles 10 microns and larger.

Fuel Systems; Fuel Flow Control

One* of the most common types of fuel flow control is electrohydraulic control. There are electrohydraulic control solutions for differing environments, including low-pressure and potentially explosive conditions.

*Source: J.M. Voith GmbH, Germany. Adapted with permission.

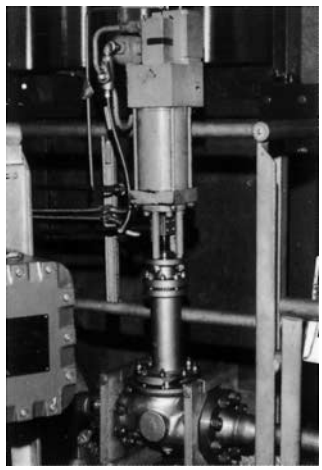


FIG. F-33 Electrohydraulic actuator in a U.S. power station. (Source: J.M. Voith GmbH.)

Modern industrial gas turbine systems require precise fuel dosage for the lowest possible NO_x emissions. Each application requires the right actuator and valve combination to achieve exact, uniform fuel distribution to the burner. See Figs. F-33 and F-34.

The controller needs to be:

1. Inherently reliable (robust construction and low-pressure hydraulics)
2. Equipped with single-stage signal conversion which results in fast, accurate response times
3. Equipped with an oscillating magnet and minimized bearing forces to avoid static friction effects
4. Easy to install because the magnet and control electronics are all one unit
5. A control with availability of 99.9 percent

The actuator is only one of the components necessary for accurate flow control. Some controls OEMs cooperate with leading valve manufacturers to offer a total control system. All valves and actuators are factory mounted and aligned to reduce labor-intensive adjustments during commissioning.

Balanced valves have low force demands. Trip time of the complete valve assembly is less than 200 ms and the related increase in pressure is absorbed by the valve.

Valves are available with soft seals as well as with bellows for gaseous fuels.

For optimum performance and safety, electrical components face a burn test operated under “cold” conditions. (See Figs. F-35 and F-36.)

Electrohydraulic actuators utilize an integrated position regulator that provides a true position signal.

Other assembly features generally provided by actuator OEMs include:

1. Minimal interfaces
2. Valves designed to run without additional breakaway thrust, even after long, continuous operation
3. Flange mounting for easy assembly

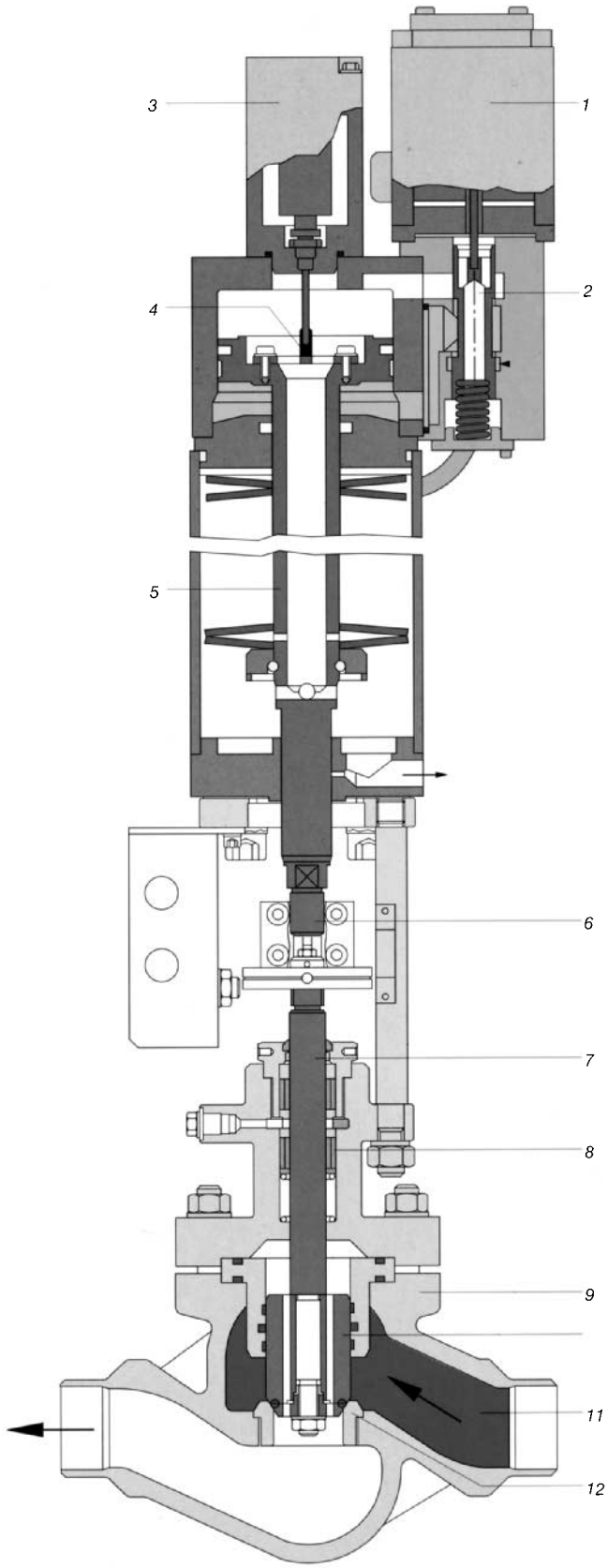


FIG. F-34 Section through an electrohydraulic actuator. Main components: 1, DC control magnet with integrated control electronics; 2, control piston; 3, position sensor; 4, clamp magnet; 5, drive piston; 6, piston rod; 7, stem; 8, stuffing box packing; 9, body; 10, trim; 11, fuel inlet; 12, seat. (Source: J.M. Voith GmbH.)

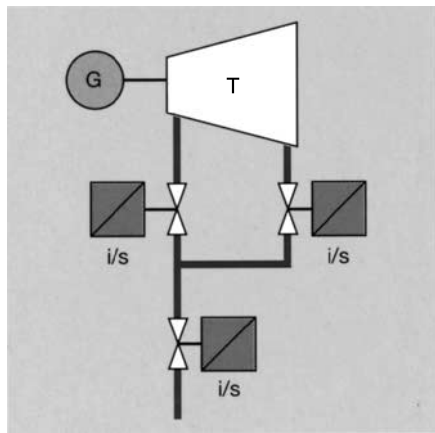


FIG. F-35 Schematic of gas turbine fuel system. ■⧖ = control valve assembly, G = generator, T = gas turbine. (Source: J.M. Voith, GmbH.)

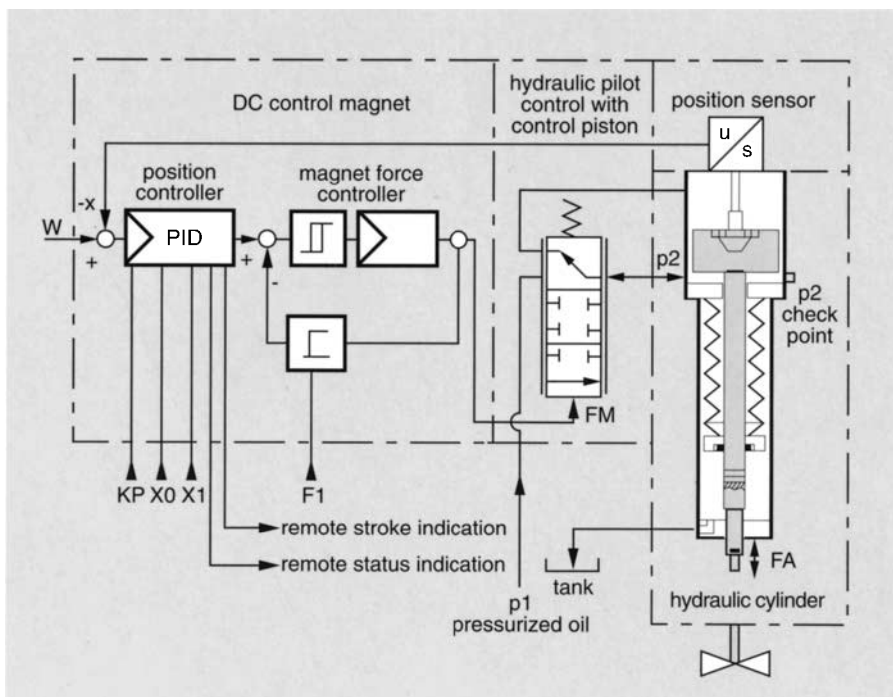


FIG. F-36 Functional schematic of control valve assembly. X_0 = pressure P_A at I = 0 or 4 mA; X_1 = pressure P_A at I = 20 mA; K_p = proportional amplification; F_M = magnetic signal/force; F_1 = feedback force/signal to controller; F_A = hydraulic cylinder force. (Source: J.M. Voith, GmbH.)

4. Fail-safe design
5. Explosion-proof design
6. Controlled emergency trip
7. Ease of commissioning and installation
8. Low maintenance
9. Compact design

10. Inherent long-life design
11. Failure indication signal
12. Insensitive to dirt due to encapsulated design
13. Control and performance insensitive to temperature
14. Precise repetition
15. Fast, hysteresis-free processing of signal
16. Friction free due to oscillating solenoid force
17. Compensation of different expansion factors with internal position regulator
18. Not affected by disturbance factors such as air gap, magnetic hysteresis, and voltage fluctuations
19. Open loop control, PID configuration

Typical range of valve actuators for gas turbines*

Control, emergency trip, and relief valves can be equipped with electrohydraulic actuators:

Medium	Nominal Widths	Nominal Pressure Stages
Natural gas	Up to 200 mm (8 in)	Up to 63 bar (914 psig)
Diesel oil	Up to 100 mm (4 in)	Up to 160 bar (2320 psig)
Water	Up to 100 mm (4 in)	Up to 160 bar (2320 psig)

Performance parameters:

- Maximum flow rate of gas valves: 450 m^3/h (1982 GPM)
- Maximum flow rate of oil/water valves: 160 m^3/h (705 GPM)
- Design of the valves to: ANSI or DIN specifications
- Internal fittings: Perforated or solid cone
- Flow curve characteristics: Linear, same percentage, open-closed, or specific
- Installation: Flanges or welded ends
- Valve tightness: Up to 0.001% of nominal K_{VS} value

Actuators*

Performance features include:

- Standard actuator stroke 50 mm (2 in) to maximum stroke 200 mm (8 in)
- Explosion-proof design is standard
- Operation with hydraulic or pneumatic auxiliary energy
- Low- and high-pressure designs up to 170 bar (2,500 psig) available
- Accuracy: ±0.1 mm absolute
- Tripping force at a stroke of 0 percent: ≥15.000 N

*Source: J.M. Voith GmbH, Germany. Adapted with permission.

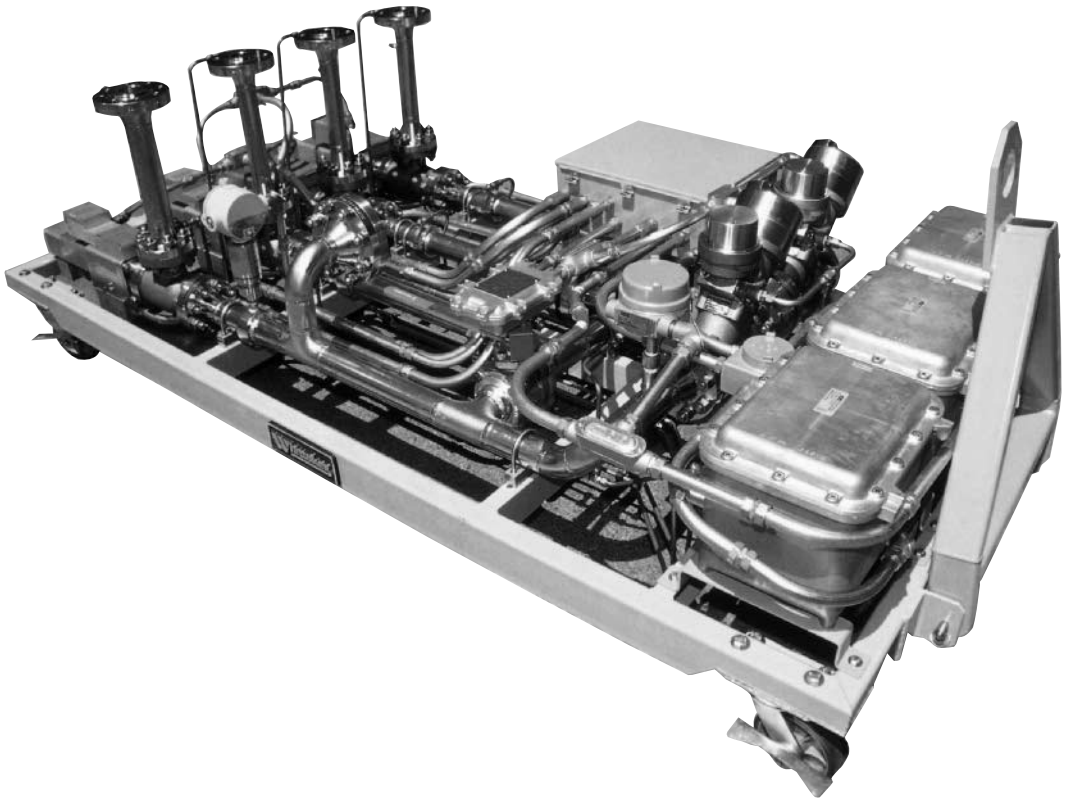


FIG. F-37 Universal all-electric DLE and SACn gas test cell fuel skid used with LM engines. (DLE and SACn are Whittaker model designations; LM refers to GE LM series gas turbines.) Approved to CSA, Ex standards. (Source: Whittaker Controls.)

- Hydraulic force: ≥41.000 N
- Opening time: ≥200 ms
- Tripping time: ≤150 ms
- Operation with negligible hysteresis: Resolution <20 μm
- Availability: >99.9 percent
- MTBF: $>5 \cdot 10^6$
- Service life: $>2 \cdot 10^6$

Fuel Skid

Fuel systems, whether part of a turbomachinery system or otherwise, generally consist of a skid to hold the main components, piping, control valves, and metering valves. The illustration of a fuel system skid in Fig. F-37 is typical of a skid that is supplied with a turbomachinery package. Typical valves in the system are indicated in Fig. F-38; they perform metering, isolation (shutoff), or staging functions. Figure F-39 illustrates a typical hydraulic-actuated modulating sleeve valve. Note the mechanical feedback on this valve type. Figure F-40 illustrates a servo motor actuated plug metering valve and its performance parameters.

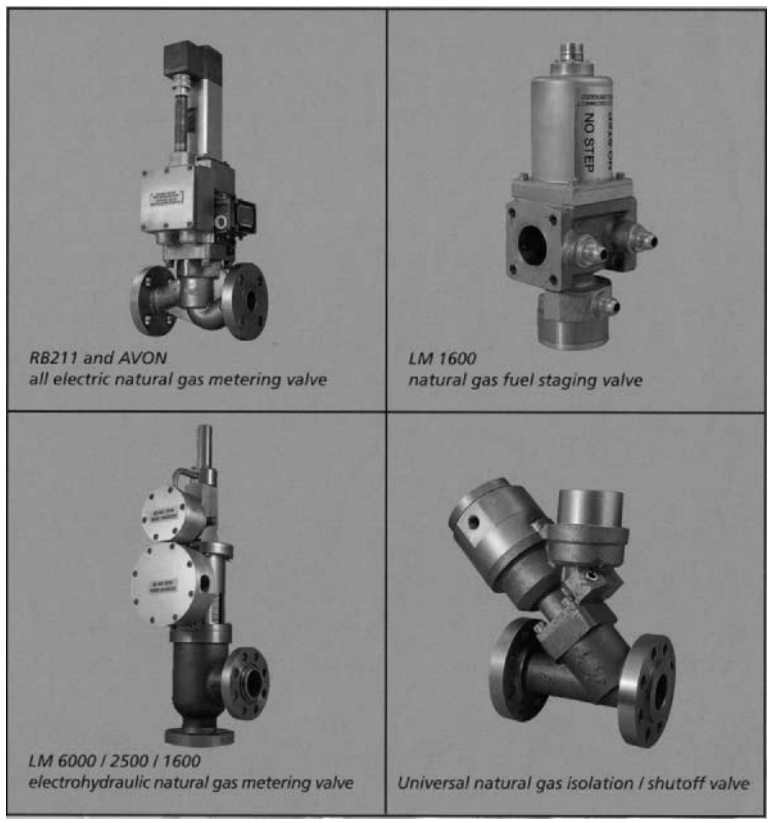


FIG. F-38 Some typical fuel system components. (Source: Whittaker Controls.)

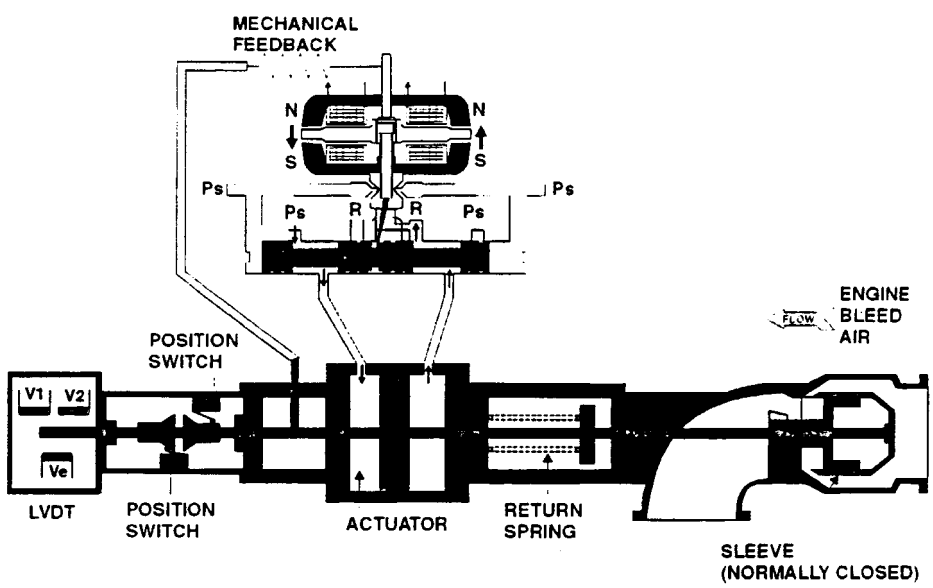
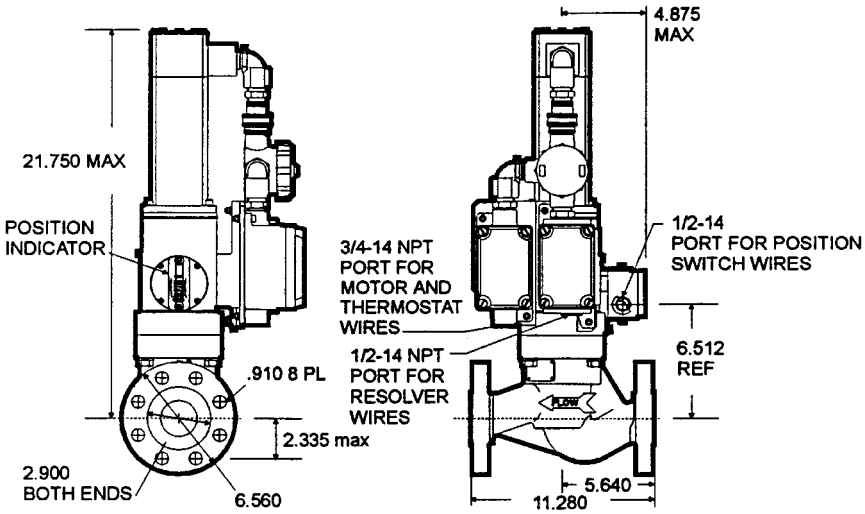


FIG. F-39 Typical 3-in hydraulic actuated modulating sleeve valve. (Source: Whittaker Controls.)



OPERATING FLUID:
Natural gas

APPLICATION:
Aeroderivative

TEMPERATURE:
Ambient: –65 °F to 350 °F
Fluid: 32 °F to 400 °F

ELECTRICAL:
Motor
Voltage: 170 VDC servo motor
Current: 0.3 amp max steady state
Resolver
Voltage: 4 VAC
Current: 25 to 60 ma max
Position switch
Voltage: 28 VDC
Current: 2.5 amp
Connector: Terminal block

INSTALLATION DATA:
Flanges: Per ANSI B16.5, 1.5 inch pipe, Class 600 both ends

WEIGHT:
80 lb max

PERFORMANCE:
Flow: 0 to 4.0 lb/sec, in direction shown effective area linearly propertional to valve stroke. Accuracy ±1% from 1.0 to 4.0 lb/sec
Pressure:
Operating: 100 to 600 psig
Proof: 1,440 psig
ΔP: 15 psid at 4 lb/sec, 500 psig at 60 °F
Hydrostatic: 2,160 psig at room temperature
Leakage:
Internal: ANSI class IV
Operating time:
Open to closed: 100 msec max
Closed to open: 100 msec max
Fail safe closed: 500 msec max

FIG. F-40 Typical 2-in servo motor actuated plug metering valve. (Source: Whittaker Controls.)

Fuel System Testing

Part of a typical test routine is outlined here. The end user concerned about fuel system malfunction needs to question the supplier, often a subvendor, about the results of some of these tests. Typically, the OEM adds its own nameplate to the system provided by the subvendor.

Typical fuel systems test specification* (for dual fuel machines or DLE systems)

A typical test specification would call for performance of the following tests:

- Fuel pressure cycle testing (DAT)
- Performance
- Endurance
- Burst
- Surge
- Fire resistance
- Leakage (internal)
- Leakage (external)

*Source: Whittaker Controls, USA. Adapted with permission.

Expected results vary based on line size. For instance, typical nominal test capabilities for a 3-in line size cover three ranges.

1. Static bench test for leakage (100 gpm at 1300 psig)
2. Self-contained dynamic test (low flow) (150 gpm at 600 psig)
3. Self-contained dynamic test (high flow) (300 gpm at 1300 psig and 1800 gpm at 600 psig)

Fuels, Alternative; Fuels, Gas Turbine*

The term *fuel* in process engineering generally means fossil fuel. The most common fossil fuels in use today are natural gas, oil, and coal. The latter two have many varying grades and sulfur contents. The emissions evolved from combustion of the fossil fuels are dealt with under other subject headings in this book, including Emissions and Turbines.

Common Gas Turbine Fuels

In broad terms, gas turbine fuels can be classified as gaseous or liquid fuels. The terms "gaseous" and "liquid" indicate the state of the fuel as it enters the gas turbine and not the state it is stored in at the site. The most commonly encountered gaseous fuels include:

- Natural gas
- LNG (liquefied natural gas)
- LPG (liquefied petroleum gas; typically a blend of propane and butane)
- Refinery gas
- Blast furnace gas
- Coke oven gas
- Coal gas
- Hydrogen

Suitable liquid fuels include:

- Distillate No. 2 (diesel fuel)
- Kerosene (K-1)
- Jet A
- Naphtha
- Condensate fuels
- Methanol (CH_3OH)
- Ethanol (C_2H_5OH)
- Crude oil
- Heavy fuel oil

---

*Source: Adapted from extracts from Narula, "Alternative Fuels for Gas Turbine Plants—An Engineering, Procurement, and Construction Contractor's Perspective," ASME paper 98-GT-122.

TABLE F-8 Gas Fuels for Gas Turbines

Fuel	Typical Lower Heating Value, MJ/Nm^3 (Btu/SCF)	Comments*
Natural gas	35.5 (900)	(1), (2), (4)
LNG	35.5 (900)	(2), (3), (6)
LPG (typical blend)	104.8 (2,700)	(2), (3), (5)
Propane	91.2 (2,300)	(2), (5)
Butane	118.5 (3,000)	(2), (5)
Refinery gas	Varies	(7)
Blast furnace gas	3.6 (90)	(7), (8), (17)
Coke oven gas	11.8 (300)	(7), (11), (14), (16)
Hydrogen	10.8 (270)	(9), (10), (16)

* Refer to comments nomenclature below Table F-9.

TABLE F-9 Liquid Fuels for Gas Turbines

Fuel	Typical Lower Heating Value, MJ/kg (Btu/lb)	Comments*
Distillate No. 2	42.7 (18,400)	(1), (2)
Kerosene (K-1)	43.0 (18,500)	(2)
Jet A	43.2 (18,600)	(2)
Naphtha	44.2 (19,000)	(2), (12), (13), (14)
Condensate fuel	45.2 (19,400)	(2), (12), (13), (14)
Methanol	19.9 (8,555)	(2), (12)
Ethanol	26.8 (11,522)	(2), (12)
Crude oil	41.2 (17,700)	(7), (13)
Heavy fuel oil	39.6 (17,000)	(7), (15)

* Comments nomenclature: (1) standard fuel; (2) clean burning; (3) needs vaporization; (4) low transportation and storage cost; (5) medium transportation and storage cost; (6) high transportation and storage cost; (7) needs scrubbing of contaminants; (8) needs enrichment with a higher heating value fuel; (9) high flame temperature results in high NO_x; (10) needs a separate startup fuel; (11) fuel needs heating to preclude condensate dropout; (12) poor lubricity; (13) low flash point; (14) high vapor pressure (highly volatile), needs a separate startup fuel; (15) needs preheating and treatment; (16) high (>40 percent) hydrogen fuel; (17) difficult fuel-to-air control because of large mass of air due to very low heating value.

Tables F-8 and F-9 list the lower heating values of these fuels and provide a brief commentary on each fuel. While many of these fuels have some unique applications for on-site use, they are not economically viable fuels for large-scale development projects. Therefore, this section only focuses on the four alternative fuels that are typically being considered on power projects worldwide: LNG, LPG, naphtha, and crude/heavy fuel oil.

Liquefied Natural Gas

Logistics

Natural gas is often found in remote locations far from the point of end use. Large reserves have been found in Siberia, Alaska, Sumatra, the Middle East, Australia, Indonesia, the Sahara Desert, and the North Sea. Where economically viable, the gas is transported by pipeline to the end user. Where the gas source and end user are separated by oceans and continents, the only viable alternative is to liquefy the natural gas and transport it via insulated LNG tankers.

Given the recent advancements in liquefaction technology, even the once-stranded gas fields are turning out to be economically viable sources of fuel supply. From the source the gas is piped to a coastal location where it is processed to remove impurities and inerts. After extracting heavy ends, the processed gas is finally refrigerated to make LNG and stored at atmospheric pressure and at a temperature of approximately –160°C (–256°F).

After the LNG is loaded into a tanker (generally 125,000/135,000 m^3) and transported, it is offloaded into storage tank(s) at the receiving terminal. The LNG is then pumped from storage to the desired pressure and vaporized for use in the gas turbine.

It is evident from the preceding description that LNG transportation, unloading, and storage is a major undertaking and entails hundreds of millions, if not billions, of dollars of capital investment. Because of these high fixed costs, this fuel is economically viable usually in large (2000 to 3000 MW) power plants. Existing receiving terminals and/or multipurpose use of LNG can make this fuel viable for smaller plants. The chief factors determining this fuel's viability include:

- Well head price of gas
- Liquefaction technology
- Size and tanker age
- Need for a breakwater at the receiving terminal
- Seismic classification of receiving terminal location
- Applicable safety criteria

Combustion considerations

Vaporized LNG is similar to, if not a little lighter and cleaner than, pipeline natural gas. There are no significant combustion concerns with this fuel since it burns just like natural gas.

BOP and energy integration considerations

Site selection, facility design, and energy integration are so critical that in many projects they can make a difference in the project being economically viable or not. The major considerations in this category include the following.

Plant site selection. A 2500 MW combined cycle facility with the associated harbor for the LNG receiving terminal and storage facility may require on the order of 60 hectares (150 acres) of geotechnically good quality land.

The harbor should be well protected from rough sea conditions to preclude the need for an expensive breakwater. It should have sufficient water depth (about 15 m) for the 125,000 m^3 (or larger) LNG carrier. The ship channel should be sufficiently deep for the LNG ship traffic and have a large ship turning basin (900 m diameter). Good seabed geotechnical conditions are also very important. Availability of fresh water for the process and fire protection systems is also very important, as this can preclude the need for, or reduce the size of, desalination facilities. Finally, the site should be compatible with the applicable environmental criteria.

Safety. Public health and safety and property protection are important issues that must be addressed at the initial stages of a project. Thus a hazards review study must be conducted. The three main aspects of this study are:

- Fire radiation analysis—Addresses ignition of the pool of LNG and levels of radiation at specified points. This is used to determine the minimum separation distances and the amount of fire water needed to cool the adjacent equipment.
- Gas dispersion analysis—Determines the dispersion of vaporized LNG for various climatic conditions. The extent of a vapor cloud is used in determining the minimum distance to sources of possible ignition.
- Detonation analysis—Addresses the resultant blast from unconfined or confined vapor explosions. This determines blast protection requirements and the safe distance for structures and equipment.

The results of the above-listed analyses are used to determine the exclusion zone—the area outside of which is considered safe for public access. The results are also used to determine in-plant separation distances. The hazards study must be conducted before finalizing the relative locations of storage tanks, liquefaction facilities, and other power plant facilities. From a capital cost viewpoint, these facilities should be kept as close as possible to each other; however, safety considerations mandate minimum safe distances anywhere between 200 and 800 m. Enlarging the exclusion zone by 1 to 2 km from any public facility such as a school, a hospital, or a highway may be necessary.

LNG cold utilization. As stated earlier, LNG is stored at a temperature of –160°C (–256°F). To use it as a fuel in a gas turbine, it must be vaporized in a heat exchanger by adding heat. The amount of energy transfer required is commonly known as LNG cold or LNG chill. Theoretically the amount of heat required has to be equal to the amount of energy required for vaporization. Further, we also know that the lower the temperature of the ambient air entering the gas turbine, the greater the amount of power it can produce. Thus, if we utilize the LNG cold effectively to cool the air entering the gas turbine, thereby increasing power plant output, we can improve the overall economics of the entire facility. Experience indicates that at a practical level we can boost the plant output by as much as 5 percent. Many other uses of LNG cold have been examined, but they are outside the scope of this book. It is, however, important to note that cost-effective use of LNG cold is heavily dependent upon the annual profile of the ambient air temperature and relative humidity at the site.

Energy and project integration. Historically, LNG trade across the Atlantic between Algeria and the U.S. or in the Pacific to Japan has been to keep the gas pipelines flowing predominantly for residential, industrial, and some limited thermal power plant use. The large-scale use of LNG for gas turbine–based power production is a fairly recent phenomenon and is predicted to accelerate in the near future. Because of the liberalization of the region's electricity market, the fastest demand for LNG is expected to be in the Asia-Pacific region in South Korea, Japan, Taiwan, India, and China. Thus, integration of the LNG receiving terminal and power plant has now become an important consideration. Having performed numerous studies and conceptual designs for a number of our domestic and international customers, the author is convinced that tens of millions of dollars can be saved by integrating the design, procurement, and construction of the two facilities. The factors contributing to these huge savings include:

- Optimization of LNG cold utilization
- Optimum layout of the entire facility based on safety and design considerations
- Integration of fuel unloading pier and water intake structure/discharge structure

- Integrated site development plan
- Common facilities for fire fighting, cooling water, electrical systems, and administration and warehouse facilities
- Common nonmanual construction staff
- Integrated schedule (this is important as LNG tank construction schedule is generally on the critical path)

Emissions. Emissions control when using LNG is analogous to natural gas, where low NO_x and CO emissions can be achieved. However, the actual allowable emissions may vary from country to country and from state to state within each country. While modern gas turbines are capable of meeting fairly low emissions, many states within the U.S. have very stringent emissions limits requiring the use of external devices such as selective catalytic reduction (SCR) and CO removal modules. Therefore the facility design must either be optimized up front to accommodate these devices or have the flexibility to incorporate such devices later depending on the site-specific emissions criteria.

Liquefied petroleum gas

Many of the design considerations discussed under LNG are equally applicable to the use of LPG as a fuel for gas turbine–based power plants. The discussion below is, therefore, limited to considerations unique to LPG.

- LPG is a by-product of natural gas treating processes or an incidental gas recovered during the oil extraction process. It generally comprises propane, butane, or a combination of both. As the spot market price for propane and butane varies with the seasonal demand, the receiving terminal and power plant facilities must be designed to handle 100 percent propane, 100 percent butane, or any combination of the two.
- Neither propane nor butane is as cryogenic as LNG. Propane and butane are stored at atmospheric pressure at temperatures of –42°C (–44°F) and 0°C (32°F), respectively. The LPG tankers are generally smaller (80,000 m^3) and less expensive than LNG tankers. The higher boiling temperatures of these gases (relative to LNG) and the owner's desire to use 100 percent propane, 100 percent butane, or a mixture of both has a significant impact on the size and design of the refrigeration system for storage tanks as well as on the design of the vaporization facility. It also impacts the design of gas turbine combustor and choice of startup fuel.
- Because these gases are heavier than air, proper attention must be paid to the plant design with respect to selection of the fire detection and protection system. To detect fuel leaks and prevent vapor access to drains, gas detectors must be located near floor level.
- As these fuels have a dew point that is higher than ambient temperature at the maximum expected gas turbine fuel system pressure, the fuel must be superheated, and fuel lines must be properly heat traced. Likewise, all the valves in fuel-forwarding stations need sufficient heat tracing. The design must avoid fuel collection by eliminating low points and by providing adequate venting to prevent two-phase flow into the gas turbine.
- The LPG cold utilization considerations are similar to those of LNG. The amount of cold energy, however, is smaller, especially with butane. Depending on the mix

of propane and butane, a study must be conducted to establish the economic payoff.

- It should be noted that because of limited LPG market, not all gas turbine suppliers are currently offering dry low NO_x combustors for their advanced gas turbines for this fuel. Use of moisture injection to control oxides of nitrogen can worsen thermal efficiency and increase BOP water treatment cost, especially at sites where desalination is necessary to make fresh water. Further, since it is a relatively new fuel for gas turbines, the emissions requirements with this fuel are not clearly defined by regulatory agencies. The actual emissions from the gas turbines when burning LPG are closer to those with liquid fuels than those with gaseous fuels.

Overall, because an LPG receiving terminal is not as expensive as an LNG terminal, a plant size as small as 500 to 700 MW is economically viable with this fuel. In smaller or island countries where total electric demand is modest, LPG may be a more viable fuel than LNG.

Naphtha

Logistics. Naphtha is a generic, loosely defined term that covers a wide variety of light distillates. It is processed from crude oil through distillation towers in petroleum refineries and is a primary ingredient in gasoline. It also has wide applications in the pharmaceutical, dry cleaning, painting and coating, rubber, and textile manufacturing industries. In developed countries, naphtha is usually cracked to produce ethylene, which commands a premium market price. In developing countries such as India, the market for cracked products is not as robust. Further, limited availability of natural gas has forced power plant operators to use naphtha as an alternative gas turbine fuel.

The naphtha classification includes common fuels such as gasoline, mineral spirits, and many petroleum solvents. In general, liquids classified as naphtha have a low flash point and high volatility and require special design and operating safety considerations. Naphthas are low viscosity liquids having poor lubricating qualities compared to No. 2 or heavier oils. They have been widely used as primary or back-up fuels for gas turbine plants in India. Other countries that have used naphtha as gas turbine fuel include Pakistan, Venezuela, Spain, Angola, France, Slovenia, Morocco, Italy, Saudi Arabia, Malaysia, Philippines, U.K., U.S., and the Virgin Islands.

Combustion considerations. In general, naphtha is a relatively clean fuel (cleaner than No. 2 oil) and can be used in gas turbines with minimum modifications to the gas turbine accessory equipment. As the fuel composition for naphtha varies significantly, its suitability for a given application must be confirmed with the gas turbine suppliers. The gas turbine suppliers generally require a fuel additive to increase the fuel lubricity. As with high-hydrogen gas fuels, naphtha, with its low flash point, requires special consideration in combustor design and an alternate startup fuel. Generally distillate No. 2 oil is used as the startup fuel.

BOP considerations

General. Hazardous operation design criteria must be considered for the fuel unloading, storage, treatment, and fuel forwarding/sendout systems. It is recommended that applicable piping and instrument diagrams, hazardous area classification drawings, and other design drawings be reviewed by the gas turbine

supplier to ensure that all criteria established by the gas turbine supplier are fully complied with.

Country-specific codes and standards must also be followed. For example, Indian codes and standards dealing with hazardous area classification are more stringent than U.S. standards regarding separation criteria and air movement. It should be noted, however, that some of these requirements are based on extreme hazards that may not exist for a particular application. The designer must exercise judgment to ensure that an appropriate level of safety is achieved and that all required approvals by local authorities are secured.

Fuel storage system. Bulk storage of naphtha presents greater risk than is experienced with less volatile liquids such as distillate oil. Special precautions associated with storage and handling of Class I flammable liquids must be followed. Vented atmospheric tanks cannot be used. Selection of the appropriate storage tankage depends on the properties (vapor pressure, etc.) of the naphtha under consideration. Options include low pressure tanks or pressure vessels. One way to cost-effectively minimize vapor formation and prevent accumulation of vapors is to use a floating roof design.

The use of Class I liquids such as naphtha may affect plant layout with regard to separation of storage tanks, structures, and other plant facilities.

Fuel handling system. The gas turbine supplier specifies the acceptable viscosity range for the fuel system, recommends the lubricity additive, and usually supplies the lubricity additive system. If site topography requires fuel storage tanks and sendout pumps to be located higher than the gas turbines, and/or if fuel delivery piping is fairly long, a transient analysis must be performed to determine the need for a surge chamber (or other measures such as controlling valve stroke times) to ensure that release of naphtha does not occur.

Emissions. In general, gas turbine emissions when burning naphtha are similar to emissions experienced with burning distillate oil. Moisture injection and use of external devices such as SCR and CO modules may be required depending on allowable emission limits. The water treatment and storage system design is often dictated by the demineralized water requirements for NO_x control.

Crude/heavy fuel oil

Logistics. Heavy fuel oil is basically what comes out the bottom of the distillation column after all the lighter oils have been removed. Lower grade fuel oil, particularly crude oil, has historically been used as a fuel for gas turbines in oil-producing countries with limited refinery capacity. The economic driver for using lower grade fuel oil is the fuel price difference between distillate oil and crude/heavy fuel oil. While spot market prices for these fuels can vary significantly, an approximate rule of thumb is that the price of crude oil is about 70 percent of the price of distillate oil, and the price of heavy fuel oil is approximately 60 percent of the distillate price. Thus, there is big cost incentive to use these fuels. Due consideration must, of course, be paid to the higher operating and maintenance costs associated with these fuels and the attendant lower plant availability. Also, the applicable environmental regulations may limit or preclude their use. It is interesting to note that the bulk of operating experience with crude oil is in the Middle East and Africa; with heavy fuel oil, it is in North and South America and Asia; and with blends of crude/distillate, it is in Europe. With the emergence of Venezuela as an oil giant, use of crude as a gas turbine fuel may increase in South America.

Combustion considerations. While fuel and machine design issues can best be addressed by gas turbine suppliers, the generic concerns with the heavy, ash-bearing fuels are:

- Corrosion of high-temperature materials due to presence of trace metal contaminants such as vanadium, sodium, and potassium
- Proper fuel atomization to ensure complete combustion and smoke-free stack over the entire operating range of the gas turbine
- Ash deposition and fouling of turbine and heat recovery steam generator components

Because of these concerns, use of crude/heavy fuel oil has been prohibited in high temperature (2400°F) "F Class" gas turbines by major gas turbine suppliers. Experience to date is with machines operating at firing temperatures of about 2000°F and below. For reliable operation of gas turbines with these fuels, it is important to follow the fuel procurement, fuel additive, and maintenance requirements imposed by the equipment suppliers.

BOP considerations

General. Crude oils typically have flash point temperatures lower than distillate oils and thus need provisions for explosion proofing. Heavy oils have high pour points that are characteristic of paraffin-based oils and dictate the need for higher storage temperatures.

Thus special consideration needs to be given to design criteria for the fuel unloading and storage, fuel treatment, fuel heating, and fuel forwarding systems.

Fuel storage system. In order to make the fuel pumpable, the fuel temperature should be at least 10°C (18°F) higher than its pour point. While crude oil is generally pumpable without preheating, this is not the case with heavy oils. Heavy oil storage tanks are typically maintained at approximately 30°C (86°F) with a bottom coil heater. In addition, an outlet heat exchanger heats the exiting oil above 60°C (140°F) as it leaves the storage tank. Further, fuel recirculation lines are provided to establish and maintain the desired operating temperatures.

Use of multiple storage tanks enables sufficient settling time for the water and other contaminants. Use of fixed roof–type frame design minimizes salt and other contaminants from the atmosphere from entering the fuel. The storage tank bottom is sloped to an area from which water and other settled material can be removed periodically to avoid buildup of microorganisms at the fuel/water interface. The treated fuel tank should have a floating suction in the fuel line feeding the gas turbine.

Fuel handling system. The transfer piping is heat traced to maintain the fuel at the elevated temperature to reduce pumping costs. Further, fuel recirculation lines are provided to establish and maintain the desired operating temperatures. The plant layout considerations and need for hydraulic transient analysis listed in the section on naphtha are also applicable here.

Fuel treatment system. Heavy fuels are frequently contaminated with trace metals and usually require both treatment to remove sodium and potassium, and injection of additives to inhibit corrosion. Crude oil requires washing to remove the sodium and potassium. Water washing of crudes in power plants is generally done by a centrifugal or electrostatic treatment system. Centrifuges are preferred because they remove contaminants more reliably. Electrostatic separators are more appealing, however, because they do not have high-speed rotating parts.

Emissions. From an emissions viewpoint, crude/heavy fuel oil is the least desirable fuel of the four fuels discussed in this paper. These fuels generally contain high levels of fuel-bound nitrogen, which increases NO_x emissions. Moisture injection may bring the NO_x emissions to acceptable levels. For sites with stringent NO_x emissions limits, however, this fuel may not be an acceptable alternative. Generally, because of the higher sulfur content in this fuel, SO_2 emissions are also higher than the other fuels.

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.

G

Gas Turbine Cleaning or Washing (*see* Turbines)

Gas Turbines (*see* Turbines)

Gearboxes (*see* Power Transmission)

Reference and Additional Reading

1. Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Gears (*see* Power Transmission)

Generators; Turbogenerators*

This section is written with reference to specific models made by the Alstom corporation. Most generator original equipment manufacturers (OEMs) use similiar standards.

Standard Design

The modular design of the turbogenerator permits the selection of a standard version with either an open or closed cooling system and static or brushless excitation systems.

The generators satisfy the requirement of IEC-34 and other relevant standards such as SEN, ANSI, NEMA, CSA, etc. This means that the generators can be used in countries in which these or comparable standards apply. Generators can be custom built to satisfy other standards.

Configuration

The turbogenerators are two-pole, air-cooled synchronous generators with cylindrical rotor and direct air-cooled rotor winding. They are intended for both basic and peak-load operations and designed to withstand high short-circuiting stresses. The configuration with journal bearings in the end shields permits delivery of the machines as a single unit which considerably simplifies installation and commissioning. See Fig. G-1.

Degrees of Protection and Methods of Cooling

The design of the generators provides for the following degrees of protection and methods of cooling in accordance with IEC 34-5 and 34-6, 1983.

Degrees of Protection	**Descriptions**
IP 21	Protection against contact with live or moving parts inside the machine and drip-proof.
IP 54	Dust-proof, splash-proof.

*Source: Alstom.

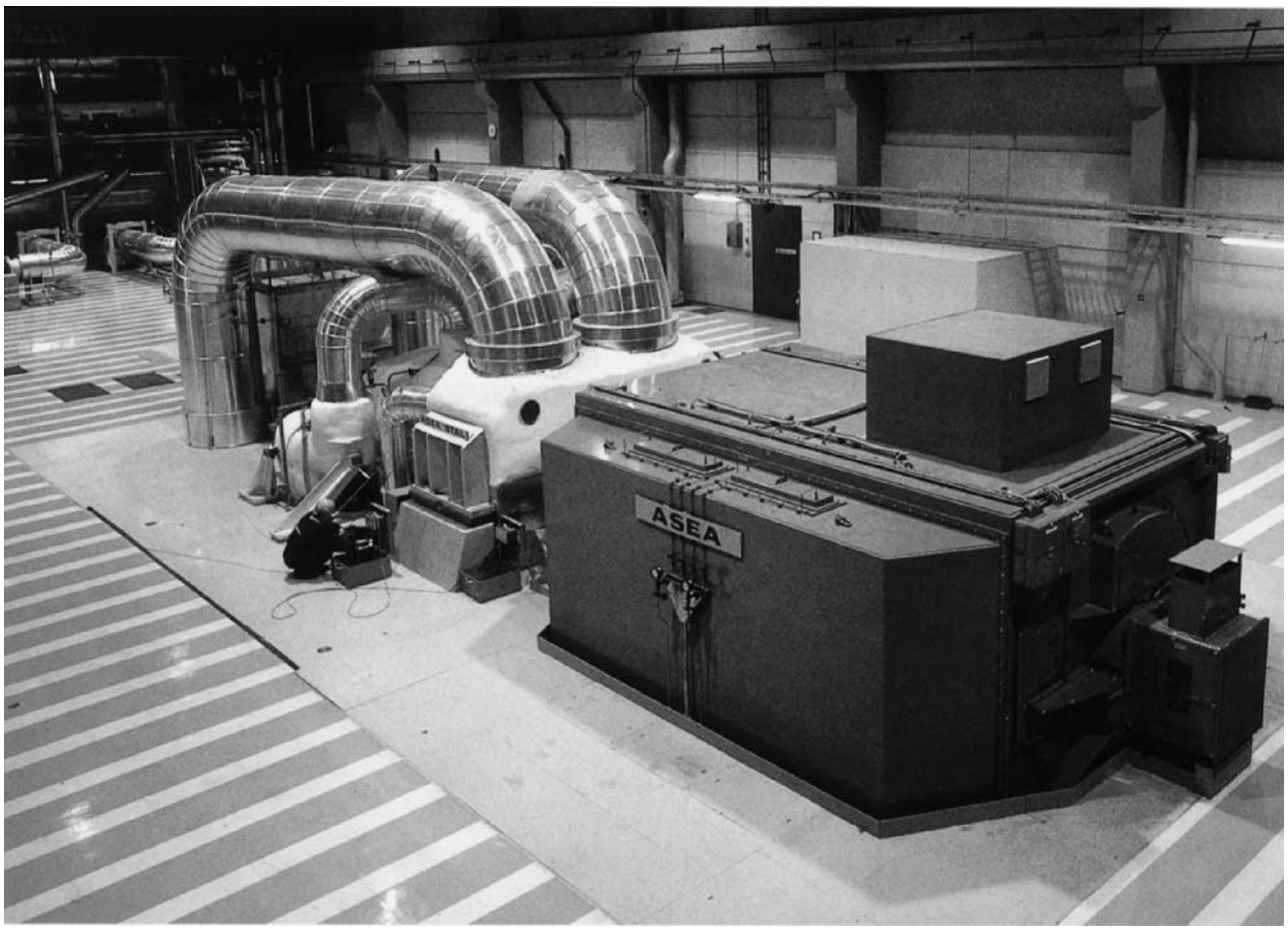


FIG. G-1 An installed generator package. (Source: Alstom.)

Methods of Cooling	Descriptions
IC 01	The cooling air is supplied to the machine through a screen-protected opening for different types of air filters on the long sides of the stator and is exhausted through a flanged opening. This is located at the middle of the generator housing and can be connected to a hood for the exhaust air. The method of cooling is then IC 21.
IC 71	The cooling air circulates in a closed circuit and is cooled in turn by a water-cooled heat exchanger.

These combinations of degrees of protection and methods of cooling are standard for this information source.

The different cooling methods require two fans mounted on the generator rotor.

The use of the IC 01 and IC 21 methods of cooling presupposes that the cooling air supplied is cleaned by thorough filtration. For this reason, IC 01 and IC 21

should be avoided when the cooling air available contains corrosive gases or large quantities of other pollution.

Arrangement Forms

The machines can be supplied in the following versions:

Arrangement Forms	Descriptions
IM 1005	Horizontal shaft, two bearings mounted in the bearing end shields, one free shaft extension end with coupling flange.
IM 1007	Horizontal shaft, two bearings mounted in the bearing end shields, two free shaft extensions with one coupling flange. This version permits powering of the generator from two directions. The stator is, in both cases, intended for foot mounting. If required, for example with an elevated turbine centerline, the above machines can be delivered with a base frame.

Excitation System

The generators can be delivered with one of two alternative excitation systems.

1. *Rotary brushless excitation system.* The winding of the generator rotor is supplied via a rectifier mounted on the shaft, from a directly connected 3-phase AC exciter. The system includes a pilot AC exciter.
2. *Static excitation system.* This system consists of a static thyristor rectifier unit supplied from an external power source. The excitation current is connected to the rotor windings via conventional sliprings. The static thyristor rectifier unit can be supplied on request.

Accessories

The following accessories can be supplied on request:

- Fire extinguishing system
- Base frame
- Lubricating oil system
- Sound insulating enclosure
- Current transformer
- Voltage regulation system

The following information and requirements should be provided by the end user to the OEM with any request for a bid.

- Relevant standards and recommendations
- Rated power
- Temperature of the cooling medium

- Power factor with rated power
- Rated voltage and voltage range
- Rated main power supply frequency
- Rated rotational speed and overspeed
- Rotation direction (as seen from the exciter end)
- Generator power in relation to the maximum and minimum ambient temperatures
- Degree of protection and method of cooling
- Arrangement form
- Excitation system
- Application
- Cooling air quality (for methods of cooling IC 01 and IC 21)
- Cooling water quality (for method of cooling IC 71)
- Special requirements, e.g., thrust bearings
- Extra testing and documentation

Technical Data on Typical Available Generators

- Power range, 20–200 MVA (40°C cooling air, temperature rise in accordance with class B)
- Insulation class F (155°C)
- Power factor 0.8 for 50 Hz, 0.85 for 60 Hz
- Standard voltage up to 80 MVA is 11 kV for 50 Hz and 13.8 kV for 60 Hz. Equipment for other voltages can be provided on request.

Design Description

Stator frame

The stator frame is of welded steel construction. See Fig. G-2.

The side plates are dimensioned to bear the weight of the complete generator during lifting. Openings are provided in the top or sides for cooling air supply and exhaust. Longitudinal foot plates are provided at the bottom of the long sides of the stator frame for fixing the stator to the foundation.

Stator core

The stator core is built up of segments stamped from thin silicon-steel sheets that give high permeability and small losses. See Fig. G-2.

The segments are varnished on both sides with a heat-resistant varnish to form an effective and permanent insulation between the sheets.

The segments are stacked to form a number of axial packages. Radial cooling air channels are formed between the packages by means of support plate segments provided with spacers.

The sheet segments are guided into place by axial guide bars. The stator core is pressed with the upper pressure ring installed to give the sheet pressure specified. The pressure ring is then locked.

The pressure rings and the superimposed pressure fingers are sprung into place during the pressing to maintain the necessary pressure in the stator core.

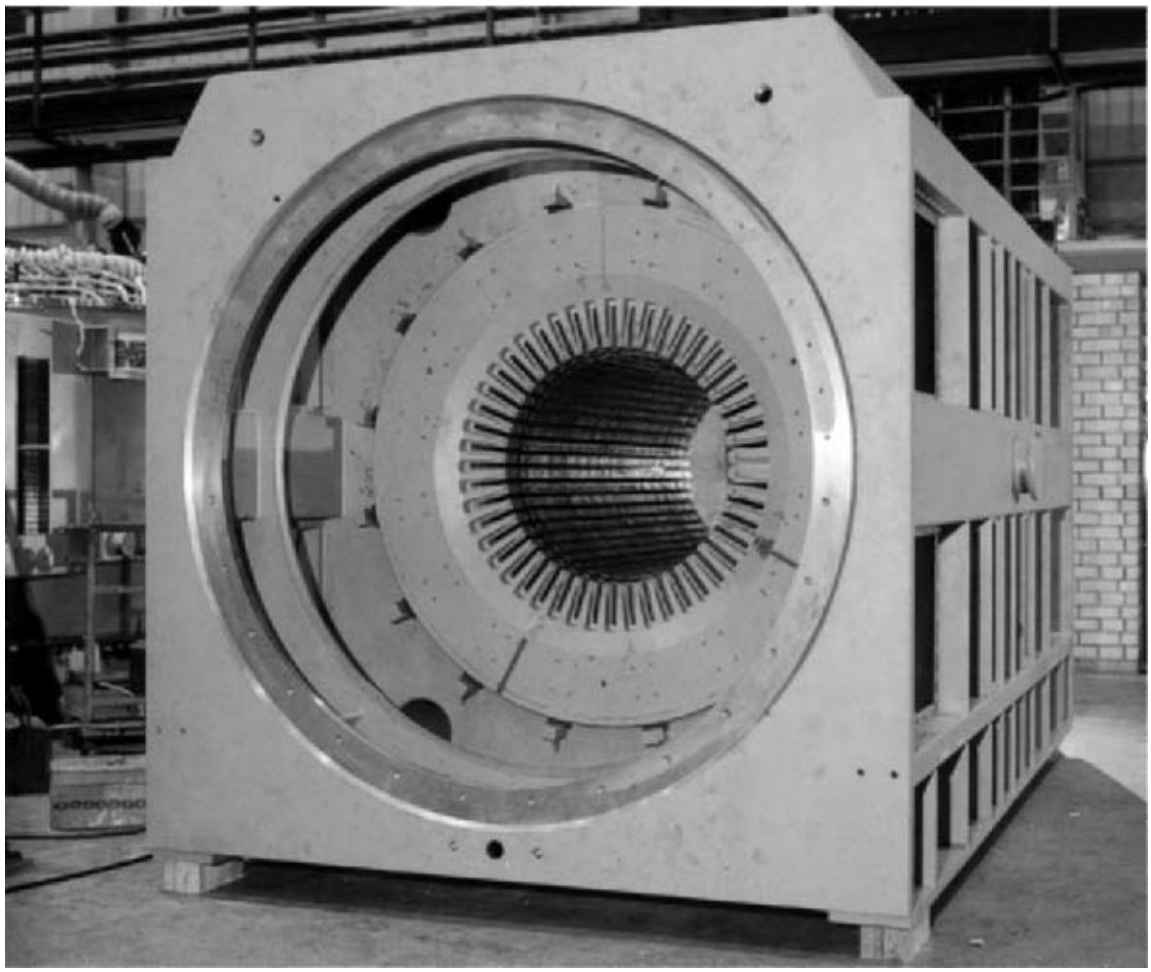


FIG. G-2 Generator stator. (Source: Alstom.)

Stator winding

The stator winding is a diamond winding installed in open slots with two coil sides per slot. The coils are manufactured in two halves that are brazed together.

The strands are transposed alternately through roebling in the slot section of the coil or by transposition, group by group, in the connections between the coils. The strands are insulated with impregnated glass fiber yarn.

The conductor insulation consists of MICAFOLD®.

The insulation of the coil sides is built up as continuous tape isolation, i.e., both the straight sections of the coil and the core ends are wound with tape. The insulation consists of epoxy-impregnated mica glass-tape. Both the preimpregnated MICAREX® system and the vacuum-pressure impregnated MICAPACT® system satisfy the requirements for temperature class F (155°C). The insulation systems used are described in more detail in separate brochures.

The straight sections of the coils are fixed in the winding slots in the stator core by means of contra-wedging. Spring elements are inserted in the slots to hold the coils in the winding slots, also after long service. These exert uniform pressure on the complete length of the straight section of the coil.

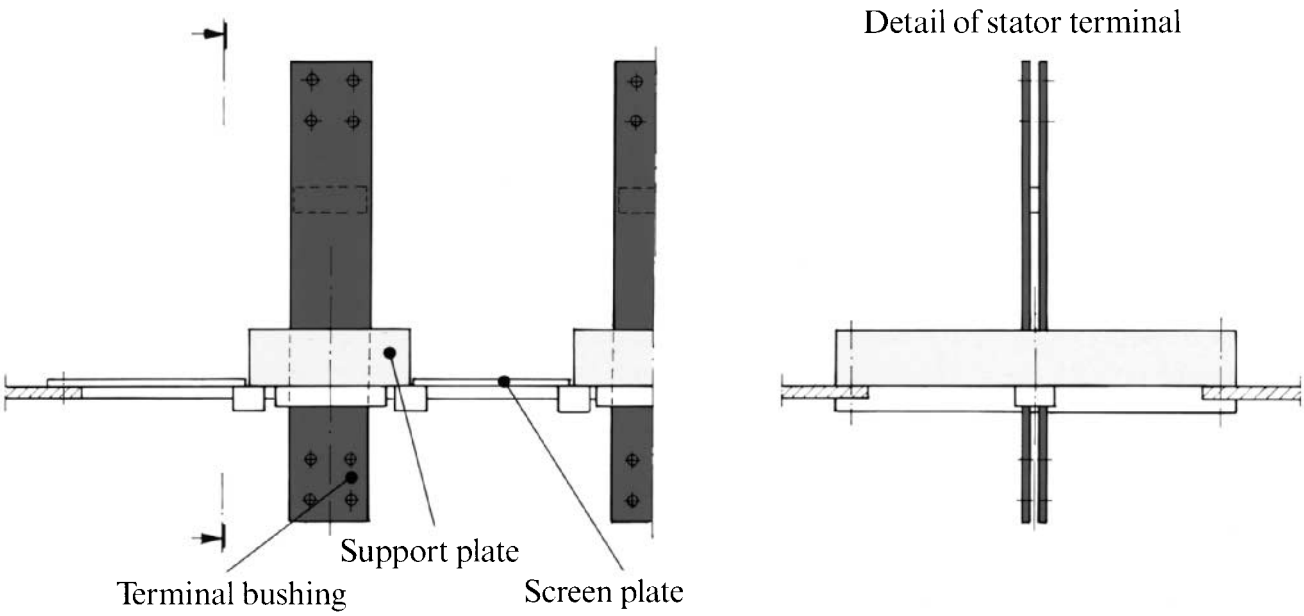


FIG. G-3 Stator terminal. (Source: Alstom.)

The ends of the stator coils are supported against the pressure rings in the stator core by radial support plates of insulating material. The coil ends are anchored to the support plates and braced against each other. The bracing system is dimensioned to withstand the stresses that can develop during normal operations and following a sudden short-circuiting of the generator.

Stator terminals

The connections to the busbars are located outside the generator casing (see Fig. G-3). The terminal bushings are connected to the terminal connections in the stator winding with clamps with generously dimensioned contact areas. The clamps are easily accessible for removing the stator terminals for, e.g., transport.

Fan covers

The covers are manufactured of heat-resistant, glass-fiber-reinforced polyester. Each cover is divided into removable segments to simplify inspection of the coil ends.

The fan covers are mounted against the stator frame over the coil end area and lead the incoming cooling air to the axial fans on the rotor shaft.

Bearing end shields

The bearing end shields consist of a very stiff lower half with bearing housing (see Fig. G-4). The upper half of the bearing housing is easily removed to facilitate inspection of the bearing. Holes for supply and drainage of oil are drilled in the lower half. The bearing housing is provided with connections for air extraction.

The upper half of the bearing end shield consists of a sound insulating screen plate divided into three parts to simplify removal for inspection.

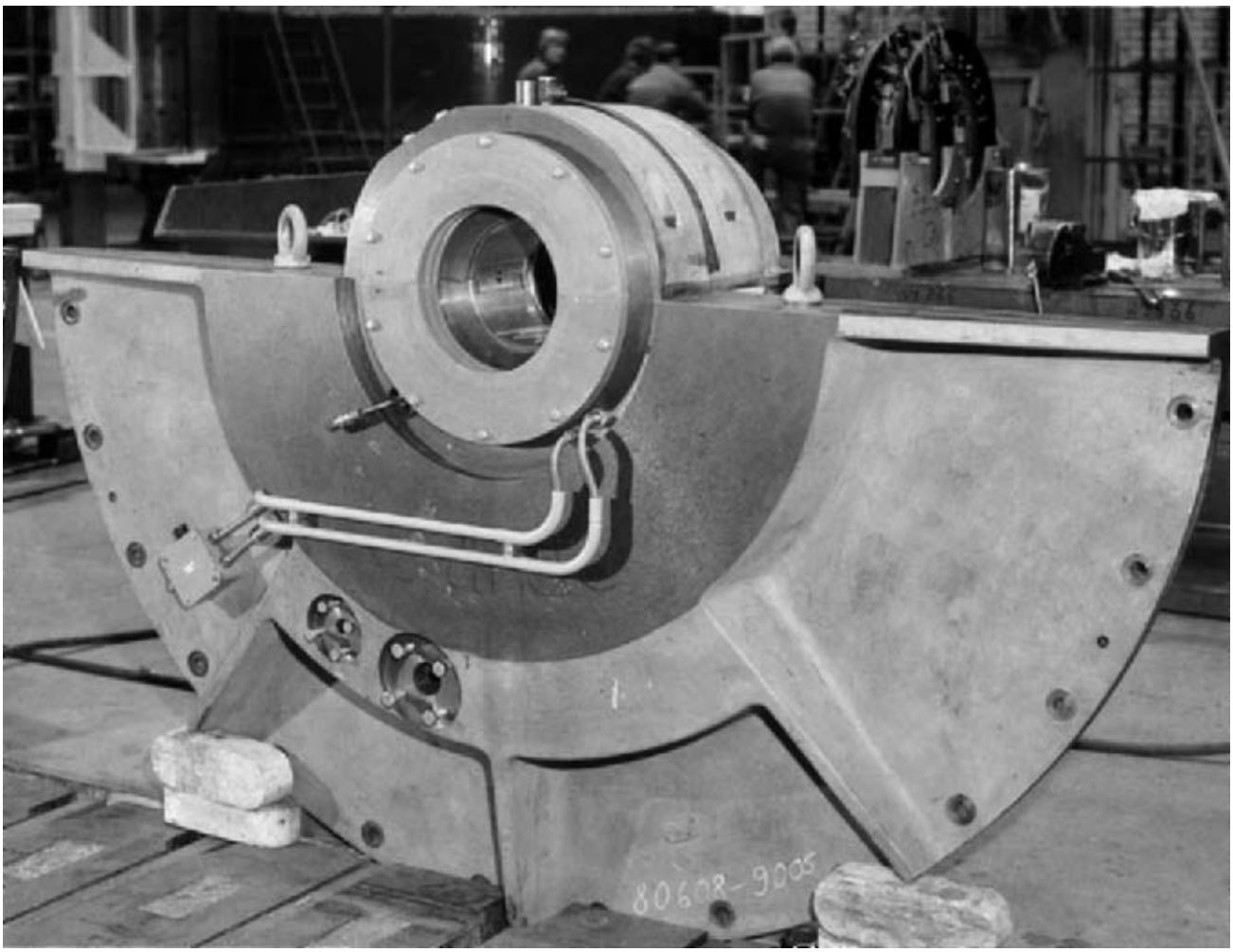


FIG. G-4 Bearing end shield. (Source: Alstom.)

Bearings

Each bearing housing contains a radial bearing (see Fig. G-5) for the generator rotor. The radial bearings are pressure-lubricated slide bearings with white-metal linings divided into staggered upper and lower segments. Lubricating oil is supplied to the bearing through drilled channels.

Where the bearings must take up axial forces, one is provided with axial thrust-bearing pads. The thrust bearing is double-acting, i.e., it takes up forces in both directions. One of the bearings is insulated from the bearing end shield to prevent the development of damaging bearing currents.

When the generator is to be driven from both ends, both bearings are insulated. The support bearing is always insulated. The shaft seals prevent oil leakage from the bearing housing and consist of a divided seal of oil-resistant insulation material. An air extraction system is connected to the bearing space to prevent oil leakage through the external seals. The internal shaft seals are provided with sealing caps and the intermediate space is connected to blocking air from the pressure side of the fan.

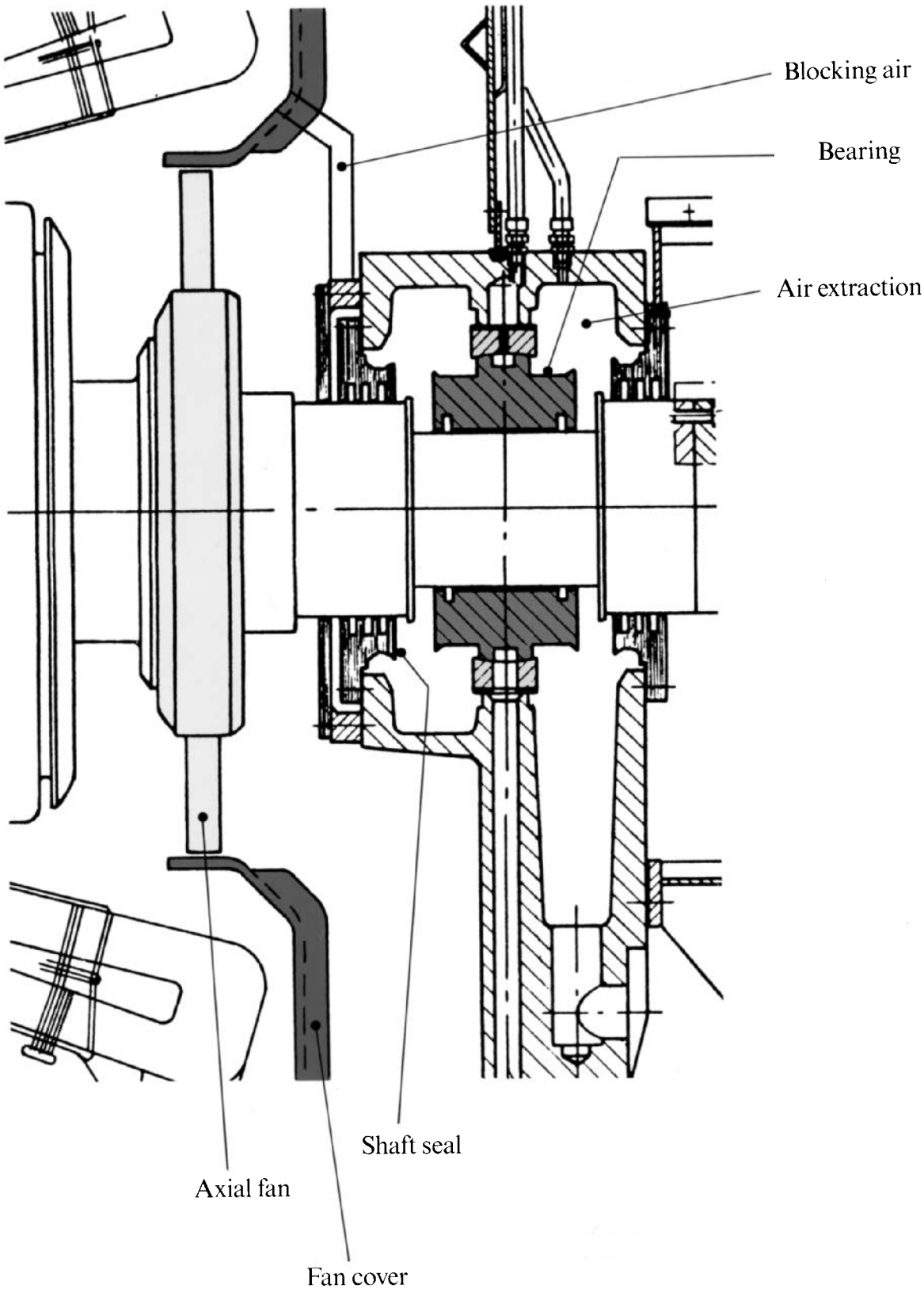


FIG. G-5 Fan bearing assembly. (Source: Alstom.)

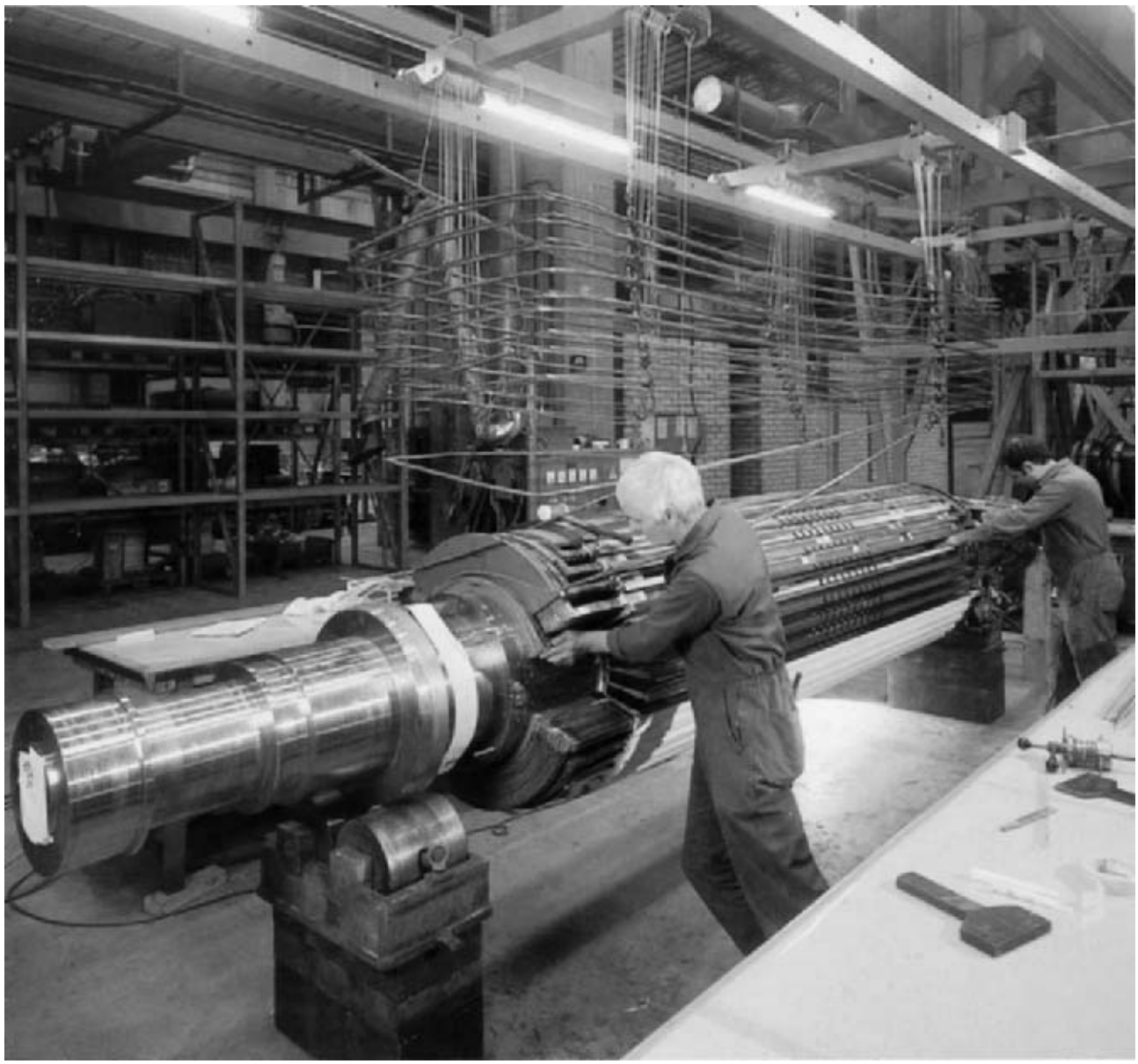


FIG. G-6 Assembling the rotor. (Source: Alstom.)

Rotor

The rotor body (see Fig. G-6) is manufactured from a cylindrical forging of high-alloy steel with suitable magnetic properties. A hole is drilled into the shaft extension for terminal conductors. The terminal conductors carry excitation current from the exciter to the rotor winding.

A number of axial winding slots for the excitation windings of the rotor are milled on both sides of the pole sections.

Rotor terminals

The rotor terminals (see Fig. G-7), are located in the shaft extension and consist of two copper conductors, enclosed in a tube of insulating material and insulated from

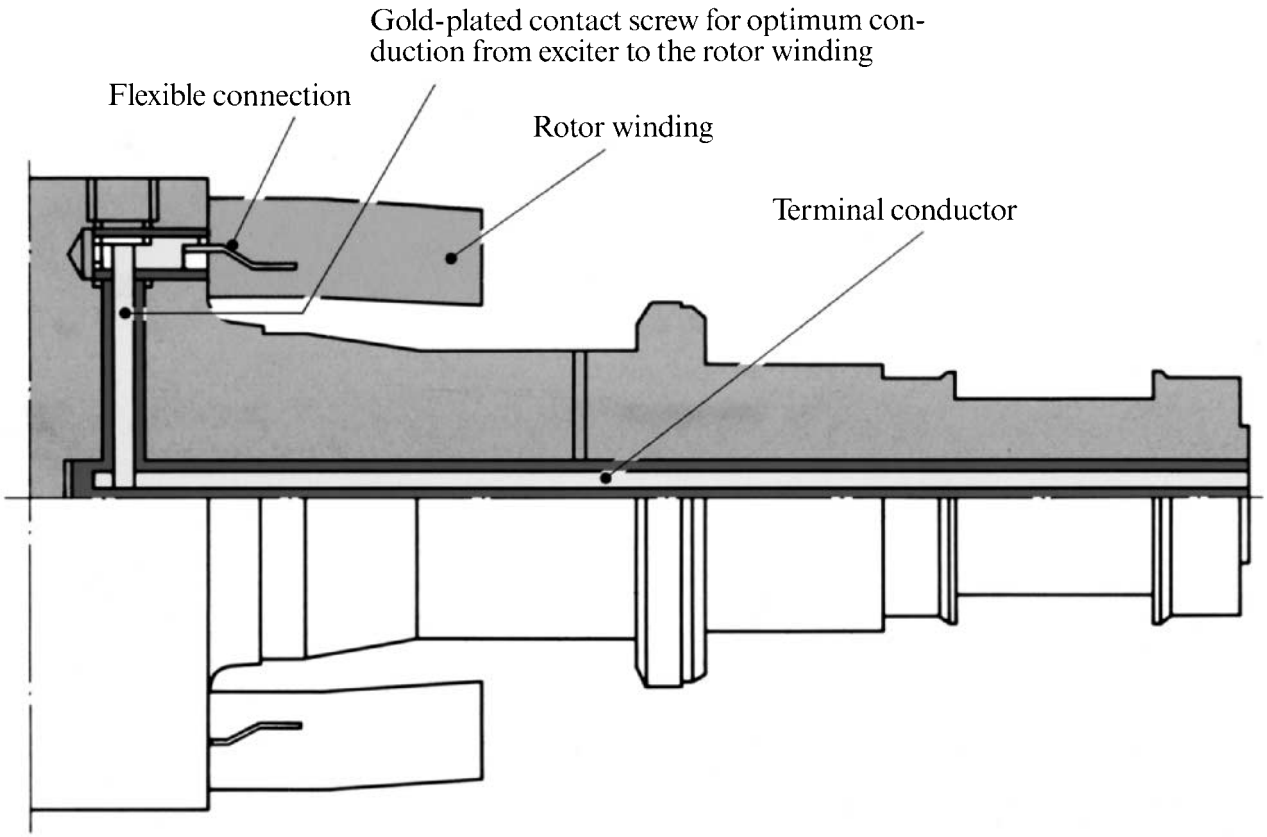


FIG. G-7 Rotor terminal. (Source: Alstom.)

each other. The connections from the terminal conductor and radially outward to the winding consist of contact screws. The design of the contact screws eliminates the risk of fatigue failure in the connection details caused by the differential movement of the rotor winding and the shaft.

All joints are provided with gold-plated spring-contact elements to ensure effective current conduction between the contact surfaces.

Rotor winding

The conductors in the rotor winding consist of silver-alloyed copper. Each conductor consists of two strands. The conductors are brazed to the rotor coils with silver solder.

The rotor coils are installed in the winding slots in the rotor body with conductor insulation of epoxy resin-impregnated glass fiber fabric between each conductor layer. The spaces at the side of the rotor coils function as ventilation channels.

The straight sections of the winding are supported tangentially with a number of bracing pins mounted in holes in the conductors and the coil insulation. The straight sections are fixed radially by means of pressure bars filled with synthetic resin under high pressure. The windings are thereby fixed radially and all play in the winding slots is eliminated.

The rotor coil ends are supported with blocks of glass fiber fabric laminate.

Impregnated glass fiber fabric is used as insulation between the retaining rings

and rotor coil ends of the winding. The coil end insulation is provided with cooling channels after hardening.

The insulation system satisfies the requirements for temperature class F (155°C).

Rotor retaining and support rings

Rotor retaining rings of high alloy nonmagnetic steel are shrunk on the rotor body to provide radial support of the rotor coil ends against centrifugal forces during rotation.

The retaining ring material is not affected by stress corrosion. To ensure that the retaining rings are sufficiently stiff to remain circular, a support ring is shrunk in the outer end of the retaining ring, free from the rotor shaft.

The retaining rings and the other parts of the bracing system are dimensioned to withstand the stresses that develop with short-circuiting and overspeeding.

Axial fan

A number of aluminum fan blades are mounted on a fan hub, shrunk on each shaft extension on the rotor. The fan blades are easily removed and replaced when the rotor is to be installed.

Balancing and overspeed running

The rotor is balanced when the rotor winding, insulation, and retaining rings are in place. It is then heated to a high temperature and test-run at overspeed. This results in the winding adopting its final form before the rotor is finally balanced to satisfy the relevant requirements. The normal overspeed is 120 percent of the rated speed. Certain of the balancing planes remain accessible when the rotor is installed in the stator.

Rotor cooling

The rotor winding (see Fig. G-8) is directly air-cooled by ventilation channels in direct contact with the winding slots. Cooling air is drawn in between the support ring and the shaft extension by the centrifugal fan effect of the rotor.

The air cooling the coil ends passes out radially through holes in the coil end insulation to the retaining ring and axially in channels under the retaining ring through holes in the support ring.

The cooling air in the winding slots is drawn in under the coil ends and into the axial cooling air channels between the sides of the slots and the conductor package. The rotor slot wedges and the beams between the winding slots are provided with a number of exhaust openings, connected in parallel, for the cooling air.

Cooling Systems

The generator is provided with two axial fans. The main task of the fans is to ventilate the stator (see Fig. G-9); the rotor itself functions as a centrifugal fan. The generator can be provided with an open or closed cooling system. The air circulation in the different cooling systems is shown in Figs. G-9 through G-11.

In closed cooling systems (see Fig. G-10), the cooler housings are mounted, with the air/water coolers, on the long sides of the generator. The cooler housings are welded steel constructions that lead cooling air from the coolers to the axial fans.

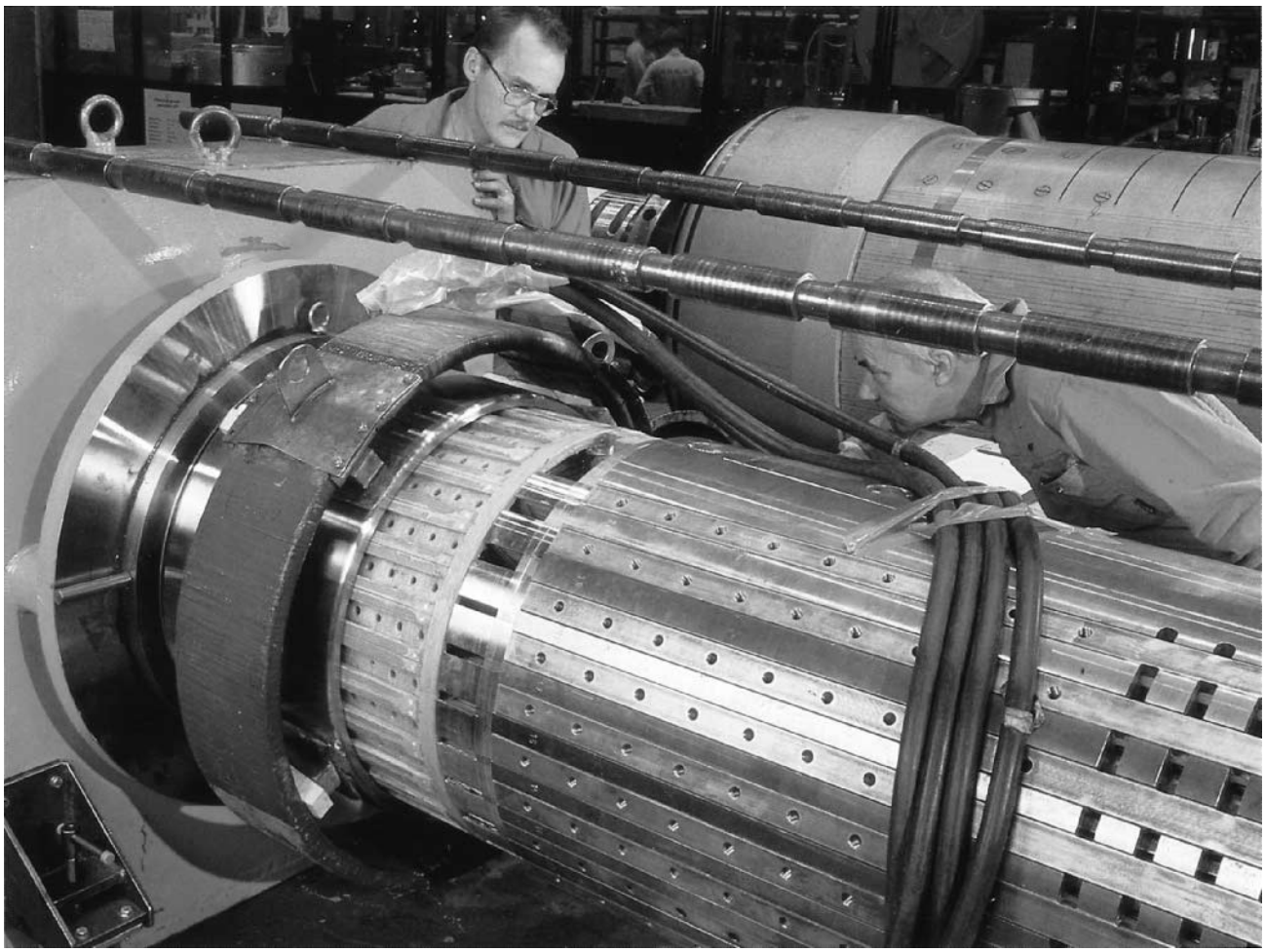


FIG. G-8 Rotor winding. (Source: Alstom.)

The air/water coolers are of the lamellar tube type. The materials chosen for the tubes and water chamber are dependent on the quality of the cooling water available.

Compensation air is drawn in through a filter to replace air leakage from the machine.

With an open cooling system (see Fig. G-11), the incoming air is to be filtered.

The choice of filter is determined by the site conditions. Recommendations for filter selection are based on the information regarding the environment provided with the request to tender.

Figure G-12 illustrates a generator assembly that has not been installed.

Brushless Excitation

Main exciter

The main exciter (see Fig. G-13) is a multipole synchronous generator with salient poles in the stator.

The stator and rotor winding insulation satisfies the requirements for temperature class F (155°C). The windings are impregnated for protection against damp and vibration.

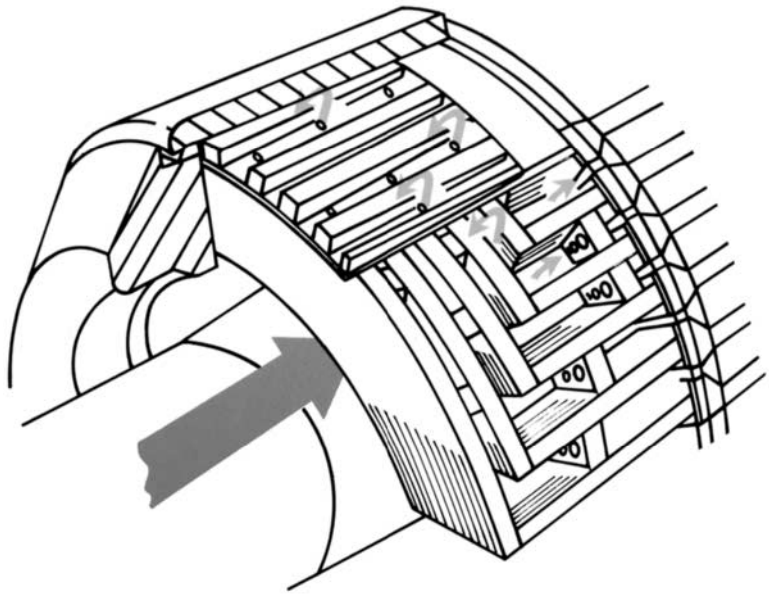


FIG. G-9 Cooling air direction. (Source: Alstom.)

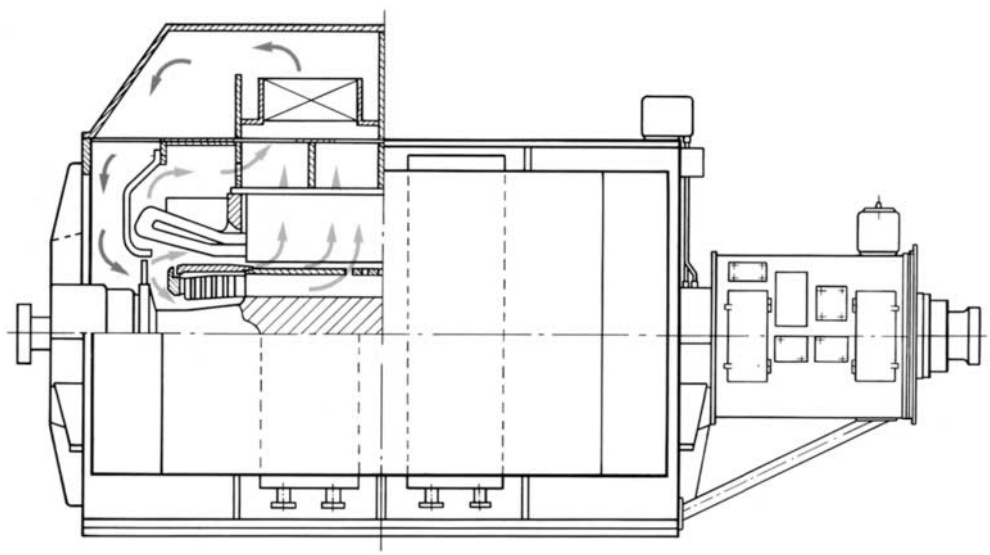


FIG. G-10 Closed cooling system. (Source: Alstom.)

Rotating rectifier

The purpose of the rectifier is to rectify the AC current from the main exciter and provide the rotor winding of the turbogenerator with DC current via connectors in the center of the shaft.

The electrical equipment in the rectifier consists of silicon diodes and RC protection.

The rectifier is built of two steel rings that are shrunk on the shaft of the exciter rotor with intermediate mica insulation. The steel rings are connected to the terminal conductors in the center of the shaft.

All contact surfaces are specially processed to guarantee a high degree of rectifier reliability and stability during operations.

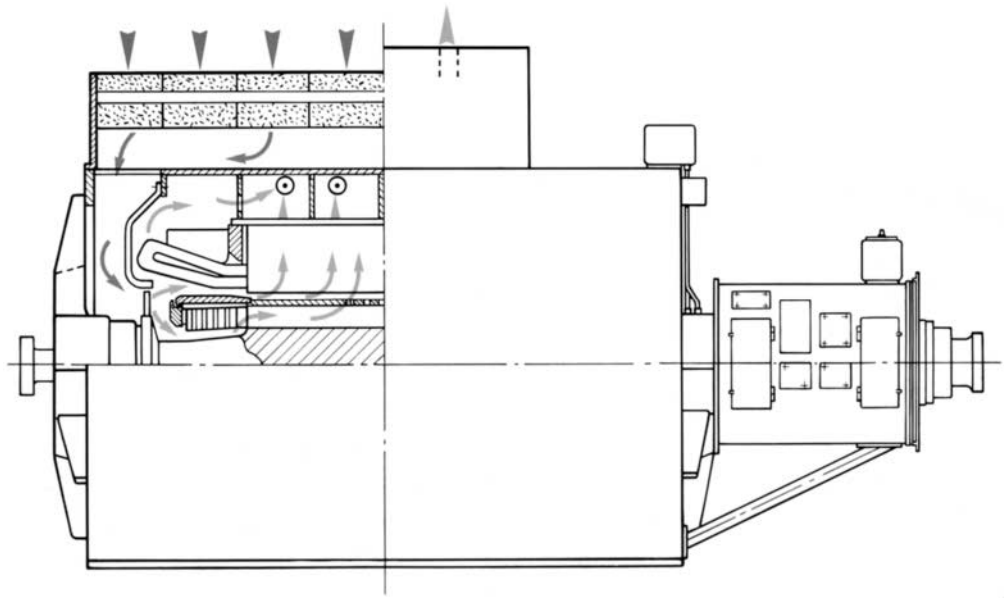


FIG. G-11 Open cooling system. (Source: Alstom.)

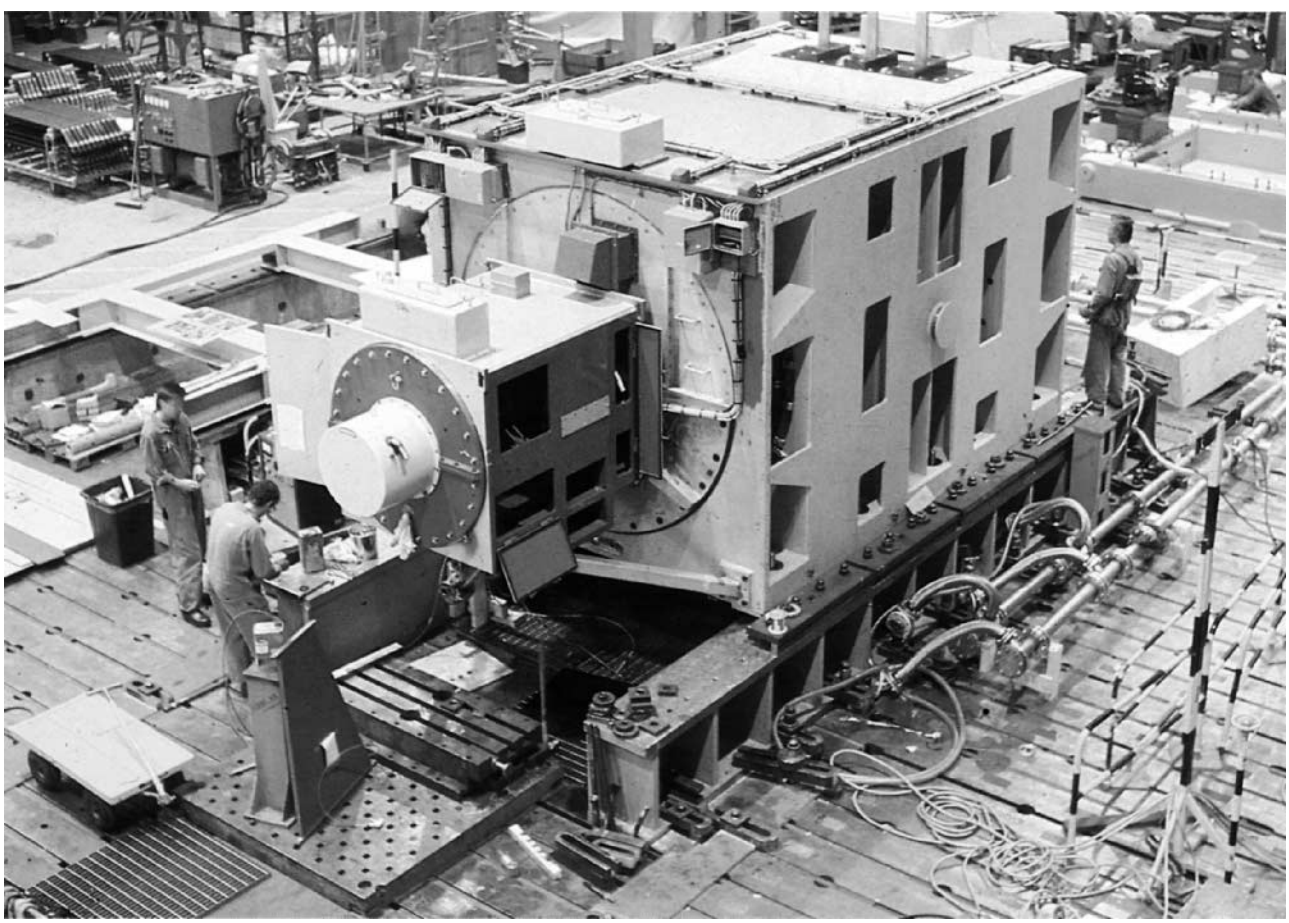


FIG. G-12 Generator assembly. (Source: Alstom.)

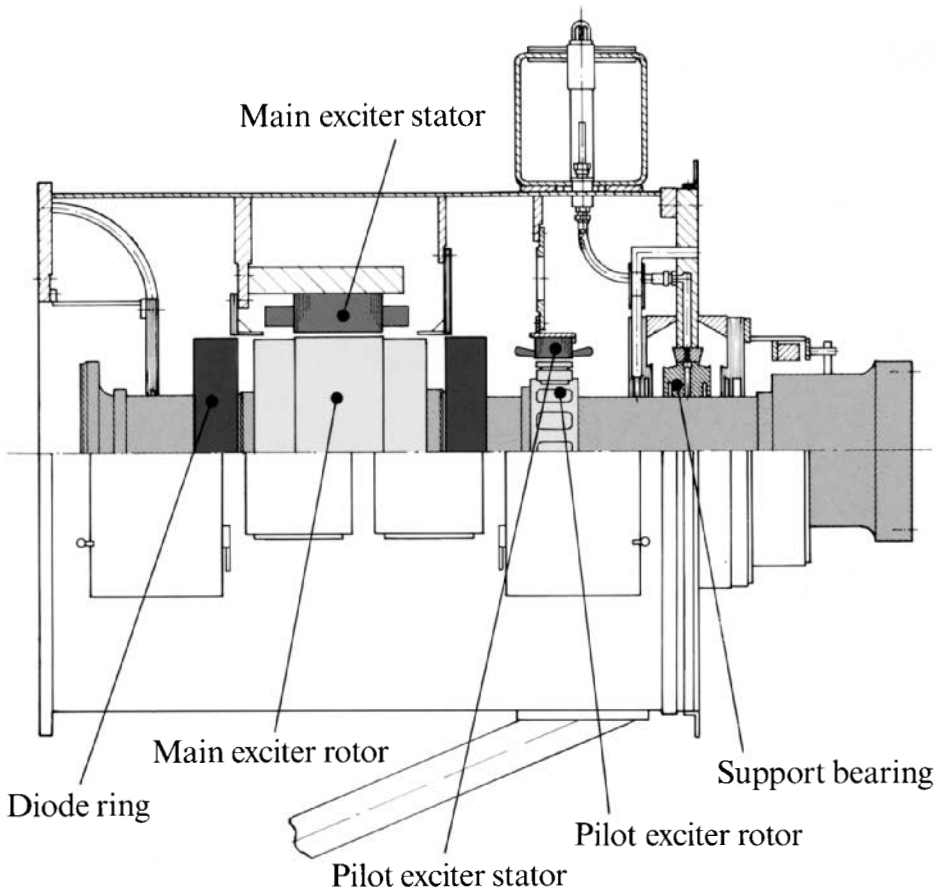


FIG. G-13 Main exciter. (Source: Alstom.)

Pilot exciter

The pilot exciter is a synchronous generator with permanent magnets on the rotor.

The rotor magnets are enclosed in a short-circuited aluminum ring that prevents demagnetization of the poles because of short-circuiting in the stator winding.

The stator winding insulation satisfies the requirements for temperature class F (155°C).

Exciter housing

Openings are provided in the side walls of the housing for cooling air intake and exhaust and for service activities.

Support bearing

The support bearing consists of shield, bearing insulation, bearing, and shaft seals.

The bearing, which consists of a bearing body with white metal lining, is insulated from the shield. The shaft seals against oil leakage from the bearing housing consist of a split seal of oil-resistant insulation material. An air extractor is connected to the bearing space to prevent oil leakage through the external seals. Internal shaft seals are provided with sealing caps and the intermediate space is connected to the blocking air from the pressure side of the fan.

Cooling

The exciter can be provided with either an open or a closed cooling system.

With an open cooling system, a housing with filter cassettes is mounted on one side of the exciter housing for the incoming cooling air. The cooling air exit is directed downward under the exciter housing.

With a closed cooling system, the filter housing is replaced with supply and exhaust air channels connected to the cooler housing of the generator.

Surface Treatment

In its standard version, the generator is painted with a lacquer of two-component type based on ethoxylized chlorine polymer. The generator is primed inside and outside and then finished externally in a neutral blue color. The paint is resistant to corrosive, tropical, and other aggressive atmospheres.

Static Excitation

When the rotor winding of the generator is supplied with current from a static rectifier unit, the rotor is provided with a slipring shaft and supplied via brush gear.

Slipring shaft

The slipring shaft (see Figs. G-14 and G-15) consists of shaft extension, insulations, sliprings, contact screws, and terminal conductors.

The shaft extension consists of a steel forging with flange for connection to the shaft of the generator rotor. The center of the shaft extension is drilled for the terminal conductors.

The sliprings are manufactured of steel and have generously dimensioned contact surfaces for the carbon brushes. The spiral-machined contact surface is carefully ground and polished. This prevents current concentrations and reduces brush and slipring wear. The sliprings are shrunk on the shaft extension on a cylinder of insulating material. The radial connections from the sliprings to the terminal conductors consist of insulated contact screws through holes in the shaft extension.

The terminal conductors in the slipring shaft and the rotor shaft are connected with contact screws.

Brush gear

The brush gear consists of a frame, insulation, brackets with brush holder, and carbon brushes.

The brackets are insulated from the frame with rings of insulation material and are connected to a supply ring to which cables for the excitation current are connected. Each bracket is provided with a brush holder pocket for connection of a handle to hold two carbon brushes.

The carbon brushes in the handle parts are mounted in holders of coil-spring type that give a constant brush pressure during the service life of the brush. The handle parts are insulated and the brush holders can be removed from the brush holder pockets by hand when the brushes are to be replaced. Brush replacement is thus possible during operations.

Slipring housing

Openings are formed in the side walls of the housing for service. These are covered with hatches provided with air filters. The opening in the end wall of the housing,

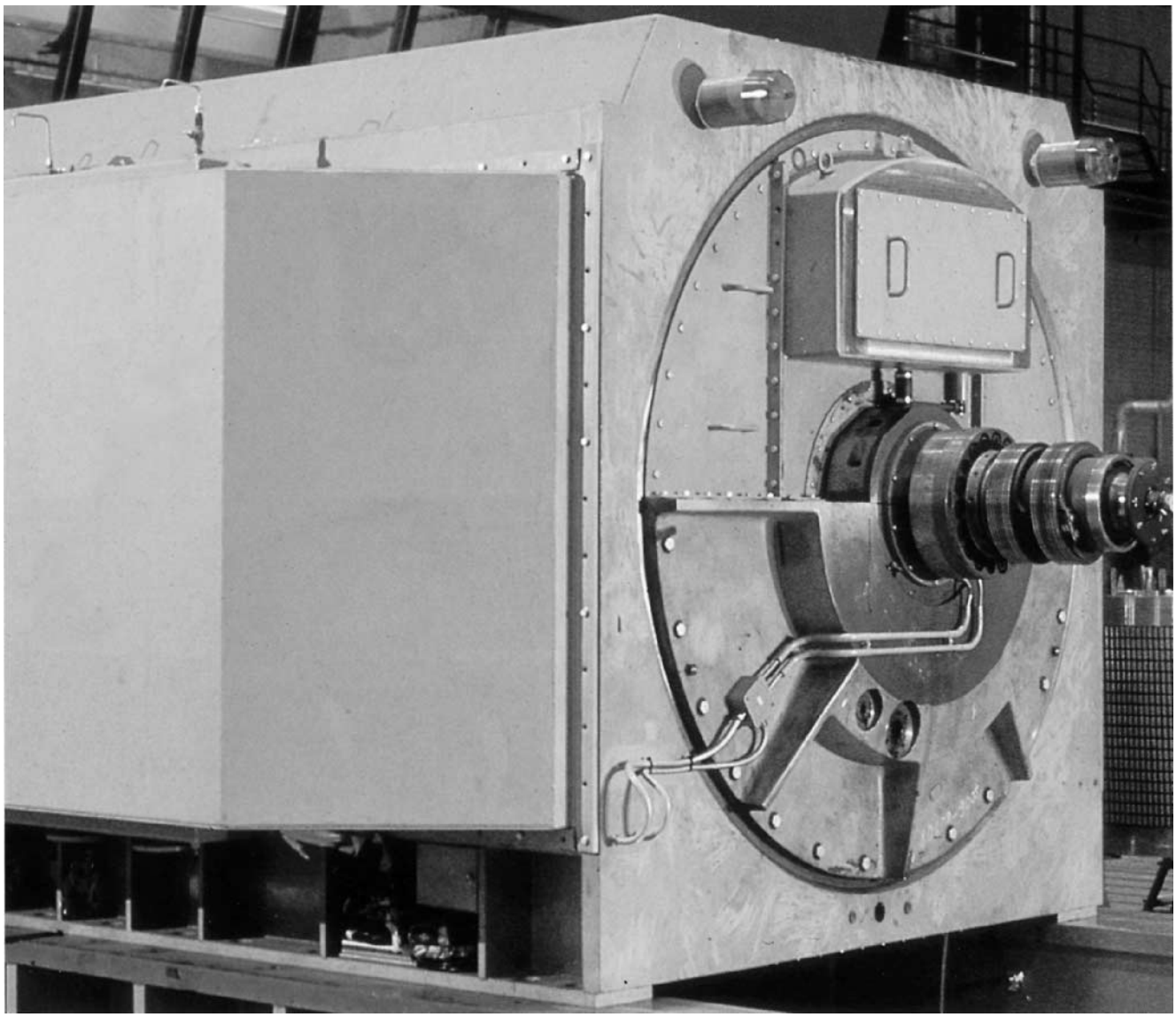


FIG. G-14 Slipring shaft end. (Source: Alstom.)

towards the generator, through which the slipring shaft passes, is provided with a seal.

Inspection and Testing

General basic inspection and testing points performed during the fabrication of the generators are included in a check plan. Each manufacturing operation is subject to extensive checking.

Final tests

Tests in accordance with Table G-1 constitute part of the normal testing before delivery. A type test is performed on the first machine in a manufactured series and

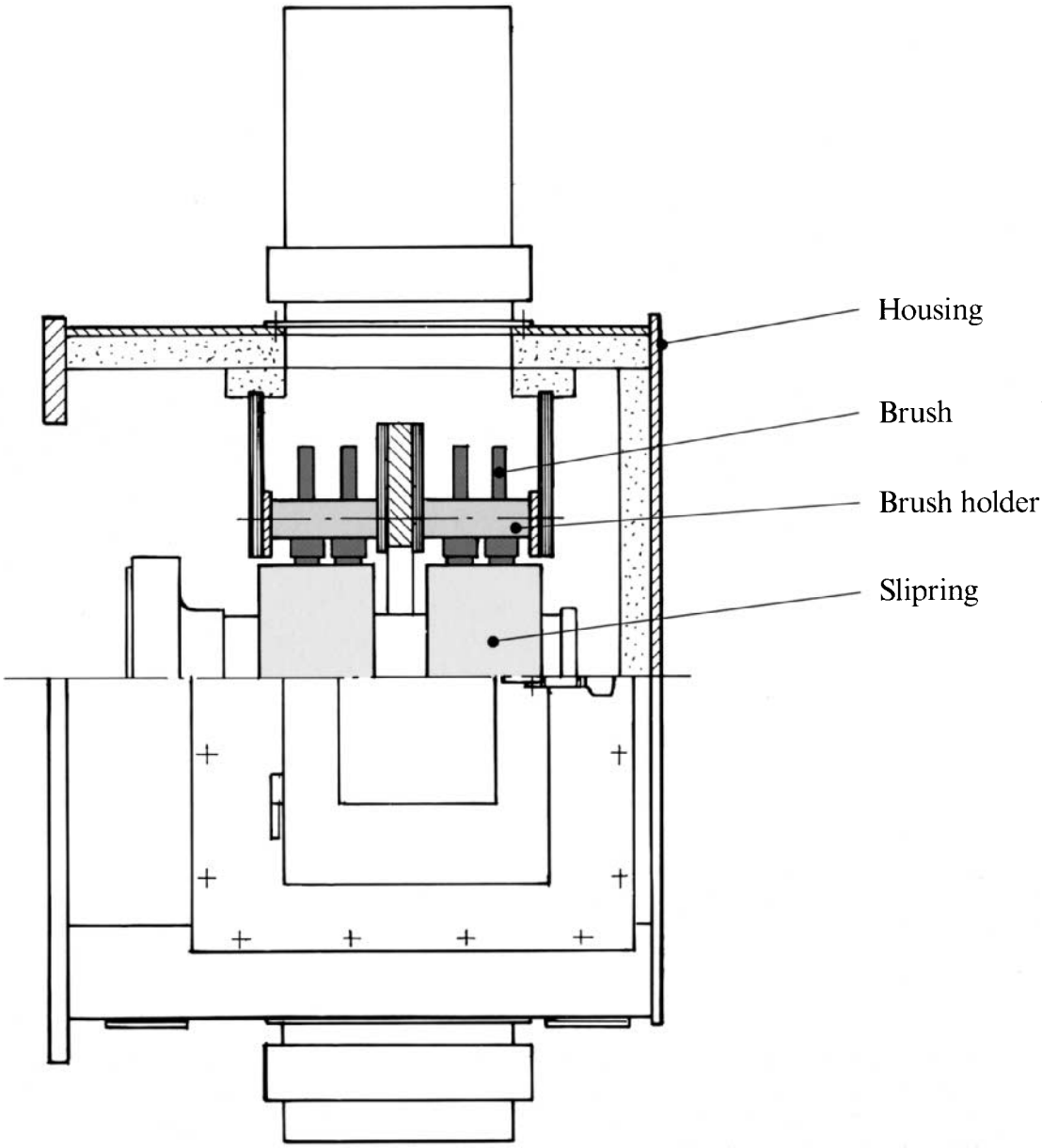


FIG. G-15 Slipring assembly detail. (Source: Alstom.)

its result is used as a reference for the subsequent machines of the same type. A more extensive test can be offered separately.

Control and Protection

Temperature monitoring

A number of platinum wire resistance elements installed in different parts of the machine are used for continuous monitoring of the temperature of the parts. The connection cables of the elements are routed to junction boxes on the outside of the stator housing. The number and location of the elements are shown in the following list:

TABLE G-1 Normal Testing

Test	Type Test	Routine Test
Overspeed test	×	×
Measurement of winding resistances	×	×
Generator characteristics	×	
Measurement of generator losses (through run-down test)	×	
Bearing vibration measurement	×	
Loading point with cos $\varphi = 0$ overexcited	×	
Heat run	×	
Measurement of the voltage curve form under no-load conditions	×	
Measurement of reactances	×	
Voltage test	×	×
Measurement of insulation resistance	×	×
Sound measurement	×	
Measurement of bearing insulation resistance	×	×

Number	Location
6	In stator winding, between coil sides (2/phase)
2	Cooling air supply
2	Cooling air exhaust
1	Cooling air exhaust from exciter

As a standard, the resistance elements have a resistance of 100 ohms at 0°C.

Bearing vibration measurement

Vibration transducers of seismic type for bearing vibration measurement can be delivered mounted on the bearing shields of the generator.

Heating elements

Heating elements are installed in both the generator and exciter unit to prevent condensation during standstill of the generator at lower temperatures.

Current transformers

Current transformers can be mounted on the stator terminals outside the generator casing. The transformers can be delivered in accordance with the purchaser's requirements.

Protective equipment

The original equipment manufacturer recommends that the generator, as a minimum, be equipped with the following protective equipment:

- Overcurrent protection
- Overvoltage protection
- Differential protection
- Negative phase-sequence current protection
- Stator earth fault protection
- Rotor earth fault protection
- Underexcitation protection and/or underexcitation limiter
- Reverse power protection (depending on the drive machine type)

- Overexcitation and/or overexcitation limiter
- Loss-of-excitation protection; in installations where there is a risk of high overvoltages, a surge diverter is to be installed and, in certain cases, protective capacitors

Operating Characteristics

Operations with constant winding temperature

With gas turbine operations, the principle described as follows is applied. This provides an optimum relation between the permitted power output of the generator and the power available from the turbine at varying cooling medium temperatures.

In accordance with international standards, particularly for gas turbine–powered generators, the generator can be loaded so that the maximum winding temperature permitted remains the same with a cooling air supply temperature other than 40°C. The winding temperature rises permitted increase or decrease as much as the temperature of the cooling medium falls below, or exceeds, respectively, the values given previously.

Synchronous compensator operation

The generators are particularly suitable for synchronous compensator operation. To permit such operation, however, mechanical disconnection of turbine and generator is usually required and one of the main bearings must be provided with thrust bearings.

Operation at low ambient temperatures

With very low temperatures the generator can be provided with a recirculation arrangement for cooling air or water.

Noise Reduction

When there are special acoustic requirements, the generator can be installed in a sound-absorbing enclosure consisting of a steel frame with panels of perforated steel sheets with sound-absorbing mineral wool in-fill.

The sealing against water leakage between the panels and the supporting structure consists of a self-adhesive rubber strip and silicon-rubber caulking.

The roof and walls of the enclosure are provided with service openings.

Base frame

The stator frame of the generator is self-supporting and therefore requires no base frame to provide stiffness. If the center height is required to be higher than standard, the generator can be provided with a separate, welded, steel base frame.

Grinding (*see* Abrasives; Some Commonly Used Specifications, Codes, Standards, and Texts)

Grinding Wheels (*see* Abrasives)

H

Hazards (*see* Color Coding; Explosion; Some Commonly Used Specifications, Codes, Standards, and Texts)

Heat Exchangers (*see also* Cogeneration; Regenerator; Vaporizers)

A heat exchanger basically removes or adds heat to a fluid. The most common types in process plants are shell and tube exchangers. Plate types (consisting of heat-conducting fins), cascade types (single pipe bent back and forth many times), and spiral plate and extended surface types are less common. The working principle behind the heat exchanger is well illustrated in the section on condensers (*see* Condensers). A heat exchanger is usually custom designed for a large process plant by the overall contract designer. Builders of items such as condensers and separators generally also make related items such as heat exchangers and will have a catalog on smaller items that can be bought without a custom order.

Some information on different commonly available heater types follows.

Heat Pumps; Heat Pumps, Geothermal; Heating Systems with a Renewable Energy Source*

Working Theory behind Geothermal Heat Pumps

How earth loops work

A system of high-density polyethylene pipes is buried in the ground or installed in a body of water to exchange heat between the building and earth. An antifreeze solution is circulated through the pipes by low wattage pumps. The plastic pipe wall becomes a heat exchanger between the fluid and the surrounding earth. In the heating mode the liquid in the pipe is cooler than the surrounding earth. In the cooling mode the opposite condition exists. Since heat flows from a warm area to a cooler one, heat exchange occurs under both conditions.

Pond and lake loops

Short polyethylene loop coils are stretched horizontally and attached to a plastic mesh to form a mat-style anchored heat exchanger. Several mats are connected together, and once in position the pipes are filled with fluid, possibly weighted, and the mats sink to the bottom. See Fig. H-1.

Open loops (well systems)

In areas where a good supply of clean ground water and an accessible water discharge system is available, an earth loop becomes unnecessary. Well water is pumped directly through the unit and heat is either extracted from or rejected back to the water table. See Fig. H-2.

---

*Source: Enertran, Canada. Adapted with permission.



FIG. H-1 Pond and lake loops. (Source: Enertran.)



FIG. H-2 Open loops (well systems). (Source: Enertran.)

Earth loop configurations

Earth loops (Figs. H-3 and H-4) are installed in either horizontal or vertical configurations; the choice depends upon geographical location and the land area available. [This information source's systems are sized to meet or exceed CSA Standard—M445 (sizing requirements), fulfilling the stringent energy efficiency requirements of the North American Building Codes.] Earth loop lengths are calculated using a sophisticated computer program that predicts annual loop performance, energy consumption, and operating costs.

Horizontal loops. Horizontal loop designs vary from a single, in-series pipe to multipipe parallel systems. Pipes are laid in trenches 4–6 ft deep, using a backhoe or trencher, and pressure tested, and then the trench is backfilled. See Fig. H-3.

Vertical loops. Vertical loops usually require less pipe than horizontal configurations. Vertical loops are connected in series or parallel or both. Drilling equipment produces small diameter holes, 75 to 300 ft deep. Two pipes are joined



FIG. H-3 Horizontal loops. (Source: Enertran.)



FIG. H-4 Vertical loops. (Source: Enertran.)

with a U-bend and are inserted into each hole. Once inserted, the pipes are pressure tested and the bore hole is backfilled. See Fig. H-4.

Liquid-to-liquid systems

Hot and cool water is forced through, or to, a hydronic transfer system, such as radiators, in-floor heating, air handlers, convectors, and fan coils.

Liquid-to-air systems

Hot and cool air is forced through an air ductwork system. Most OEMs also offer both "liquid to air" and "liquid to liquid" in one unitary system.

Partial hot water (PHW) heating

The PHW heating option transfers excess heat produced by the unit into any domestic hot water tank, whenever the system is operating. Savings of up to 65 percent per year are possible with PHW.

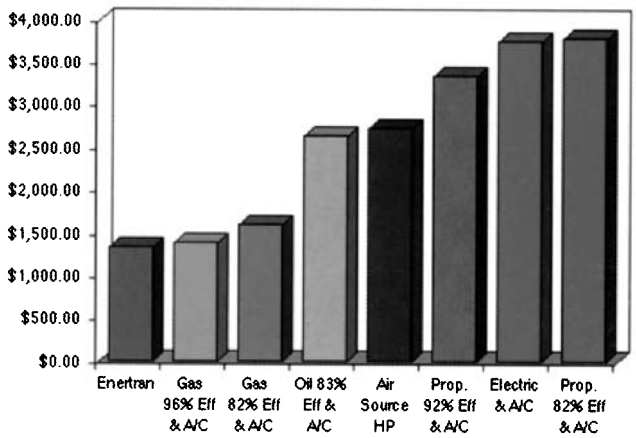


FIG. H-5 Operating cost comparison (reflects heating costs only). (Source: Enertran.)

On-demand hot water (ODHW) heating

This heating option transfers heat into hot water on demand, at any time it is required. ODHW is suitable for heating swimming pools and spas, or in-floor heating in combination with a forced air heating and cooling system.

Liquid-cooled air conditioning

Operating at approximately 50 percent less than the cost of traditional air conditioners, including very low flow rates (1 US GPM per ton), liquid-cooled air conditioners can transfer heat from a building into a liquid heat sink—such as domestic hot water, swimming pools, commercial laundries, ground water, or an earth loop. They can be retrofitted to most central forced air systems, or used for spot-cooling in difficult applications. All components are installed inside the building, reducing maintenance service cost and installation cost. Liquid-cooled air conditioning systems eliminate the need for potentially noisy outdoor condensers. See Fig. H-5.

High humidity conditions

Excessive humidity can cause major structural damage if left unattended for long periods of time. High humidity also accelerates the growth of mold, bacteria, and viruses. Indoor swimming pools, spas, and commercial operations such as indoor ice rinks and underground pumping stations, all depend on good humidity control. In a situation where dehumidification is needed and heat is needed in other areas, various heat transfer methods, including dehumidification, as well as simultaneously transferring the process heat to water/air, can be used.

Mechanical dehumidification systems

Dramatic savings on the energy cost of dehumidification, compared to conventional air makeup systems is possible. Moist air drawn through the system is cooled below its dew point, drawing moisture from the air. Heat captured by the system is

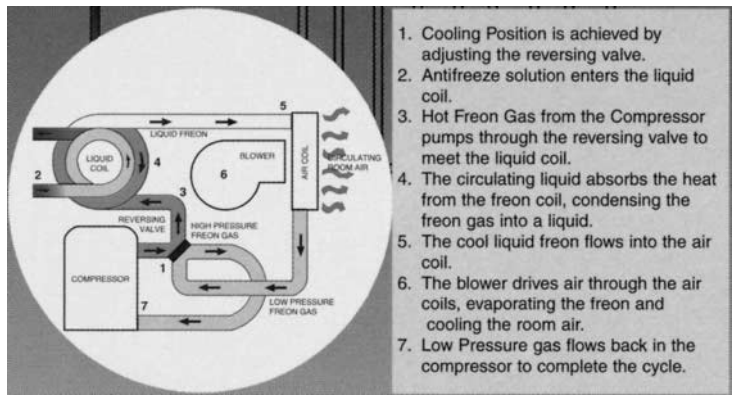


FIG. H-6 Cooling position. (Source: Enertran.)

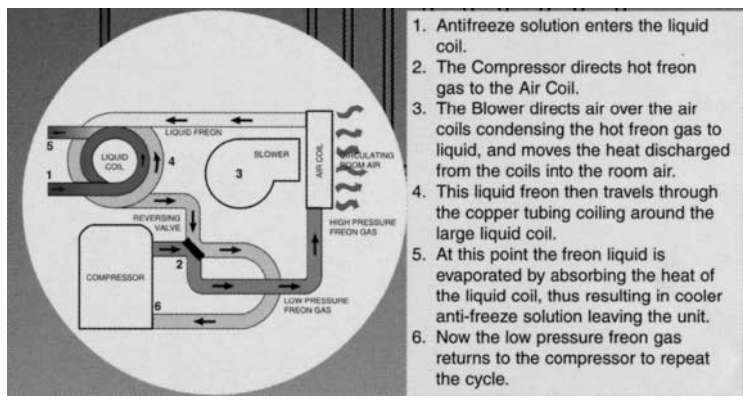


FIG. H-7 Heating position. (Source: Enertran.)

recycled back to the pool enclosure as warm dry air or back into the pool as warm water. Constant positive humidity control ensures comfortable conditions at all times. These mechanical dehumidification systems can be easily integrated with a geothermal system and other products, including liquid-cooled air conditioning, to provide economical air conditioning.

Hybrid systems

Hybrid systems combine the highest efficiency, liquid cooled air conditioning with an integrated hot water heated, forced air delivery system in one package. Heat source options can be a hot water heating system, tanks, liquid to liquid heat pumps, etc. Cooling options are identical to those used for liquid cooled air conditioners. (See Figs. H-6 through H-8.)

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.

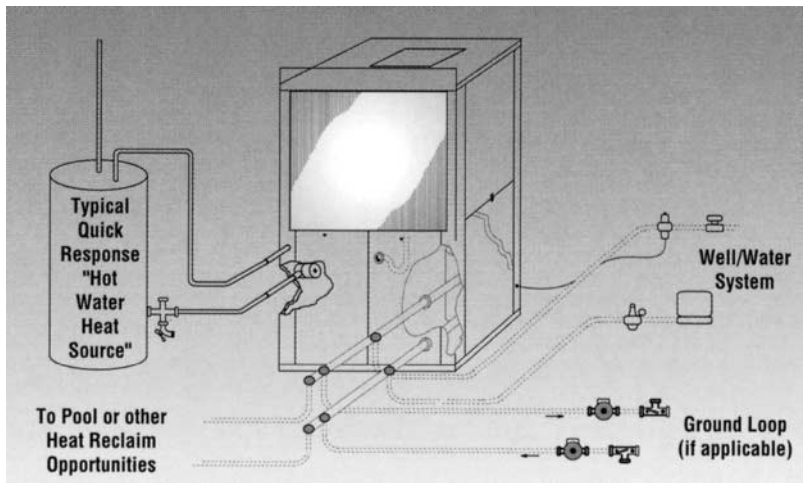


FIG. H-8 Heat reclaim options/opportunities. (Source: Enertran.)

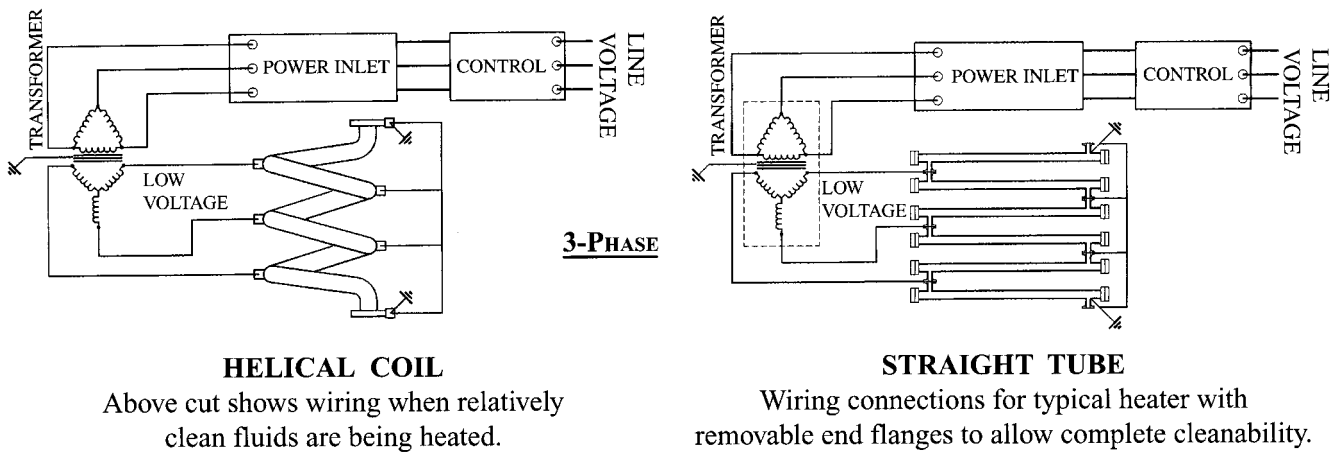


FIG. H-9 Impedance-type electric heaters. (Source: Armstrong Engineering Associates.)

Heat Treatment (*see* Metallurgy)

Heaters, Electric*

Impedance-Type Electric Heaters

The advantages of impedance heaters (see Fig. H-9) are as follows:

1. Low life-cycle costs
2. Low maintenance cost
3. Long operating life
4. Simple operating cycle
5. Easy to clean—easy access

*Source: Armstrong Engineering Associates, USA. Adapted with permission.

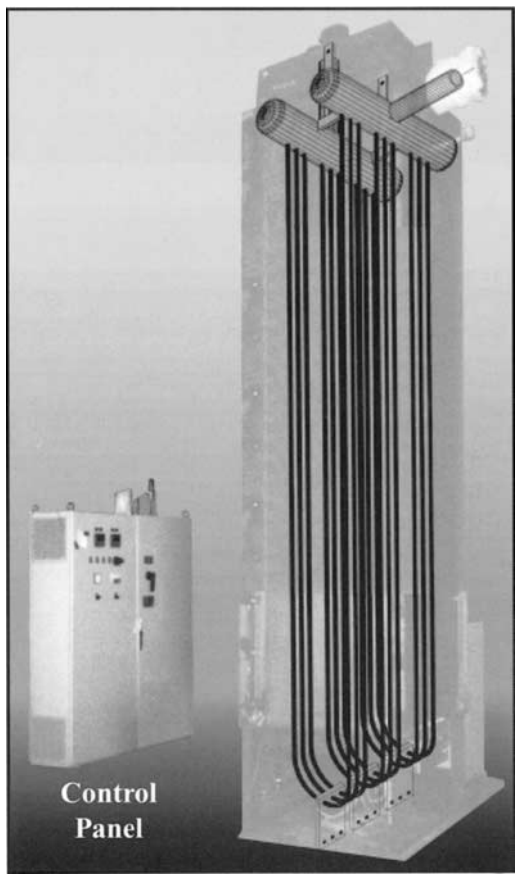


FIG. H-10 Typical impedance heater. 180 kW heating air to 1500°F (816°C) in a U.S. chemical plant. (Source: Armstrong Engineering Associates.)

6. Available in most metals
7. Close and precise temperature control
8. Ideal for heating acids, other corrosive fluids
9. Just the tube is needed—no shell is required
10. High electrical flux density is possible
11. Handles 2-phase mixtures, liquids and solids mixed
12. Suitable for high-temperature (2000°F/1093°C), low-temperature (–325°F/ –198°C), and high-pressure (up to 5000 psi/352 kg/cm^2) operating parameters
13. Heaters operate at very low voltages so there is no safety issue
14. Flow pipes carry the current and heat the fluid passing through directly
15. Less corrosion due to thicker tubes for same exit temperature of fluid
16. Accurate heat transfer data (in tube flow) allows accurate heater design

How impedance heating works

Electrical connections are secured to each end of the pipe through which the fluids flow. Refer to Fig. H-10 for the location of electric terminals on a common single-

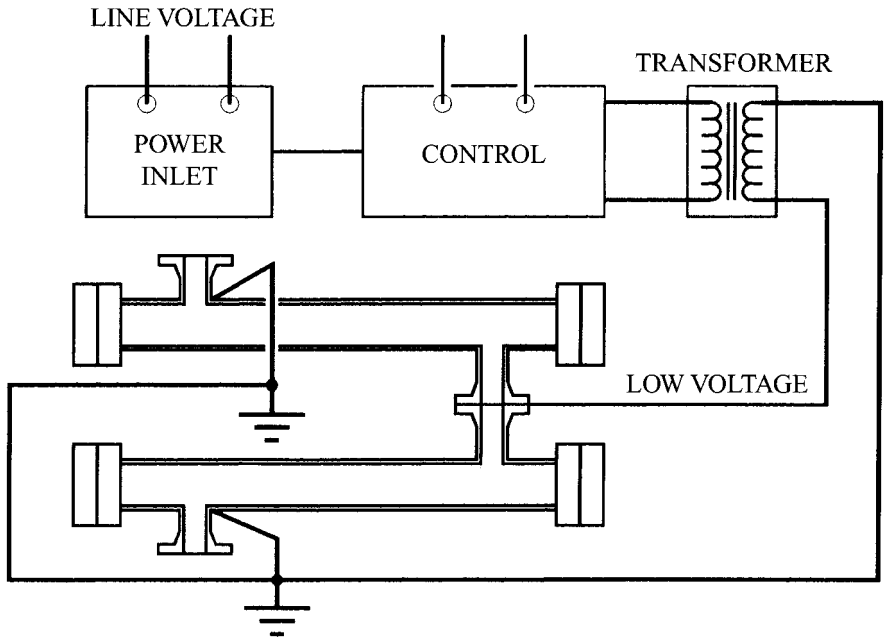


FIG. H-11 Single-phase, straight tube impedance heater. (Source: Armstrong Engineering Associates.)

phase connected arrangement. The pipe's electrical resistance causes the pipe to heat up and impart heat to the fluid flowing inside the pipe. Voltages used for this application are typically relatively low (50 V or below). See Figs. H-10 through H-12.

Heat transfer/pressure drop in impedance heaters

Since the tube wall is hot and the flow is through normal tubes or pipes, the heat transfer is easily calculated by engineers using experience in design and rating of chemical plant miscellaneous chemical fluids.

In Fig. H-12, the heater is heating air from atmospheric intake to over 1850°F (1010°C) exit temperature.

"Electrofin" Heaters

This information source has designed a specific model of heater trademarked "Electrofin." It is a compact design with the following features:

1. Up to 40 kW per single tube in a 20-ft-long (6-m) unit and more surface per tube often results in smaller shell diameter (lower cost)
2. Available in most metals including steel, stainlesses, nickel, Monel, Inconel, Incoloy, Hastelloy, etc.
3. Single tube or multiple tube combinations; lengths to 20 ft (6 m)
4. Accompanying control panels available with thyristor or contactor designs
5. Design pressures available up to 6000 psi (442 kg/cm^2)
6. Fluid temperatures available –292°F to +1202°F (–180°C to +650°F)
7. Suitable for most liquids and gases including high viscosities

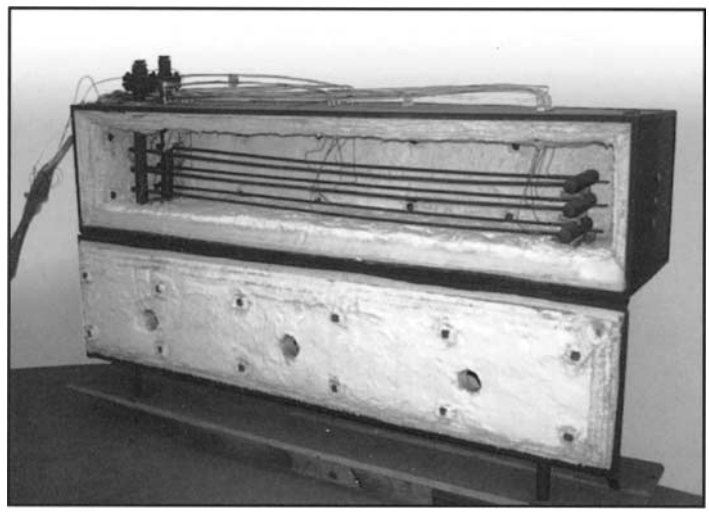


FIG. H-12 Pilot impedance heater in Romansville plant heats air to 1850°F (1010°C). (Source: Armstrong Engineering Associates.)

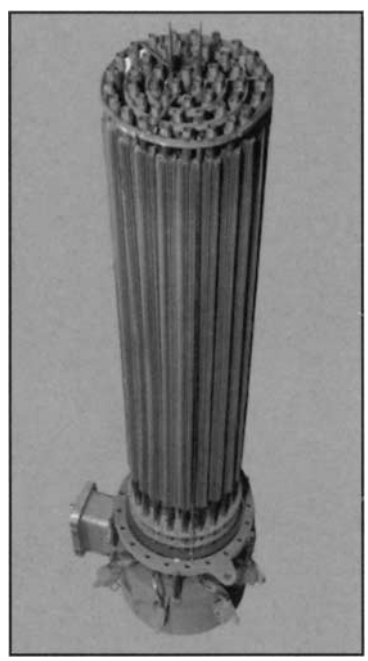


FIG. H-13 24-in-diameter (610-mm) multitube Electrofin heater operating in African chemical plant. (Source: Armstrong Engineering Associates.)

See Figs. H-13 through H-19.

"Electrofin" tank heaters (see Fig. H-20) have an attached thermocouple allowing monitoring of the heating element that can be connected to a relay in the control to allow the unit to be shut down if the internal heater overheats due to debris, fouling, etc.

Fintube tank heaters can also use steam or other normal heating fluids.

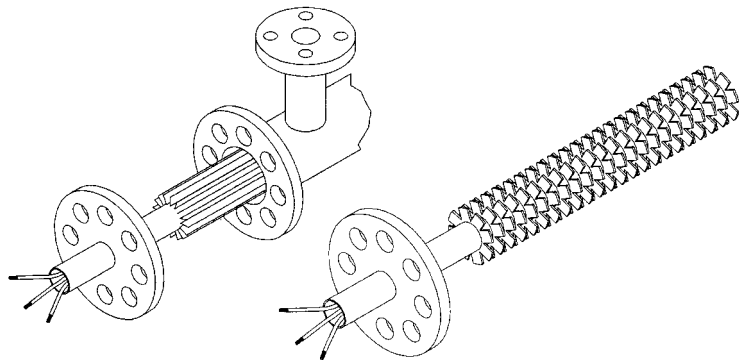


FIG. H-14 Two types of single tube heater. (Source: Armstrong Engineering Associates.)

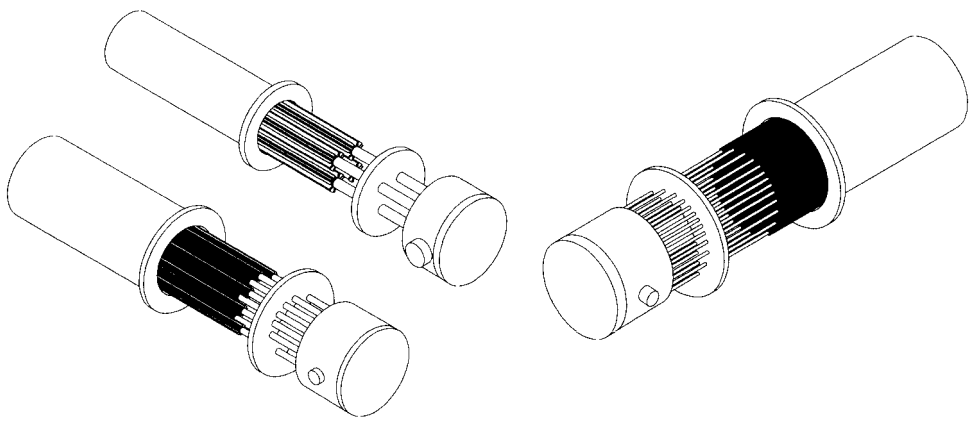


FIG. H-15 Multitube longitudinal fin tube-type heater. (Source: Armstrong Engineering Associates.)

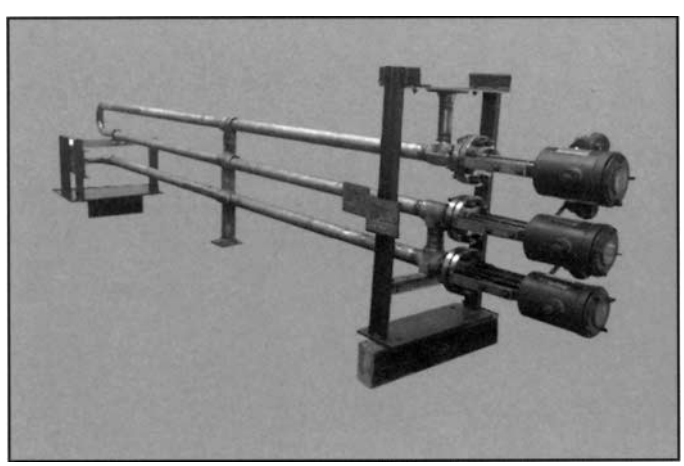


FIG. H-16 Three-stage single tube Inconel heater for heating viscous organic fluid. (Source: Armstrong Engineering Associates.)

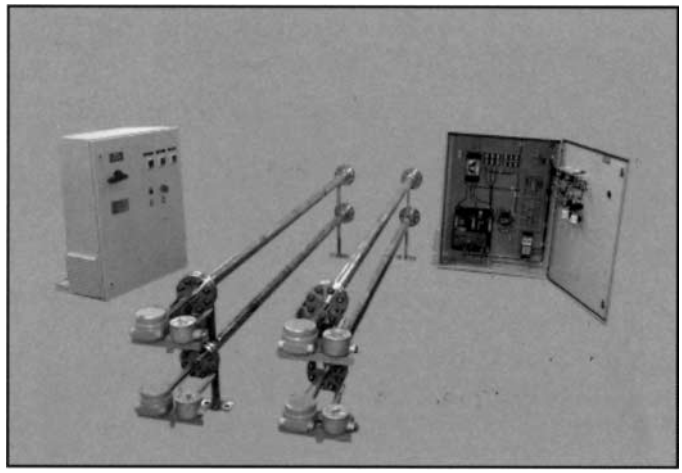


FIG. H-17 Twin single tube vapor heaters of stainless steel type 304 construction heating ethyl ether from 392°F (200°C) to 842°F (450°C). (Source: Armstrong Engineering Associates.)

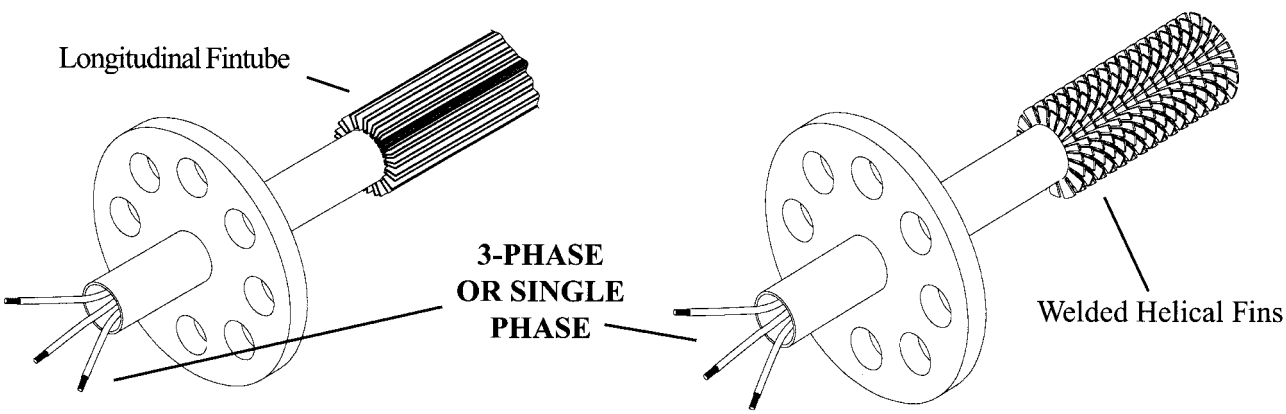


FIG. H-18 Heavy-duty electrical resistance heating elements with welded fins. (Source: Armstrong Engineering Associates.)

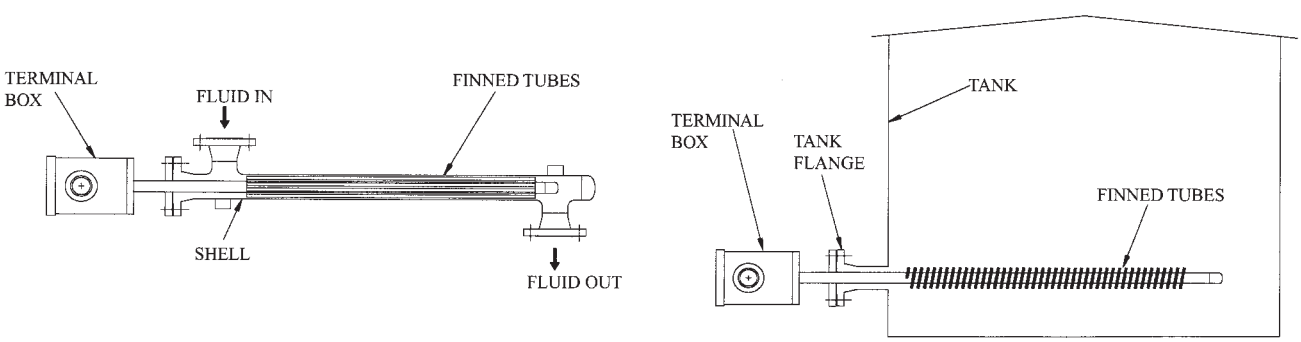


FIG. H-19 Robust thick fins are continuously welded to the electrical heating element. (Source: Armstrong Engineering Associates.)

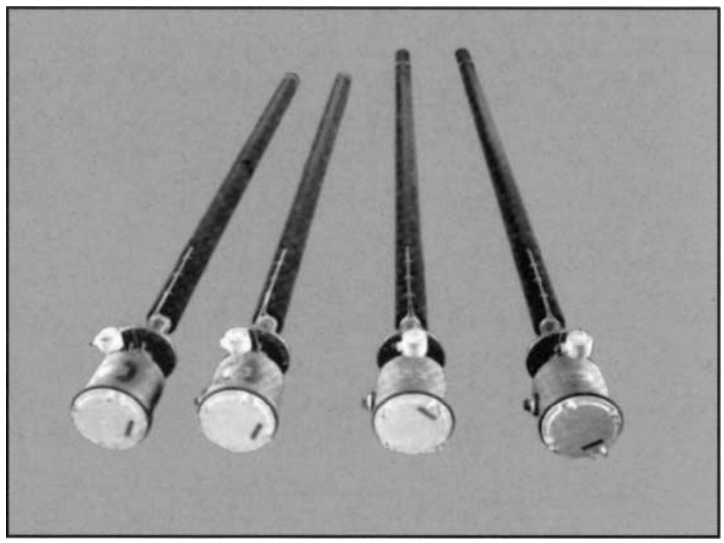


FIG. H-20 Shop photograph of CSA-approved electric hot pipe tank heaters showing terminal boxes of explosion-proof Ex(d) design. Serrated fin tube outer pipes. The internal heating elements are removable without draining tank. (Source: Armstrong Engineering Associates.)

Electric Hot Pipe Tank Heaters

This is a heavy robust heater type, strong enough for very high viscosity fluids. Features include:

1. Fins are welded to pipe. Strong contact (between fins and pipe) does not deteriorate with cycling over time as compared to wrapped or unwelded thin fins, which lose heat transfer due to rising contact resistance. The heaters do not lose capacity with age. Fins are 1.5-mm thick. Liberal fin surface gives low watt density on heating surface.
2. Fins are strong enough to allow high pressure hose cleaning and user walking on heaters.
3. Electric heaters are not subject to condensate freeze-up in the event of low temperatures.
4. Heaters may be supplied to hold temperatures from cryogenic levels up to 1200°F (650°C).
5. High turndown capacity possible due to electric heat control. Not limited by steam condensing at a minimum of 212°F (100°C).
6. Relatively simple control available.
7. Internal electric heating elements can be withdrawn from the tank without draining the tank while the outer fintube remains in the tank.
8. Available in various metals: steel, stainless steels, Monel, Inconel, Incoloy, nickel, Hastelloy, silicon bronze, etc.
9. Heaters can be supplied in long lengths. Also with bare heating pipes (no fins) if preferred.

Tank Heater Types (Steam or Electric Heated)

See Figs. H-21 through H-26.

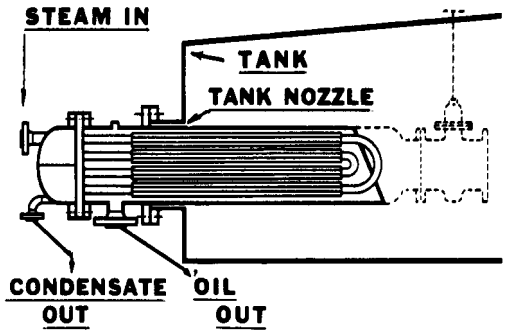


FIG. H-21 Suction-type heater, normally installed in tank side nozzle, to heat up relatively larger flows of viscous materials using steam, hot water, or other heating fluid in the tubes. These heaters normally have low shell side pressure drops to minimize npsh problems on pumps. For very high viscosity fluids, consult the manufacturer for methods of reducing intake viscosity. An internal shutoff valve can be used to allow easy bundle removal. Main uses of this type heater include bunker C, fuel oils, asphalt, molasses, caustic, etc. (Source: Armstrong Engineering Associates.)

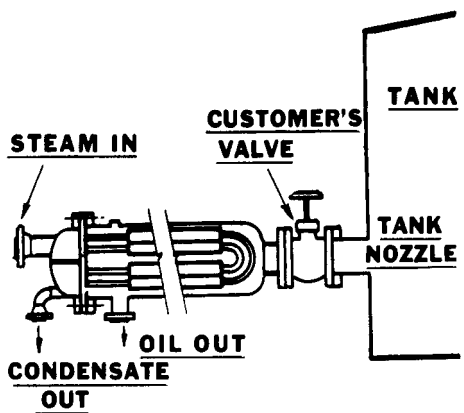


FIG. H-22 Line-type heater. This is simply a suction-type heater with an enclosed end so that the heater can be installed outside the tank. This permits addition of an external valve between the tank and the heater to allow servicing of the heater without pumping down the tank. Basically, the incentive to use a line heater is either maintenance ease or piping simplicity. (Source: Armstrong Engineering Associates.)

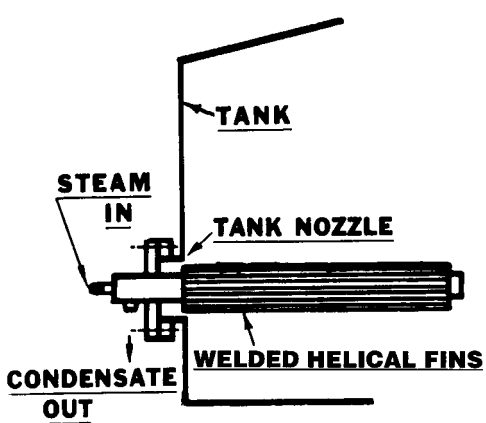


FIG. H-23 Bayonet-type stab-in tank heater. For installation in side entering tank nozzle. Bayonet heaters are usually single pipe, with fins to allow good natural convection circulation of the fluid being heated. These heaters are available in a multiplicity of metals, including the stainless steels. (Source: Armstrong Engineering Associates.)

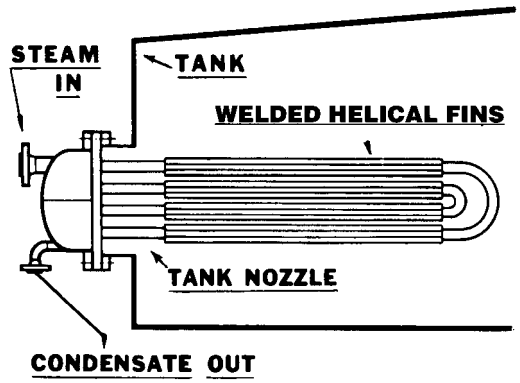


FIG. H-24 Coil-type stab-in tank heater. This type heater is installed through a side entering tank nozzle, and differs from the bayonet type in that usually more pipes or tubes than one are used, requiring therefore a tubesheet and bonnet, or a continuous coil to allow for usually lower flow of heating medium such as heat transfer oils, diphenyl, etc. Fins are helical for natural convection. (Source: Armstrong Engineering Associates.)

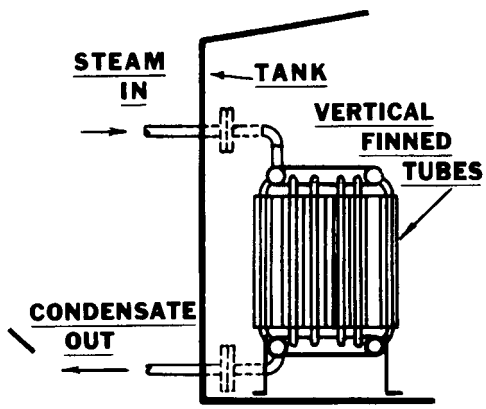


FIG. H-25 Vertical cluster "in tank" heater. This type of heater can be supplied in many metals but is usually steel or stainless steel, is used to give very large amounts of surface in a relatively confined space. These heaters can be either steam or heat transfer oil heated. Very high viscosity fluid application: By placing several such heaters around the intake of a suction heater, the fluid viscosity can be reduced sufficiently to allow the normally highly viscous fluids to enter the suction heater intake, where otherwise they might not flow well enough to ensure normal operation of the suction heaters. (Source: Armstrong Engineering Associates.)

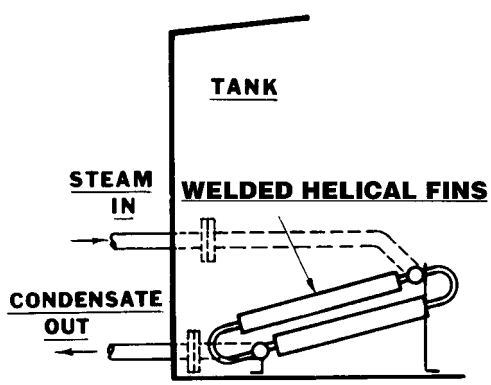


FIG. H-26 Horizontal "in tank" heater. This type heater has fins to allow good thermosiphon circulation of the fluid being heated, and gives a wide thermal current stimulation. Due to the compact surface, much larger amounts of heat can be added to the tank contents than when using bare tube coils of comparable area. (Source: Armstrong Engineering Associates.)

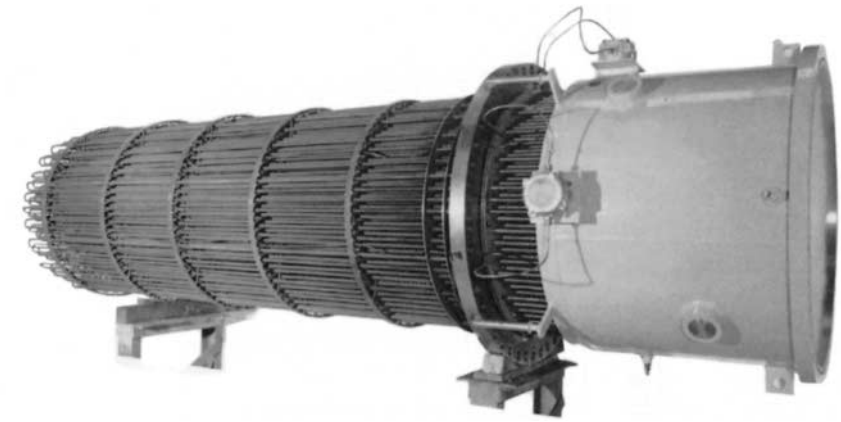


FIG. H-27 Process-type heaters are available from 2 in (51 mm) to 48 in (1219 mm) shell diameter. Unit is 42 in (1066 mm) shell diameter. (Source: Armstrong Engineering Associates.)

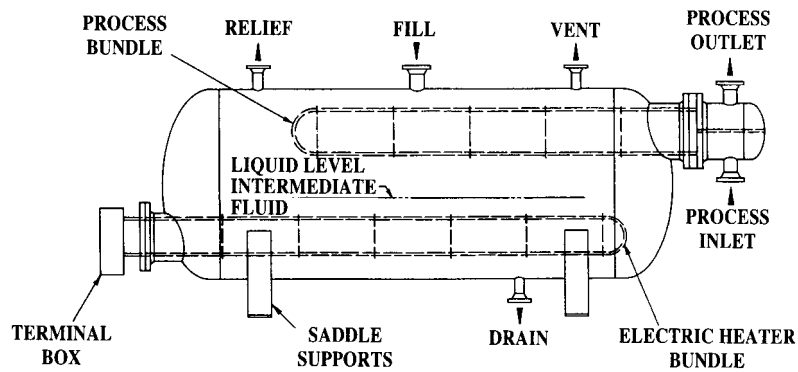


FIG. H-28 Bottom bundles are sheathed element heater. Liquid level of intermediate fluid is boiled by bottom bundle and then condenses on top bundle. Intermediate unit is used when pressure is very high or upper heating bundle is of very costly metal to avoid having the whole unit of the costly metal or the high pressure design, which is thus confined only to the upper bundle. (Source: Armstrong Engineering Associates.)

Sheathed Element Electric Process Heaters

Design pressures available depend somewhat on bundle diameter but bundles up to 10 in (254 mm) are available to very high design pressures [up to 10,000 psi (703 kg/cm^2)]. Heaters are supplied to various codes (ASME, Stoomwezen, most European, and also Lloyds, etc.). See Figs. H-27 through H-31.

Design temperatures of this unit are available from cryogenic levels up to about 1250°F (677°C) design operating temperature.

Most sizes are CSA and BASEEFA approved as well as Australian Code approved. Typical shell diameters run from 1.5 in (38.1 mm) OD up to and including 48 in (1220 mm) OD.

Shells and pressure parts are available fabricated of most pressure vessel materials (steel, stainlesses, nickel, Monel, Hastelloy, Inconel, Incoloy, etc.).

Design voltages include 600 volts and higher.

Connections for sheathed element electric process resistance heaters as well as circuit equations are in Fig. H-32.

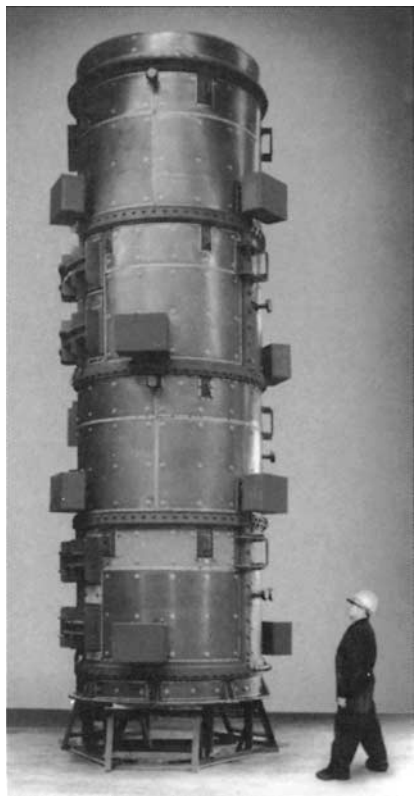


FIG. H-29 Large radiant furnace used to heat either fluidized bed vessel or to heat high pressure, high alloy, or high temperature fluid containing coils. (Source: Armstrong Engineering Associates.)

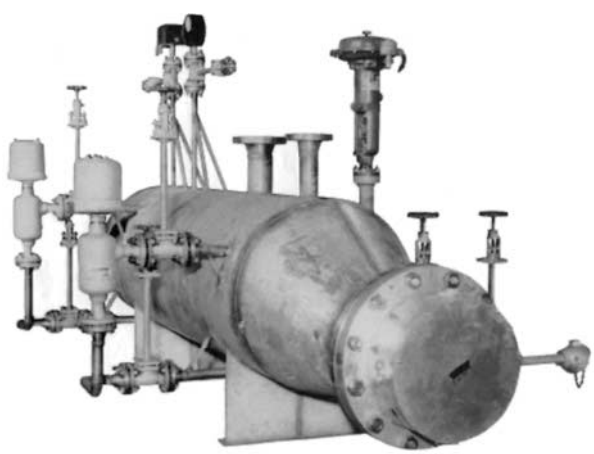


FIG. H-30 Stainless steel reboiler for mixed organic acids. (Source: Armstrong Engineering Associates.)

Controls for Electric Heaters

Controls are thyristor type or contactor type with many variations. Panels are pretested where numerous tests, such as *heat rise under load*, confirmation of *control functions*, *RF emissions*, *waveform photography*, etc. as may be required, are conducted.

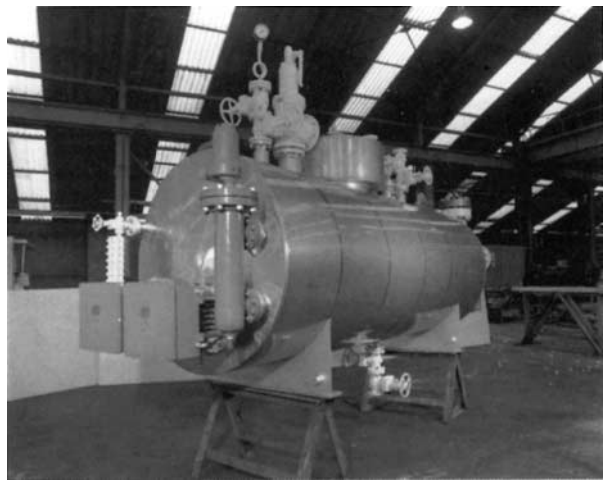


FIG. H-31 Indirect fluid electric vaporizer with controls mounted. High pressure heater of corrosive fluid. Fluid side 3175 psi (223 kg/cm^2) design pressure. (Source: Armstrong Engineering Associates.)

	THREE PHASE DELTA CONNECTION	THREE PHASE WYE CONNECTION	SINGLE PHASE CONNECTION
PHASE VOLTAGE (V_P)	$V_P = V_L$	$V_P = \frac{V_L}{\sqrt{3}}$	—
PHASE CURRENT (I_P)	$I_P = \frac{I_L}{\sqrt{3}}$	$I_P = I_L$	—
PHASE POWER (W_P)	$W_P = \frac{(V_L)^2}{R}$	$W_P = \left(\frac{V_L}{\sqrt{3}}\right)^2 \div R$	—
CIRCUIT VOLTAGE (V_L)	$V_L = V_P = \frac{I_L \cdot R}{\sqrt{3}}$	$V_L = \sqrt{3} \cdot V_P = \sqrt{3} \cdot I_L \cdot R$	$V_L = I_L \cdot R$
CIRCUIT CURRENT (I_L)	$I_L = \sqrt{3} \cdot I_P = \frac{\sqrt{3} \cdot V_L}{R}$	$I_L = I_P = \frac{V_L}{\sqrt{3} \cdot R}$	$I_L = \frac{V_L}{R}$
CIRCUIT POWER (W)	$W = 3 \cdot \frac{(V_L)^2}{R}$	$W = \frac{(V_L)^2}{R} = 3 \cdot \frac{(V_P)^2}{R}$	$W = \frac{(V_L)^2}{R} = (I_L)^2 \cdot R$
RESISTANCE (R)	$W = \sqrt{3} \cdot V_L \cdot I_L$	$W = \sqrt{3} \cdot V_L \cdot I_L$	

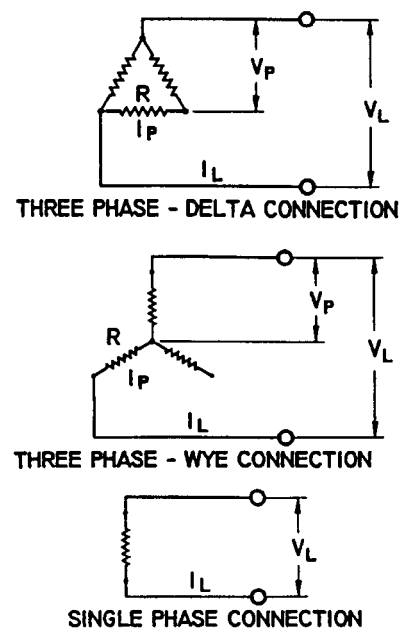


FIG. H-32 Circuitry of sheathed element electric process resistance heaters. (Source: Armstrong Engineering Associates.)

Special attention is given to heat rejection facilities for tropic or warm zone operations to ensure control elements are not overheated. In some cases, air conditioners may be attached to panels to keep cooling close to 122°F (50°C).

Features of explosion-proof control panels

The panel shell (see Figs. H-33 and H-34) must be heavily built and capable of withstanding, without damage, an explosion inside the box due to combined process fluid and air and must not cause failure or damage to the containment vessel.



FIG. H-33 Group of three control panels for 200- to 500-kW electric heaters, thyristor controlled. Note separation of control section from power section. (Source: Armstrong Engineering Associates.)

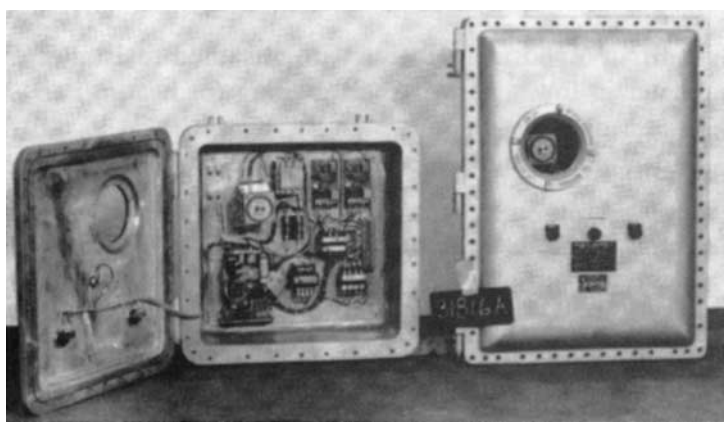


FIG. H-34 Explosion-proof NEMA 7 or Ex(d) control panel. CSA-approved explosion-proof panels may be supplied on special order. Note heavy-duty bolted enclosure for internal explosion containment. (Source: Armstrong Engineering Associates.)

External cooling may be required in the case of thyristor controls enclosed in an explosion-proof panel box. This is most easily done by attachment of a circulating water cooling heat exchanger to the back of the explosion-proof control panel box itself. A separate source of water cooling is required to furnish the cooling needed and must either be remote or explosion proof as well.

Certified control boxes for explosion-proof duty are available and nonsparking control elements for insertion into these boxes plus purge arrangements can be supplied. The final approval must be obtained from the local inspection authority.

Thyristor (SCR) control (see Fig. H-35) of three-phase power can be achieved with

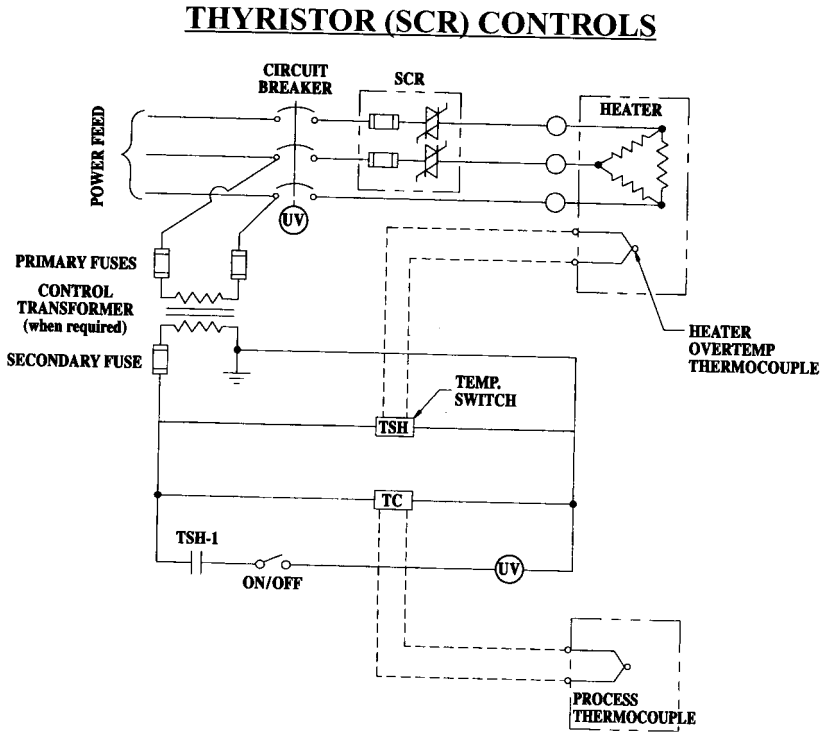


FIG. H-35 Typical, extremely simple ladder diagram of basic control system. Many additions of further control features can be included to suit the buyer. (Source: Armstrong Engineering Associates.)

either two or three thyristors. In two-leg control (more commonly used in the U.S.), the third feeder line passes directly to the load.

Note that a three-leg thyristor produces 50 percent more parasitic heat than a two-leg; therefore, 50 percent more panel cooling capability must be provided.

Extensive tests determine optimum panel cooling methods and verify cooling effectiveness. Air cooling is by natural or forced convection and water cooling as necessary to meet cooling requirements.

Contactor controls (see Fig. H-36) are generally simpler and less costly than thyristor controls. However, being on-off devices, they do not offer the high level of outlet temperature control and turndown capability available from thyristors. Contactor control is most frequently used for tank heating applications.

Also see Figs. H-37 and H-38.

Ohmic Electric Heaters

How ohmic heating works

Electrodes are immersed directly into a suitable electrically conductive fluid that flows through a nonconductive piping system. The resistance to the electric current passing through the fluid generates heat within the fluid itself. See Fig. H-39.

Controls for ohmic heating

Incoming power is transformed to suitable voltage consistent with the electrical resistance of the fluid being heated. Ohmic heating applications may often involve

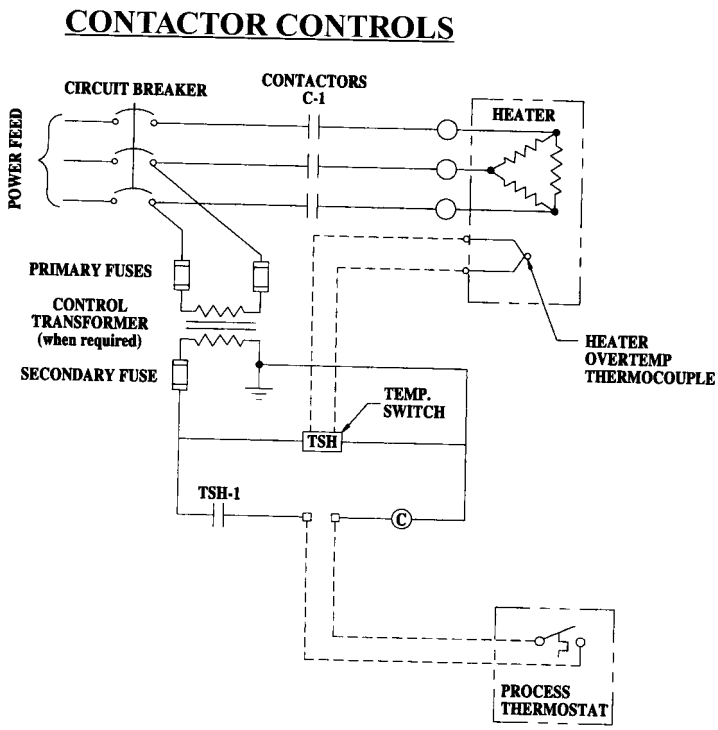


FIG. H-36 Typical simple control schematic ladder diagram including a thermostat. Many optional control features may also be included to suit process specific needs. (Source: Armstrong Engineering Associates.)

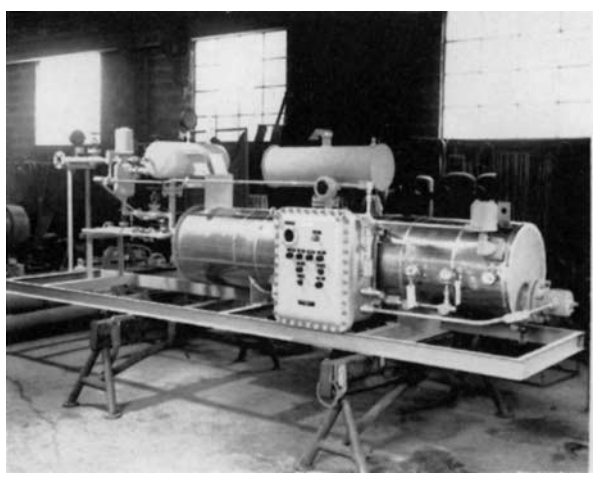


FIG. H-37 Indirect fluid electric heater insulated and mounted on skid with all controls in place. One of several at the same site in South America. Capacity 24 kW (20,650 kcal/hr) vaporizing organic fluids. (Source: Armstrong Engineering Associates.)

relatively high voltage and low current. The controls for temperatures and heat input are installed ahead of the transformers in most cases.

Piping materials used in ohmic heaters must be nonconductive. Many forms of plastics and glass-lined metal have been successfully used in ohmic heating applications.

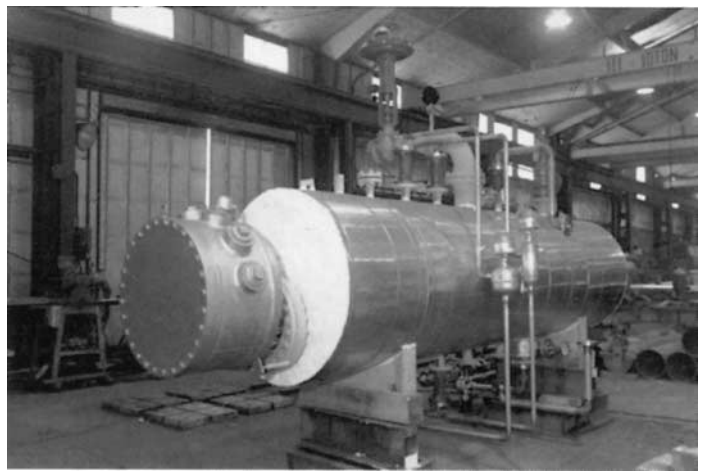


FIG. H-38 Electric vaporizer with insulation factory applied. 967 kW (831,620 kcal/hr) boiling mixed organics. Stainless steel construction. (Source: Armstrong Engineering Associates.)

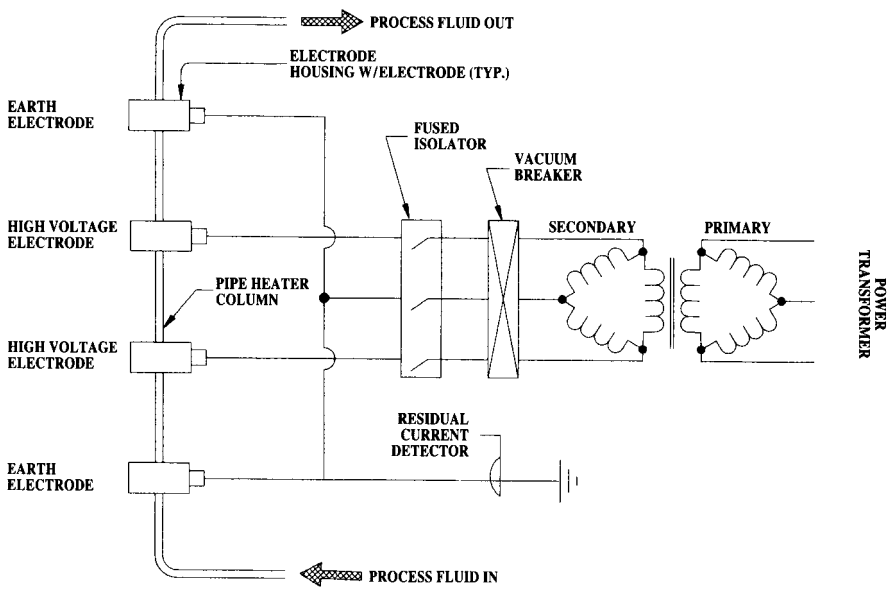


FIG. H-39 Typical three-phase delta connected ohmic heater. In this arrangement, one of several possible, heating occurs within three pipe sections supplied with four electrodes. (Source: Armstrong Engineering Associates.)

Advantages of ohmic heating

Ohmic heater electrodes and piping systems tend to remain clean with fluids that may heavily foul other types of equipment. This is because of the absence of any elevation in fluid film temperatures at the heating surface.

Certain highly corrosive yet conductive chemical fluids can be heated by ohmic methods.

Since the current flows through solids as well as liquids, the heat transfer to certain flurries can be made much more uniform than by other methods of heating.

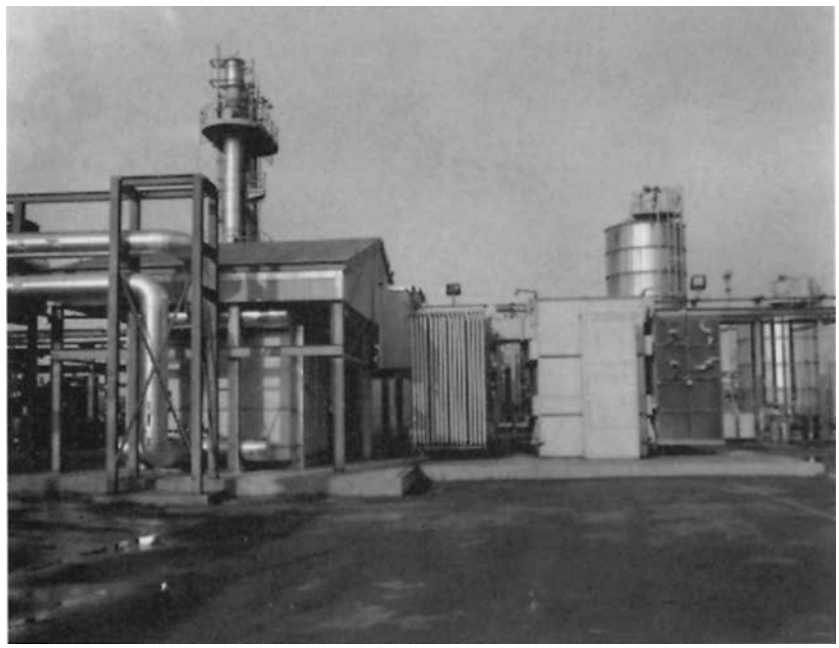


FIG. H-40 16,000 kW Dowtherm heater in west coast U.S. chemical plant. (Source: Armstrong Engineering Associates.)

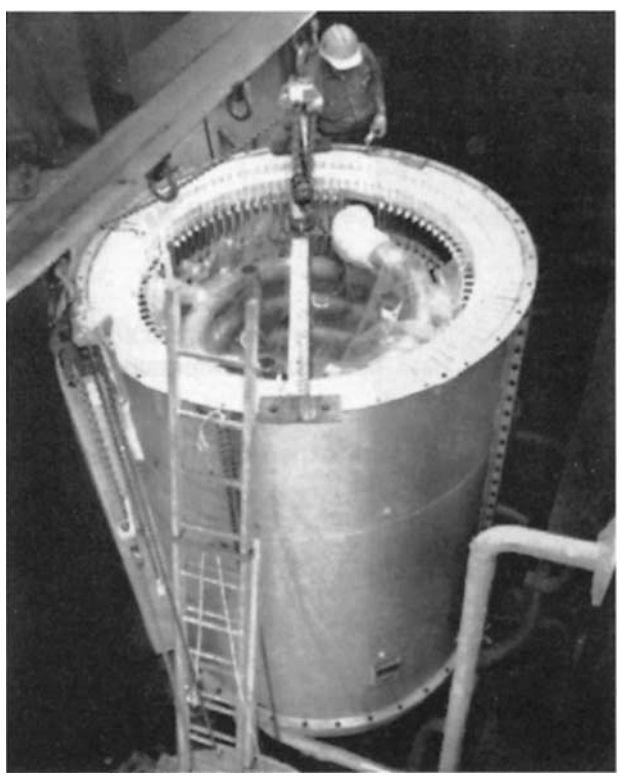


FIG. H-41 900-kW radiant heater for liquid sodium. (Source: Armstrong Engineering Associates.)



FIG. H-42 1000-kW radiant furnace. Steel vertical vaporizer to heat 1000 kW, 94-in-diameter (2388 mm), 42 ft (12.8 m) overall height. (Source: Armstrong Engineering Associates.)

Heavy viscous materials that may be difficult to heat by conventional equipment can be handled by ohmic heating, including certain sludges, etc.

Radiant Heaters

Radiant heaters are furnaces supplying electric radiant heat to process fluids. They are useful for very high temperature heating (up to 2000°F/1094°C) or for some very corrosive fluids. Excellent control is possible. They are useful for high kW heating and generally the largest electric capacity available over 15,000 kW. See Figs. H-40 through H-42.

High-Speed Drive Systems (*see* Power Transmission)

Hot Isostatic Pressing (or HIPing) (*see* Metallurgy)

Hydraulic Filters (*see* Filters)

Industrial Ecological Park (*see* Ecological Parks)

Industrial Ecology (*see* Ecological Parks)

Inlet Ducts and Silencers (*see* Acoustic Enclosures, Turbine; Air Filtration; Ducting)

Instrumentation (*see* Condition Monitoring; Measurement)

Insulation (*see* Some Commonly Used Specifications, Codes, Standards, and Texts)

Irradiation, Food Product*†

The push to improve food quality through irradiation began early in the 20th century, when researchers aimed newly discovered X rays at foodstuffs to preserve them. In the 1950s, the availability of manmade isotopes such as cobalt-60, used to sterilize medical equipment, changed the course of food irradiation. Gamma rays emitted by the isotope were able to destroy pathogens in food as effectively as more expensive technologies such as an electron accelerator. Since then, the technique has been applied to such items as poultry, fruits and vegetables, and spices.

For example, in 1986, a company called SteriGenics began irradiating dry-food ingredients such as pepper, onion powder, and dehydrated vegetable powder at facilities in Tustin, Calif.; Schaumburg, Ill.; Rockaway, N.J.; and Salem, N.J. These plants irradiate approximately 50 million pounds of spices annually. The company was founded in 1979 to sterilize single-use, disposable medical products such as syringes and gowns.

The spices arrive at SteriGenics plant warehouses in bulk form in bags and drums, or sometimes in their final form in boxes. Workers affix dosimeters, such as those made by Far West Technology in Goleta, Calif., on the containers before loading them into metal containers or totes. The totes are loaded onto carriers that are suspended from an overhead monorail to move them into an irradiating treatment cell. The cell walls and ceiling are $6^1/_2$-feet-thick concrete poured around steel rebar to ensure that no crack can penetrate the walls. See Fig. I-1.

The totes are exposed to gamma rays with short wavelengths, similar to ultraviolet light and microwaves, emitted from an array of cobalt-60 "pencils" installed on either side of an 8- by 16-foot stainless-steel rack. The pencils are actually stainless-steel tubes containing two zircon alloy tubes that encapsulate nickel-coated pellets of cobalt-60. When the pencils are not in use—during maintenance, for example—they are submerged in a 26-feet-deep pool of deionized water, more than twice the depth needed to protect maintenance workers when the array is submerged, and raised when irradiation recommences.

A U-shaped overhead conveyor in the cell guides the totes until they are exposed

---

* Source: Adapted from extracts from "Keeping Food Germ Free," *Mechanical Engineering*, ASME, March 1998.

† Note: Financial figures in this section reflect known costs and prices in 1998.

for a timed interval to the desired absorbed dosage of gamma radiation, a maximum of 30 kilograys for spices. (A gray, measuring the absorbed dose of ionizing radiation, is equal to 1 joule per kilogram.) The treated totes are returned to the warehouse on the other side of the conveyor dividing the loading and unloading operations. The spice containers are removed from the totes and shipped to customers.

SteriGenics retrofit its Tustin plant in 1996 to treat low-dose foods requiring less than 1 kilogray of radiation: "fresh vegetables, including avocado, onions, celery, bell peppers, and broccoli, that are sold either for retail sale or as ingredients for other products such as a salsa." The shelf life of fresh produce irradiated at Tustin is extended by up to two weeks.

Palletized Loads

The year before SteriGenics began irradiating food, FTSI, another American food irradiator, was formed because the Florida Citrus Commission sought an alternative to methyl bromide as a quarantine treatment for citrus. The agency was acting on an Environmental Protection Agency suggestion that methyl bromide, used to prevent the spread of fruit flies, would be banned in 2001.

The company treats several truckloads of packaged agricultural products per week for both local and national brokers and distributors. These foods are sold to retail establishments and institutions. The food-irradiation industry is based on public health concerns, such as food poisoning, rather than economic benefits, according to some, although such economic benefits do accrue, specifically by extending shelf life and serving as a quarantine measure. The process eliminates sprouting in tubers such as potatoes, garlic, and onions, for example. Irradiation also delays the ripening of certain fruits and vegetables, including strawberries, tomatoes, and mushrooms, which are the main crops treated at FTSI.

As a quarantine measure, irradiation kills the larva of insect pests such as fruit flies and seed weevils in mangoes, preventing them from spreading between growing regions. Irradiation can also be used to pasteurize seafood.

FTSI's irradiator and safety control system were designed and built by MDS Nordion in Kanata, Ontario, a major supplier of cobalt-60 as well as a designer of medical-sterilization plants and research-irradiator equipment.

The MDS engineers used their own controls and interlocks for the FTSI safety system, which also included radiation monitors, restricted openings, and a procedure to replace cobalt-60 pencils underwater with magnifying lenses and manipulators. Several hundred different conditions will automatically shut down the system in case of component failure or system inconsistency.

Unlike the process loops at other facilities, FTSI can irradiate large pallets of packaged foods. MDS engineers had to build a plant that would handle the heavier loads than their earlier systems could. This involved scaling up the 48- by 24-in conveyors used in facilities like the Canadian Irradiation Facility in Montreal to 48 by 42 in to handle U.S. pallet sizes. This involved testing monorails, bearings, wheels, and I-beams to build an overhead conveyor that could transport the 2-ton loads in a single carrier. They also used hydraulic cushions to gently stop the larger loads, and built forklift clearances so the pallets could be loaded and unloaded from the carriers.

These carriers are 18-feet-tall, 4-feet-wide, 4-feet-deep aluminum boxes holding a shelf at floor and mid-level. Forklift operators load a pallet on each shelf of the carrier. The carriers are transported from the warehouse into the $6^1/_2$-feet-thick concrete treatment cell via a pneumatic overhead monorail. Once inside, hydraulics move the carriers, primarily because greater locational accuracy of movement is required; this also reduces the number of cylinders needed to carry the heavy loads.

As in the SeriGenics process, the FTSI carriers follow a U-shaped trajectory in the treatment cell that exposes them to gamma rays from cobalt-60 pencils. The exposure time in the concrete cell and other process parameters is directed by Omron programmable logic controllers.

After being irradiated, the carriers return to the warehouse on the other side of an interior chain-link fence that separates the two halves of the irradiating process. The pallets are removed by forklift and placed on trucks for delivery.

Irradiation's Future

The most likely scenario for plant construction in the future is building dedicated food-irradiation plants either at or as near as possible to the point of transportation and distribution, after the final packaging and labeling is complete, to prevent the possibility of recontamination after irradiation.

Location is important, especially because rising freight costs could exceed irradiation costs. (An increase in gasoline prices could also affect the economics of irradiation.) Relatively few plants for FTSI, each estimated at $10 million to $12 million, would suffice. For example, a well-built, well-located spice plant could treat up to 150 million pounds of product a year.

Food irradiation is a technology whose time has come, until a better alternative is found, because food inspection and testing are revealing more incidents of food-borne disease.

Accelerating Irradiation

Only two companies—SteriGenics International in Tustin, Calif.; and Food Technology Service Inc. in Mulberry, Fla.—irradiate food in the United States, and both use the same basic process involving cobalt-60 isotopes. Linear accelerators, however, may be the technology of future plants. These devices emit a beam of electrons that directly contact the product or convert the accelerated electrons into X rays, which can penetrate deeper than an electron beam but are less efficient.

Most linear-acceleration operations opt for showering the product with electrons. The Utilization Center for Agricultural Products at lowa State University in Ames uses this technique. A meat scientist and director of the center is conducting food-irradiation experiments on meat products with a Circe 3 irradiator built by Thomson CFE in Saint-Aubin, France (see Fig. I-2). The center's work is sponsored by the U.S. Army, the Agriculture Department, and the Electric Power Research Institute in Palo Alto, Calif., with grants from meat producers' groups including the Cattlemen's Association and Pork Producers.

The device is used to address two major concerns facing linear-accelerator irradiation. The first is to determine which packaging materials can be used in irradiation without causing compounds to migrate into the food. The second goal is to determine the optimum packaging environment for meat products being irradiated—specifically, vacuum packing, a modified atmosphere such as nitrogen blanket, or oxygen.

The researchers load meat products onto carts attached to conveyor chains that transport the carts through the $9^{3}/_{4}$-feet-thick concrete walls of the irradiation area. The floor-mounted Circe 3 contains an electron gun comprising a cathode and anode that generate electrons, which are pulsed into an accelerating tube. At the same time, radio-frequency power is pulsed into the accelerating tube by a klystron, forming waves for the electrons to follow.

A series of alternating magnets in the tube accelerate the electrons to the high energy levels required for irradiation. When they reach the end of the tube, the

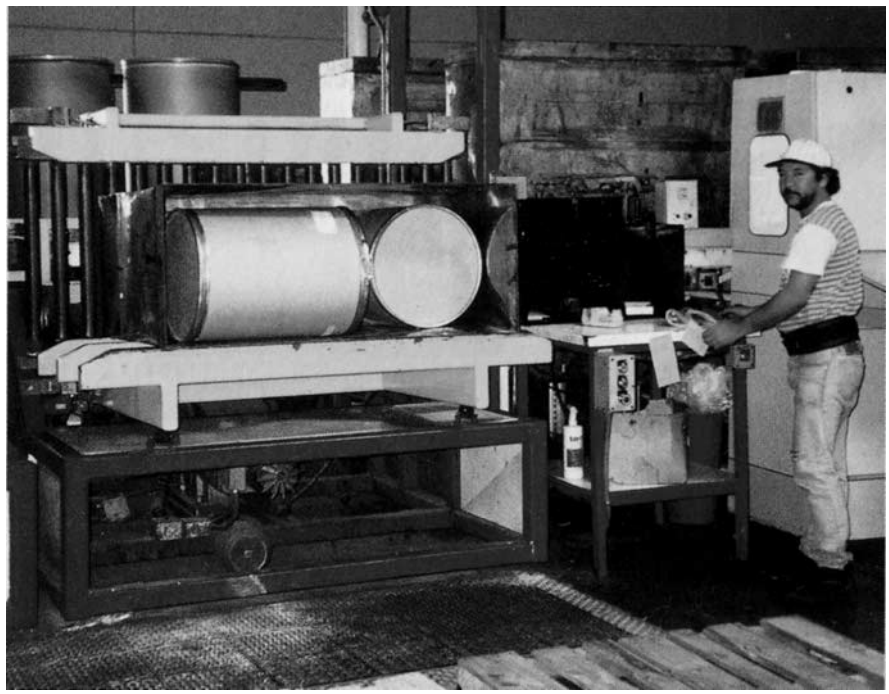


FIG. I-1 A worker at the SteriGenics irradiation facility in Tustin, Calif., loads drums of spices onto carriers that will be transported to an irradiation cell to reduce the presence of microbes in the food. (Source: "Keeping Food Germ Free.")



FIG. I-2 Dennis Olson conducts research using a Circe 3 linear accelerator to irradiate meat products at Iowa State University. (Source: "Keeping Food Germ Free.")

electrons pass through a Glaser lens that focuses them into a beam. The beam is bent by a magnet to a 107° angle, so only the particles of the selected energy level are emitted. Those filtered electrons pass through a scanning magnet and sweep across the meat's surface, changing the DNA of microbes in the food and killing these pathogens.

Three different energy levels can be selected: 5 million, 7.5 million, and 10 million electron volts, which can penetrate $^3/_4$ in, 1 in, and 1.5 in on one side, respectively, or $1^3/_4$ in, $2^1/_4$ in, and $3^1/_2$ in if both sides are irradiated. Some electrons are picked up when irradiating both sides, which is why the penetration is more than double.

A multilayered safety system is used at the Ames irradiation facility, starting with the maze through which the carts are conveyed—three 90° turns that serve as a biological shield, preventing electrons or X rays from ricocheting to the product-handling area. This and other safety devices are wired to the control panel and to the Circe 3. If one electric path fails, the other will trip the electron source and shut down the facility.

L

LCA (*see* Life-Cycle Assessment)

LNG (*see* Liquid Natural Gas)

Laser Cutting, Drilling, Machining, Welding (*see* Metallurgy)

Life-Cycle Assessment (LCA) (of Turbomachinery)*

Life-cycle assessment is the art of being able to use data about:

1. A machine, its operating history, and any failures
2. The fleet of that type and model of machine
3. Changing process conditions in a plant to assess or extend the life of turbomachinery components or the machine itself.

The data can be used to:

1. Avoid catastrophic failures
2. Improve the quality of the machine's operation (lower operational temperature, promoting less hot erosion)
3. Assess better the life cycles being accumulated on a machine to avoid premature overhauls or parts change
4. A combination of all of these

Many consulting companies today sell life-assessment services. Some of them provide a package that includes monitors, interpreting the data for their clients and so forth. Even if clients decide to pay for this service, they should get some basic training for in-house staff, so they can better assess their options. Also, any consulting service could be an interim measure if staff training is provided, so the clients can eventually proceed on their own.

For accurate life-cycle assessment, it is essential that the individual machine's parameters be used. Even among machines of exactly the same type and model, stress levels may be considerably different in actual operation.

Consider the following severe example. In a squadron of aircraft doing acrobatic maneuvers in formation for an air show, the squadron leader, generally out in front, sets the pace. The other aircraft take up their positions, in tight formation, attempting to stay a specific distance from their leader's wing tips. The stress of the aircraft's engines that are not leading is much higher than that of the leader's. This is because their power levels have to "hunt" back and forth to maintain their required positions. These engines may incur stress of as much as 20 times that of the engines in the leader's aircraft.

In a less extreme example, consider three similar pumps operating in parallel

* Source: Soares, C. M., LCA course notes, 2000.

TABLE L-1 Hypothetical* Sample Format for Component Allowable Lives

Part	Allowed Cycles	Values of Constants
LPC		$c_1 = 1 = c_2$ $k = 0.3$ $p = 3$ (Constants provided by the OEM)
St. 1 disk	6,000	
St. 1 spacer	5,000	
St. 2 disk	10,000	
St. 2 spacer	4,500	
And so on		
LP coupling shafts	40,000 and 20,000	
LP turbine		
Shaft	8,000	
St. 1 disk	5,000	
St. 1 spacer	5,000	
And so on		
HPC		
St. 1 disk	8,500	
St. 1 spacer	4,800	
St. 2 disk	8,000	
St. 2 spacer	7,500	
And so on		
HP coupling shafts	10,000 and 15,000	
HP turbine		
Shaft	15,000	
Disk	5,500	

* Imaginary values chosen for illustrative purposes. These data values are normally proprietary to the individual manufacturers.

in a plant. One of them always seems to have alignment and balance problems, although they all operate at nominally the same process conditions. When they are checked, it is found that extremely high piping stress exerts cantilevered load on the problem pump. The vibration, caused by misalignment and imbalance, even though it may not result in catastrophic failure, may result in this pump requiring overhaul before the other two.

Original equipment manufacturers (OEMs) conduct life-cycle analysis (LCA) to track the validity of their designs as a whole and the performance of newly designed modifications.

Life Usage

Life cycle

A *cycle of life* is a unit that defines a measurable unit of life in a piece of turbomachinery. In many cases, with lower turbine inlet-temperature gas turbines, one cycle is defined as an hour. If the gas turbine always ran at base load, this might be a fair assessment. However, if the gas turbine is at peak temperature, overload, or otherwise stressful condition, an hour of running may add up to more used-up life than if the machine were at base load. If one cycle is what is incurred by running for 1 hour at base load, then more demanding operation will "cost" more.

A usage cycle may or may not, depending on the OEM, embody temperature as a parameter necessary for definition. The OEM may define its cycle for a given model, for instance, by maximum and minimum speed values, each as a percentage of maximum speed versus time.

OEMs might conduct LCA at the request of customers who are concerned about premature parts removal. The reason for removal of parts might vary from machine model to machine model or from component to component. (See Table L-1.)

Alogarithms for life-cycle use

An "algorithm to calculate life-cycle use" is just a technical term for a calculation to determine how much useful life of a component has been used up. If this is information that an operator needs to know (in other words, the OEM may or may not track it for you), then the OEM can give all owners of that machinery model a service bulletin (SB) to cover the details they need to know.

Let us consider the following hypothetical (actual OEM service bulletins to describe their algorithms are proprietary) example that describes the information that an OEM would provide in such an SB.

Format of an LCA SB

1. Number and title of the SB.
2. Model number(s) applicability.
3. Statement of why the SB has been introduced: At operator's request/for improved parts life/optimized operational cost/and so forth.
4. Definition of how the OEM defines "cycle" in terms of
   - Cycle type [for instance, "fatigue cycle" if low-cycle fatigue (LCF) is a factor].
   - Operational parameters that define the cycle [for instance, "a speed change from 0 percent gas generator speed to gas generator (GG) reference speed and back to 0 percent].
5. Calculations and corresponding action outlined in detail, i.e., calculation procedures, definition of reference speeds, which will vary for each rotor, and

$$\text{Sample cyclic usage factor}^{*} = \left[\frac{c_1(N_1)^2 - c_2(N_2)^2}{(N_{\text{ref}})^2 - k(N_2)^2} \right]^p$$

where N_{ref} = reference speed for that rotor
N_1 = highest operating speed
N_2 = lowest operating speed
k = stress at reference speed/ultimate tensile stress at design temperature
p = slope of low-cycle fatigue curve
c_1 and c_2 are constants.

The OEM will specify the calculation sequence. A sequence may be laid out for how to represent all operational peaks in one cycle, which peak to apply the calculation to first, and how to proceed to the next step.

When this is complete, the peaks that equate to a life cycle are totaled to give a cyclic usage factor for that operational cycle only. Some machinery comes with life-cycle counters to automatically add up these life cycles, or the counters and the LCA process may be retrofitted.

For a cycle with only one peak (or plateau), Fig. L-1B, the simplification to Fig. L-1A can be made.

For a cycle with more than one peak (or plateau), see Fig. L-2A. This can be simplified to Fig. L-2B.

Calculation of the first peak (specified as the smallest speed change) is shown in Fig. L-2C. Note dotted line. This then makes the curve shown in Fig. L-2D, which then simplifies to the curve given in Fig. L-2E, so the calculation then is (for constants $k = 0.3$, $p = 3$, $c_1 = c_2 = 1$):

---

* Note: This is an actual equation used in a comtemporary gas turbine OEM's SB that saved its end users' component lives by factors exceeding 200 percent.

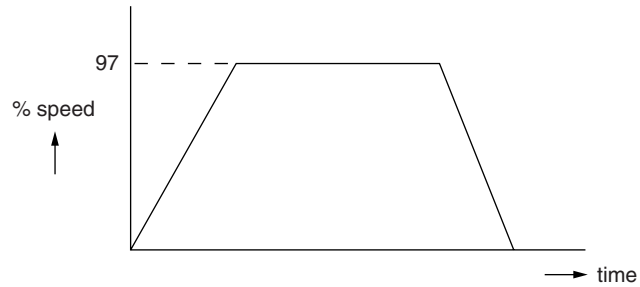


FIG. L-1A Hypothetical curve of percentage (%) speed versus time for a gas turbine.

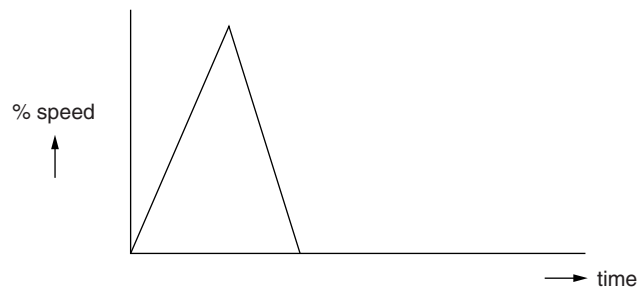


FIG. L-1B Modification of the curve in Fig. L-1A.

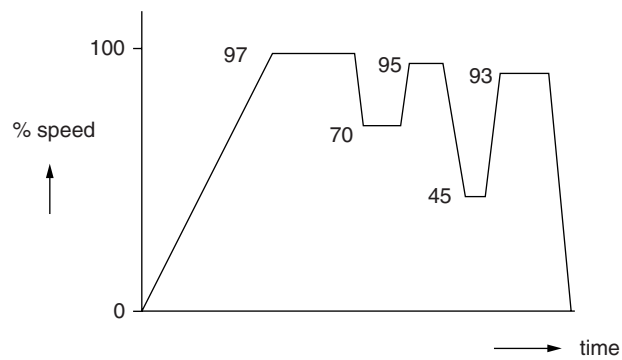


FIG. L-2A A cycle with more than one peak.

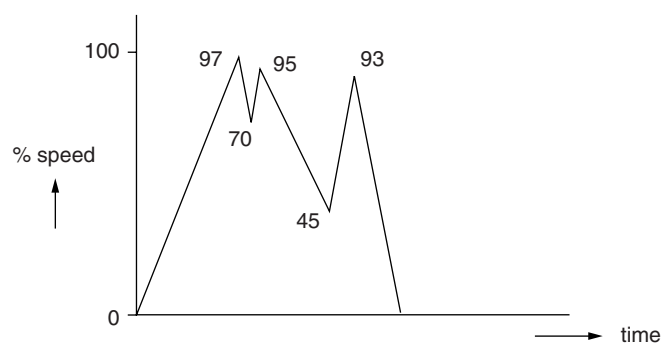


FIG. L-2B Simplification of Fig. L-2A.

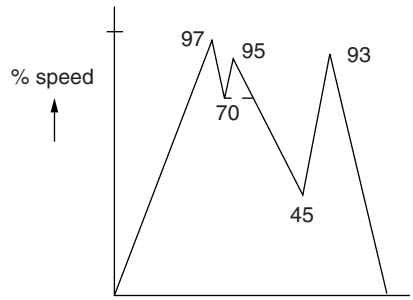


FIG. L-2C Calculation of the first peak.

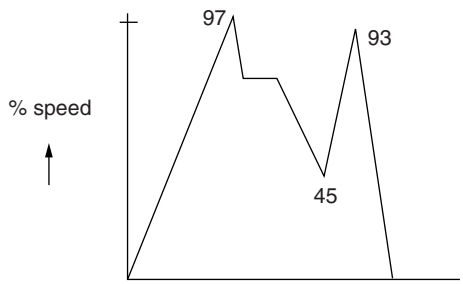


FIG. L-2D Curve after first peak calculation.

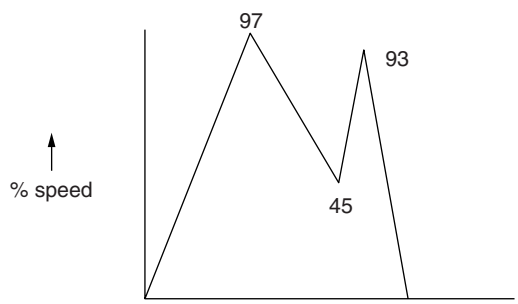


FIG. L-2E Curve simplified after first peak calculation.

$$\text{First peak cyclic usage factor} = \left[\frac{95^2 - 70^2}{100^2 - 0.3(70)^2}\right]^3$$

$$\text{Second peak cyclic usage factor} = \left[\frac{97^2 - 45^2}{100^2 - 0.3(45)^2}\right]^3$$

$$\text{Third peak cyclic usage factor} = \left[\frac{93^2 - 0^2}{100^2 - 0}\right]^3$$

For the overall cycle, add the previous three peaks.

The OEM may also provide graphs that assist with calculation of life-cycle usage. Typically, these would specify curves corresponding to the upper operating speed. The lower operating speed might be along one axis and effective cycle usage on the other axis (see Fig. L-3).

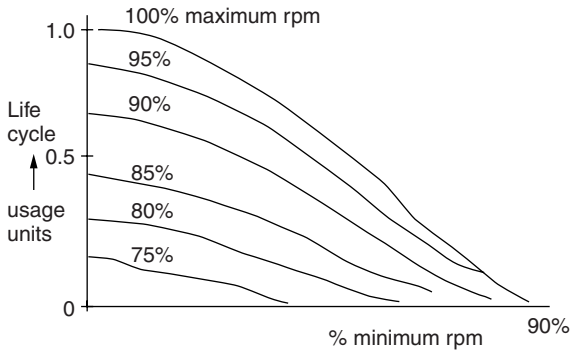


FIG. L-3 Cycle usage curves valid for specific constants.

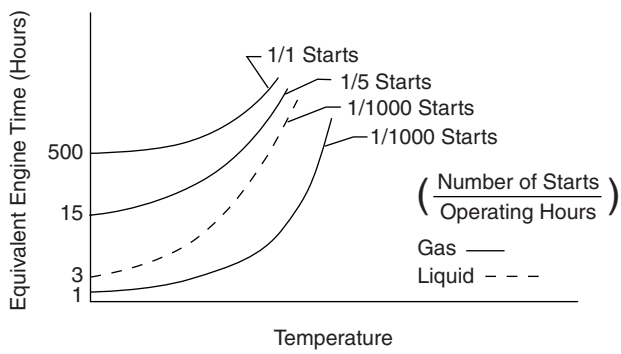


FIG. L-4 Equivalent engine time in the combustor section.

The OEMs would also provide tables of lives for each of the individual components involved. The component with the lowest number of allowed cyclic lives will determine when the unit has to be removed so that component can be changed. The others, however, may be left in until it is economically practical to remove them.

Note that OEMs whose LCA counting uses time and speed, but not temperature, are likely to be quite confident in the cooling capability of their design.

Life-cycle usage in specific applications

Life cycles are always used up faster if the application requires many stops and starts. This is indicated in Fig. L-4. The type of fuel also plays a major role in life-cycle usage, as we will see in the next section. Also, the type of turbine involved affects how many cycles or hours are used up in a normal start. Depending on the algorithms used, these figures will vary. One study estimated that the hours (at base-load conditions) used up, with a stop and start of a large industrial gas turbine (about 20,000 hp), was about 500 h. The equivalent for one of about 3000 hp was 5 h. See also Table L-2 for representation of an independent study of LC usage on an industrial gas turbine. (The algorithms used were not revealed, as is often the case with independent studies.)

Example case history 1. A gas turbine's compressor rotor developed blade tensile stresses while in operation that would quickly lead to failure. The OEM specified that a glass-bead peening process needed to be conducted all over the surface of the

TABLE L-2 Operation and Maintenance Life of an Industrial Turbine

Type and Number Load, Fuel and Starts	Type Inspection (Hours of Operation)			Expected Life (Replacement) (Hours of Operation)		
	Service[f]	Minor[g]	Major[h]	Combination Liners	First-Stage Nozzle	First-Stage Buckets
Base[a,b] natural	+	+	+	+	+	+
Natural gas, 1/1,000	4,500	9,000	28,000	30,000	60,000	100,000
Natural gas, 1/10	2,500	4,000	13,000	7,500	42,000	72,000
Distillate oil, 1/1,000	3,500	7,000	22,000	22,000	45,000	72,000
Distillate oil, 1/10	1,500	3,000	10,000	6,000	35,000	48,000
Residual, 1/1,000	2,000	4,000	5,000	3,500	20,000	28,000
Residual, 1/10	650	1,650	2,300			
System peaking[c,d]						
Natural gas, 1/10	3,000	5,000	13,000	7,500	34,000	
Natural gas, 1/5	1,000	3,000	10,000	3,800	28,000	
Distillate, 1/10	800	2,000	8,000			
Distillate, 1/5	400	1,000	7,000			
Turbine peaking[c,e]						
Natural gas, 1/5	800	4,000	12,000	2,000	12,000	
Natural gas, 1/1	200	1,000	3,000	400	9,000	
Distillate, 1/5	300	2,000	6,000			
Distillate, 1/1	100	800	2,000			

[a] 1/5 = one start per 5 operating hours.
[b] Base = normal maximum continuous load.
[c] No residual usage due to low load factor and high capital cost.
[d] System peaking = normal maximum load of short duration and daily starts.
[e] Turbine peaking = extra load resulting from operating temperature 50 to 100°F above base temperature for short durations.
[f] Service = inspection combustion parts, required downtime approximately 24 h.
[g] Minor = inspection of combustion plus turbine parts, required downtime approximately 80 h.
[h] Major = complete inspection and overhaul, required downtime approximately 160 h.
NOTE: Maintenance times are arbitrary and depend on staff availability and training, spare parts and equipment availability, and planning. Boroscope techniques can help reduce downtime.

compressor blades. This peening would insert a layer of compressive stress that would reduce the net effect of the tensile stress and the rotor could be run safely. Past a certain number of cycles, the compressive stress layer would dissipate and the entire compressor rotor would need to be overhauled again.

During one of the turbine's overhauls, the repair facility mistakenly only glass-bead peened one side of the compressor blades. Two engines failed during operation before the error was discovered. It was not known with which turbines the error had been made. The only way to determine this without doubt was to destructively test the compressor blades with X-ray diffraction. The entire fleet of engines that the overhaul shop had handled with that SB included had to be recalled.

Modifications to Operating Parameters

Parameter adjustment for extension of TBOs (time between overhauls)

Example case history 2. A gas turbine's design required that the first-stage turbine blades be removed at a time halfway between the specified times for major overhauls. The entire row had to be replaced, with one-half of an overhaul cycle of life wasted.

The OEM redesigned the blades. With the newer models, the blades were of a different size, but in terms of LCA, the most important change was to change the

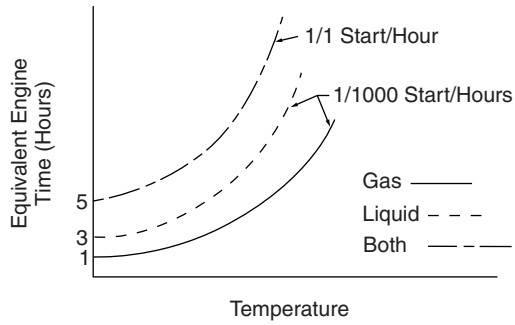


FIG. L-5 Equivalent engine hours—turbine.

material selection, which then affected the actual life cycles used up during operation by changing the stress that the blades saw in operation. The blades were under the stress endurance curve and their life was "limitless and on condition."

Parameter adjustment/machinery choice for different fuel quality

Some gas turbines run on many different kinds of fuel—gas, diesel, and residual fuel. The life of their hot-section components will be very different, depending on the type of fuel they are running. Natural gas is generally a "clean" fuel and causes the least toll in parts life in a gas turbine.

The main parameter adjusted to allow for bad fuel quality is turbine inlet temperature. It is lowered. Frequently, this prompts a choice of a different model of gas turbine or combined cycle (gas turbine/steam turbine) package. Additional features, such as water/steam injection and fuel treatment, may have to be added before life-cycle analysis indicates an economically targeted value for component lives, TBOs, and so forth. See example case history 3.

Figure L-5 shows that liquid fuel causes a higher toll on turbine life, even when an equal number of starts per operating hour is involved.

Design Assessment for Maximum TBOs

OEM design features provide end user options

Example case history 3. Analysis of gas turbine (combined cycle) design features to adapt it to run on residual fuel.

Residual fuel can be hard on the life of machinery parts. The fuel is extremely dirty and requires expensive fuel treatment to be burned, even at reduced turbine inlet temperatures (TITs). Just lowering the topping temperature in the turbine fuel control might not suffice in terms of "hot corrosion" (lowering topping temperature would reduce wear and tear on components). Choice of a different model of gas turbine is what is required to make the situation acceptable.

The ideal turbine for residual fuel application is a relatively low temperature, sturdy, preferably cast, simple design that then results in minimal maintenance. The former ABB GT13D (and its close cousin, the GT11N2), for instance, worked in applications such as the Shunde power plant, Quangdong province, China. (Note that ABB stopped making the GT13D—50 cycles—and replaced it with a GT11N2—60 cycles—and a gearbox. This arrangement costs about 2 percent in efficiency, but consolidates the OME's spares stocking requirements. Note also that ABB is part of the Alstom Corporation as of 2000.) Far greater turbine sophistication with respect to alloys and turbine inlet temperatures would be self-defeating. China, which has residual fuel, needs to run on as cheap a fuel as possible with maximum efficiency and time between overhauls.

Residual oil as a fuel is a potentially difficult prospect with inadequate design features. Corrosion, plugging, and fouling could occur quickly in certain circumstances. Higher firing temperature turbines require complex blade cooling, expensive superalloys, and substantial derating. The -13D has integrally cast blade and vane cooling passages, with relatively simple geometry (versus an aerofoil with laser-produced cooling passages) and large-flow cross sections. This provides better resistance against plugging.

The single-piece welded rotor supported by two bearings is a simple, less vibration-prone design. No through bolts are used, which is another useful maintenance feature.

This design has only one silo combustor and a solid cast design. It has one large bore-fuel nozzle, which helps avoid clogging and erosion. No air atomization is required, which means no compression airstream is required.

The nature of the burner design means that water injection is required. Typically, water injection is 1.3 times the fuel flow rate (maximum 10.5 L/s). Water injection adds 9 to 10 MW of power.

No flow divider is required in this design, so no temperature imbalance is observed in that section. This also helps cut down on maintenance and life-cycle costs of the components in this section.

The generator is driven from the cold end, which means turbine exhaust-end inspections are easier. All bearings are accessible without disassembly and no elbow conduits are required. As the generator is air cooled, no hydrogen system or hazards have to be allowed for. The cooling loop is closed and maintenance free.

The boiler, a vertical-assisted circulation, single-pressure design type, has a preheating loop. It delivers 44 kg/s of 37.5-bar steam at 475°C. Sodium phosphate (Na_2SO_4) is used for anticorrosion measures in steam treatment.

Sodium (Na), sulfur (S), and vanadium (V) content in the fuel are the major problems. Sodium is removed by mixing preheated fuel with water and demulsifier and then centrifuging. Potassium (K) impurities are removed in the same manner and at the same time as the sodium down to 0.5 ppm total (for both Na and K). The sulfur left in the fuel becomes SO_x upon combustion.

Magnesium additives combine with the vanadium to form salts that deposit onto the blade surfaces. When the turbine is shut down, the salts fall off with the drop in temperature. The remaining salts are washed off with plain water. Typically, the wash is done every 100 operating hours for heavy oil. If gas or diesel fuel are used, no wash is required.

[The limit for magnesium addition is 1105°C, as at 1120°C, magnesium oxide (MgO) solidifies to the extent it can only be chiseled off, and V_2O_5 (vanadium oxide), with its low melting point, corrodes. (Both MgO and V_2O_5 are formed from the safe additive compound after 1120°C.) The turbine inlet temperature of such units is typically maintained at 990°C.]

When the gas turbines are started, diesel fuel is used until synchronous speed and then heavy fuel is used. This helps prevent clogging. The turbines are run for 5 min on diesel when shutting down. Again, this prevents clogged nozzles and ignition problems.

The -11N2 can also handle the same rough fuel as the -13D. Peak metal temperatures, internal metallurgy, and fuel-treatment requirements are all quite similar. The single-burner design for this model can get NO_x down to 42 ppm with water injection. An EV silo combustor (several fuel nozzles) option is available if the end user has gas or diesel fuel. NO_x can then be reduced to 15 ppm when at base load on natural gas.

The preceding operational description indicates that fuel choice radically affects the required daily maintenance, overhaul, time between overhauls, and therefore effective life cycles of components. Undoubtedly, fuel choice can radically affect life-cycle assessment results.

However, cheap fuel more than offsets the capital expenditure required for fuel treatment and additives, washing the fuel, and other costs. This cost savings increases with the power capacity of a plant. Using a difference in residual oil and diesel prices of $50/ton, a 300-MW facility typically could save $22 million at 0.5 capacity factor and $36 million at 0.75 capacity factor.

Typical savings of $264 million and $432 million, respectively, are indicated over the life of the plant. It is likely that these figures could increase, as there is a global glut of residual oil.

Risk and weighting factors for design features on various OEM models

When faced with an array of factors embodied in different models that affect LCA, it is advisable to do a risk and weighting analysis. The main steps of this process are as follows.

1. List all your options. Let us say the task is to choose a model of gas turbine that will give you maximum component lives and TBOs. The options would be model numbers of gas turbines. Draw columns representing these options. Subdivide each of these columns into two, for the weighting (priority) and risk (probability).
2. List all the factors that are important to you. These may include:
   a. Maximize component lives
   b. Optimum TBOs
   c. Fuel cost savings
   d. Fuel consumption savings
   e. Avoiding fuel-treatment system requirement
   f. Avoiding water-injection system requirement
   g. Comply with/allow for:
      i. Current environmental regulations
      ii. Future environmental regulation (CO_2 tax)
   h. Minimize expertise/training level required from operators
   i. Maximize efficiency (may be linked with *c*)
   j. Design that is easy to overhaul (aimed at making the overhaul process local and/or in-house) and so forth

   Obviously, some of these factors will work in opposition to each other. List the factors at right angles to the columns in 1.
3. Give a weight (importance) to the factors in item 2. (Use a descending scale of 1 to 10.)
4. Estimate likelihood of these factors occurring, in the right column for each choice. (Use a scale of 1 to 10.)
5. Multiply the weight and the risk. Make the resulting quantity either positive or negative, depending on whether the item favors choice of that option or not.
6. Sum the products in item 5 and arrive at your selection.

Gas-Turbine Deterioration and Diagnosis

Performance recovery

After some hours of operation, the gas turbine(s) in a train may be observed to deteriorate in terms of mechanical condition and/or performance (flow, pressure) deterioration. The deterioration is detrimental to component lives and should be corrected for in the short and long term. Generally, faults with these two aspects of operation can be diagnosed with condition monitoring (vibration analysis included) and performance analysis (PA), respectively. *See* a discussion of vibration

analysis under Condition Monitoring earlier in this book. Performance analysis will be dealt with below.

Metallurgical recovery, refurbishment (*see* Metallurgy)

Steam-Turbine Deterioration and Diagnosis

Performance recovery

After some hours of operation, the gas turbine(s) in a train may be observed to deteriorate in terms of mechanical condition and/or performance (flow, pressure) deterioration. The deterioration is detrimental to component lives and should be corrected for in the short and long term. Generally, faults with these two aspects of operation can be diagnosed with condition monitoring (vibration analysis included) and performance analysis (PA), respectively. *See* a discussion of vibration analysis under Condition Monitoring earlier in this book. Performance analysis will be dealt with below.

Metallurgical recovery, refurbishment (*see* Metallurgy)

Performance Analysis (PA) with Gas Path Parameters

Basically, PA is a check of the fluid path of a piece of turbomachinery (gas turbine, compressor, and so forth) to determine that it is doing what it was designed and bought to do. To perform this check, the pressures, temperatures, and flow along the machinery's gas path may be used to compare its performance curves of pressure versus flow, power developed/required versus flow or other performance parameters, efficiency versus power developed/required, and so forth. The readings may then be computed for the operator to either deduce what corrective action is required or read off the required action on an automated system display.

Performance analysis systems occur in three main formats:

1. Systems that have a calculation module based on actual instrumentation readings. Such systems may cost from about $15,000 up for the first unit, with discounts for additional units of the same model.
2. Systems that have a calculation module based on actual instrumentation readings, as well as a calculation module that uses predictive flow formulas theory and compares the two for better quality information and indicates which instrumentation may be malfunctioning. Such systems may cost from about $60,000 up for the first unit, with discounts for additional units of the same model.
3. Systems that perform the functions of item 2 and are part of an overall on-line (frequently real-time as well) health-monitoring system that incorporates vibration monitoring and other monitoring. Such a system uses artificial intelligence techniques to attempt automated problem solving. Such a system is generally extremely expensive. Typically, these systems cost from about $100,000 per machine train (no discounts for similar trains sometimes) to $250,000 per train.

They may frequently also be "overkill" for an application. Typically, they take about 600 readings/signals as inputs to arrive at their conclusions. Frequently, their specifications also demand requirements that eliminate the competition. Their manufacturers may claim their system's use absolves the operator of using his or her own system knowledge or reasoning by providing "ready" answers. Sometimes this works for certain problems if they were programmed into the artificial intelligence logic. The customers may get the best value for their money from comprehensive vibration analysis and a system that incorporates the capabilities of the system in category 2.

In this section we will discuss a generic system in category 2.* This generally provides the best economic return in terms of dollars saved/dollars of capital cost for a PA system and therefore is one of the most important items in this course.

Consider the following:

- Aims (goals) of a PA system
- Cost, operational benefits, and return on investment (ROI) of a PA system
- How such a system works, so you can compare it against other options and ask manufacturers the right questions
- Advantages and additional applications of PA systems
- Use of a PA system to extend TBOs and reduce repair costs

Aims of a PA system:

1. Check detailed performance for different modules
2. Predict performance through the load range
3. Check machine in normal operation, not just when it has just been cleaned or newly overhauled
4. Check performance of combined cycle (CC) or other complex cycles
5. Self-check and help troubleshoot
6. System must be interactive and simple to run
7. Check cost for different configurations
8. Check cost differential for different operational conditions

When successful, a PA system enhances:

1. Predictive maintenance
2. Reduced performance test cost
3. Improved operations cost prediction
4. Improved system efficiency

In a commercially available PA system, which will be used for illustrative purposes, while setting up reference data, the following parameters were varied:

1. Barometric pressure
2. Ambient temperature
3. Axial compressor flow degradation
4. Axial compressor efficiency degradation
5. Axial turbine efficiency degradation
6. Anti-icing system operation

The diagnostic module input data are (see Fig. L-6):

1. Barometric pressure
2. Ambient dry-bulb temperature
3. Ambient wet-bulb temperature
4. Fuel consumption
5. Compressor delivery pressure

---

* Source: For this PA system type and the included example case histories 4 and 5, the information source is Liburdi Engineering, Canada. Material adapted with permission.

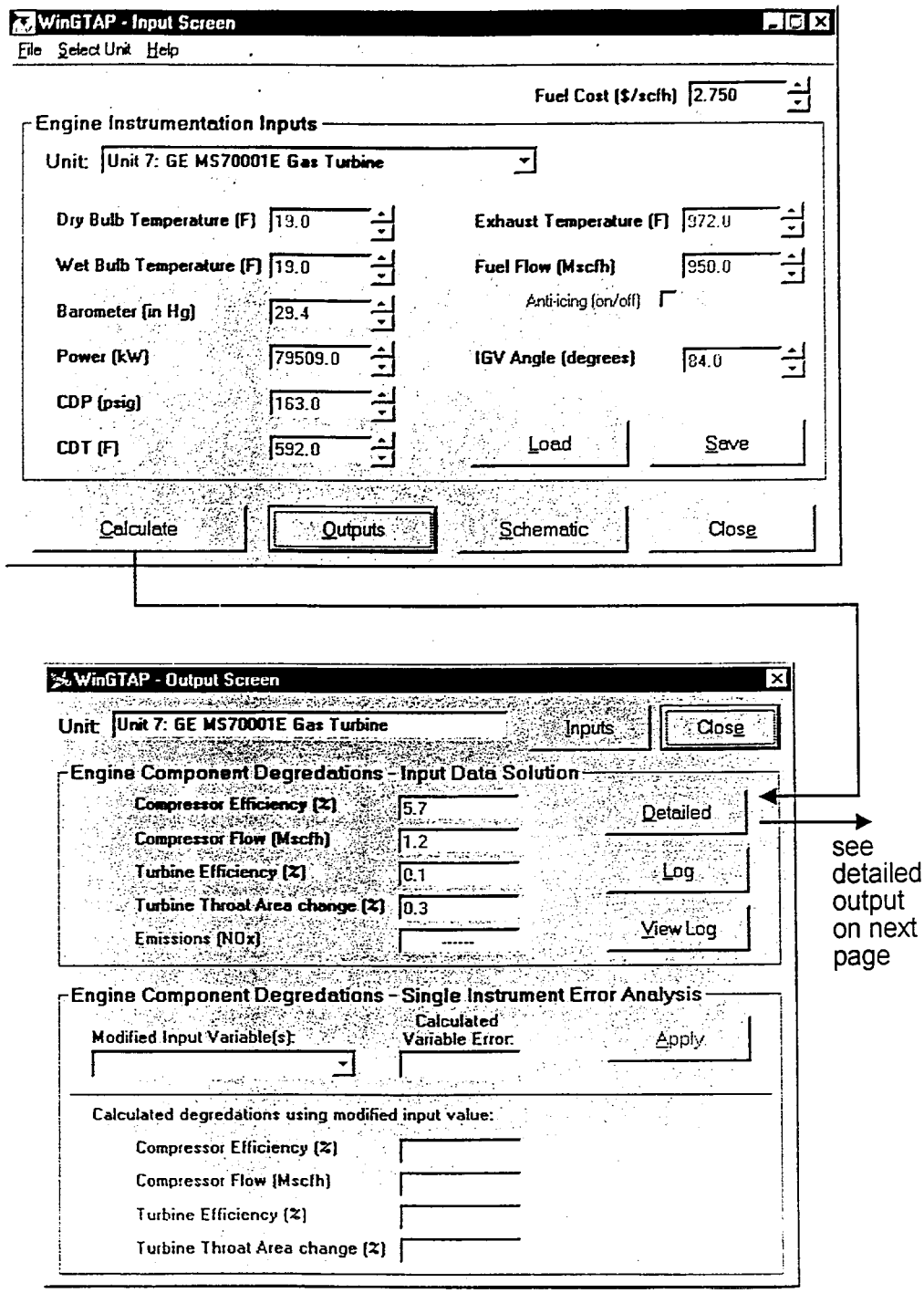


FIG. L-6 Screen displays from PA system (GE Frame 7 engine). (Source: Liburdi Engineering.)

6. Compressor delivery temperature
7. Gross power
8. Exhaust gas temperature
9. Inlet guide vane settings

Any measurement errors are usually with items 4, 5, 7, and 8.

Gas path analysis steps (from the preceding instrument data) are:

1. Calculate compressor inlet flow
2. Calculate TIT from turbine swallowing capacity
3. Calculate turbine efficiency degradation
4. Calculate compressor efficiency
5. Calculate clean compressor flow and efficiency, then calculate flow and efficiency degradation

Experience indicates that the following ranges usually apply:

1. For compressor flow degradation, <0 to >8 percent
2. For compressor efficiency degradation, <0 to >6 percent
3. For turbine efficiency degradation, <0 to >3 percent

An appropriately designed program will calculate items 1 to 3.

In this PA system's predictive module, predictive values were calculated (fuel flows are interpolated).

Fuel consumption was calculated for all combinations of:

1. 3 barometric pressures
2. 5 ambient temperatures
3. 8 degradation combinations
4. 5 to 11 power levels, which gives 600 to 1320 data points

A second set is required if you have an anti-icing system.

Predictive programs are useful for CC, cogeneration, and complex cycles.

This set of grid values can be included in automatic, on-line, or real-time systems.

A practical use of modeling is in the sale of product (gas, power) versus fuel consumption.

When considering PA systems, look for the following features that offer considerable advantages over others:

- Analysis is aerodynamic, not just a heat balance with the machine represented by a black box
- Predicts stage efficiency, pressure ratio, exit angle, gas properties, and swallowing capacity
- Subroutine handles the following losses: profile, incidence, trailing edge thickness, tip clearance, shock, end wall, and secondary
- Cooling air, flame flare, and similar flow interruptions treated as blockages to help accurate modeling

Advantages and uses of having these features:

- Application in sales based on fuel consumption measured more accurately than with normal fuel flowmeters (see Fig. L-7)
- System can be set up to determine optimum combination in a machinery train for a certain load level
- Cooling-flow modifications can be analyzed and then made to prevent problems such as disc cracking
- Similarly, blade cooling modifications can be made

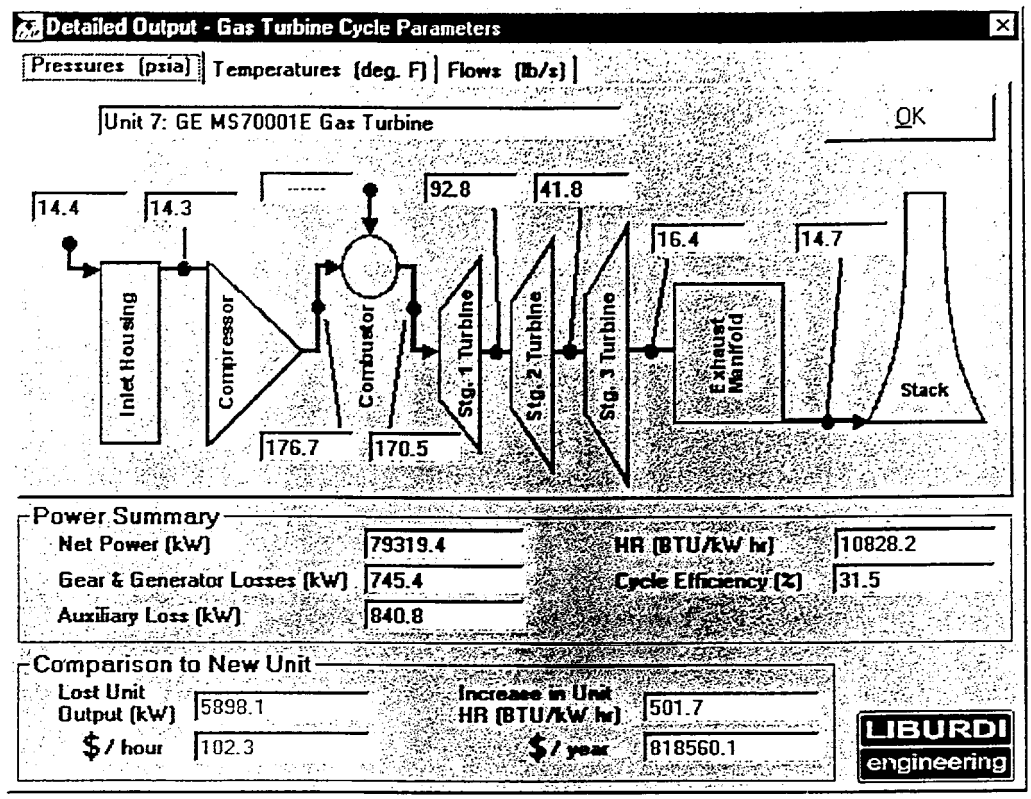


FIG. L-7 Screen display in working PA system (GE Frame 7). (Source: Liburdi Engineering.)

- Analyzing steam- and water-injection effects (reduced maintenance, power increases, NO_x reduction)
- Analysis of gasifiers (GT operated to drive the compressor only) to supply pressurized gas

In one case a W191 was used to provide 1,000,000 lb/h at 10 psig back pressure and below 1000°F. Being able to analyze each stage helped this development of modified running of a gas turbine and aided predictive emissions monitoring.

Note that it should be simple to correct for data inputs that can cause errors due to factors such as:

1. Compressor delivery temperature being affected by heat transfer in the combustor. Solution: Calibrate each engine.
2. Inlet and exhaust pressure loss errors. Solution: Measure periodically.
3. Fuel composition. Solution: Get accurate samples.
4. Modifications [e.g., inlet guide vanes (IGVs)] are added. Solution: Model into solution.
5. Atmospheric condition errors. Solution: Do not forget barometer and relative humidity readings daily (if not incorporated in real time into program).

Example case history 4. Extending TBOs of gas turbines by preventing premature component failure in a GE Frame 5 (see Figs. L-8 to L-10).

Consider that turbine disc lives in some industrial GTs are limited to <100,000 h. This is due to high-temperature creep cracks that are formed when the engine is operated at full load. This, in turn, causes costly repairs and also

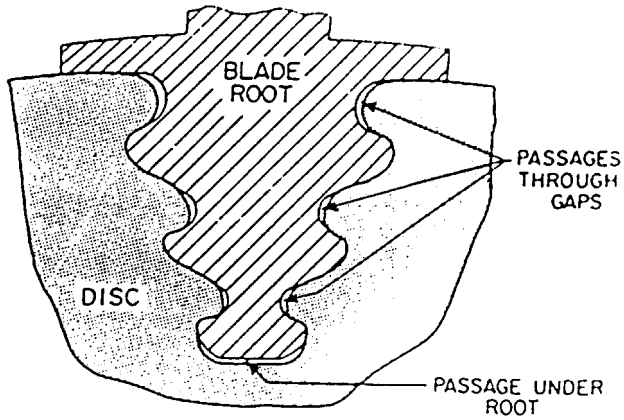


FIG. L-8 Typical blade root-disc serration configuration. (Source: Liburdi Engineering.)

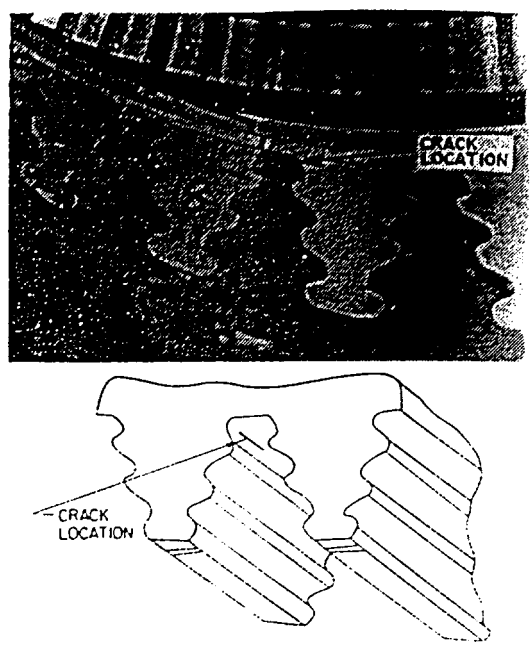


FIG. L-9 The cracks generally start inside on the bottom radius and are very difficult to detect in inspection. (Source: Liburdi Engineering.)

disastrous failures. This can be avoided and disc lives extended to >100,000 h by using performance-monitoring software to analyze changes to the disc cooling.

Note that before this was done, changing the disc material was tried but this did not work. The cracks persisted.

The cracks were at the bottom of the fir tree and difficult to see. Note the following details from the figures:

- Where cracks occurred
- Cracks along grain boundaries
- Root-disc gap configuration
- Compressor air and air hot-gas air paths are located at each disc root. Hot air accumulates where it shouldn't.

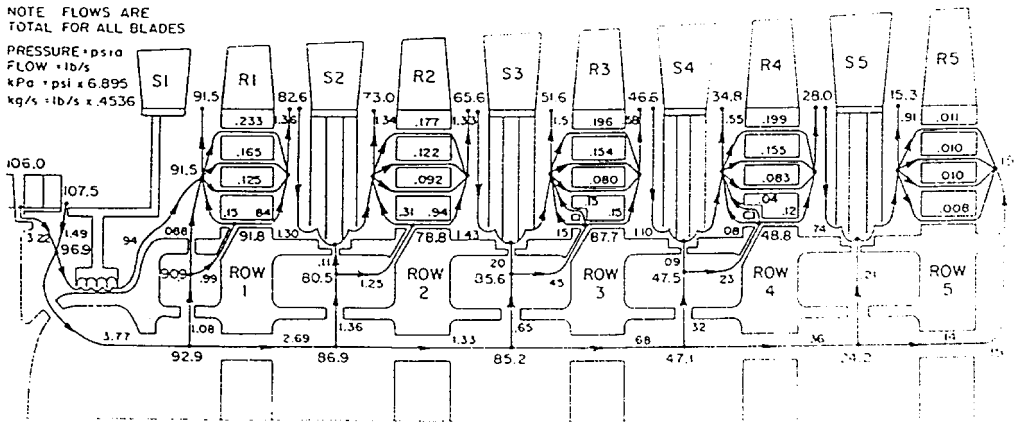


FIG. L-10 "As found" turbine disc cooling flows. (Source: Liburdi Engineering.)

TABLE L-3 Compare Measured and Predicted Values of Engine Parameters (Source: Liburdi Engineering)

	Predicted	Actual
Increase combustor temperature	6.7–7.2	6.7
Increase exhaust temperature	2.2–3.9	1.1
Decrease compressor exit pressure, lb/in^2 (kPa)	0.9 (6.2)	0.7 (4.8)
Increase in fuel flow, lb/s (kg/s)	0.011–0.019 (0.005–0.009)	0.016 (0.007)

- Solution: Feed cool air through diaphragms. No rotating components affected, just the diaphragms. Rows 3 and 4 had compressor delivery air, row 5 had intermediate stage compressor air from bleed valve.
- Diagram of cooling air distribution

(See Table L-3.) Net effect on performance: negligible

- 70-hp (52-kW) increase
- 0.05 decrease in thermal efficiency
- 12°F (7°C) increase in TIT
- 1°C (2°F) increase in exhaust temperature
- 0.75-lb/in^2 (5.2 kPa) decrease in combustor shell

Example case history 5. Power augmentation for a gas turbine in cogeneration service using steam injection. Operation of this system works best when:

- Steam is injected only when a certain power is reached.
- All excess steam is injected and then the control system is allowed to vary IGVs and fuel flow.
- Keep steam lines hot with a small amount of condensate even when steam is not running.

Summary: 30 percent more power is possible when injecting steam equivalent to 7.5 percent of compressor inlet flow. *Note:* NO_x levels are down from 83 to 12 ppm.

Vibration Analysis and Its Role in Life Usage (*see also* Condition Monitoring)

Vibration is a key factor in how long a machine component lasts. The extent to which vibration occurs, and its cause, can be measured by vibration analysis. This is covered in the section on Condition Monitoring.

Note, however, that vibration analysis and performance analysis may be linked in many instances. For example, a cracked combustion liner results in a change in TIT and PA calculations. As the cracked metal disturbs the airflow and is set into a vibration mode of its own, vibration sensors pick up indication of the cracked liner. Depending on the accuracy of the vibration probes, the sensors may pick up the problem before monitoring of gas path parameters.

Vibration analysis is the best detector of problems with components not directly in contact with the gas path, such as bearings, accessory drives, and so forth.

Experienced engineers can do what an expert system does, i.e., arrive at diagnosis of a problem by using indicators from the vibration analysis probes and transducers that are monitoring the gas path.

Example case history 6. The following observations on a compressor could confirm the existence of fouling in the compressor.

Vibration: Rises
PA system data: P_2/P_1 drops, T_2/T_1 rises, compressor efficiency drops
Corrective action: The compressor is washed, and performance recovery is monitored.

For a compressor in surge:

Vibration: Fluctuates, often wildly
PA system data: P_2/P_1 varies, T_2/T_1 does not change, compressor efficiency drops
Other data: Bleed chamber pressure fluctuates, temperature differential across the bearing may be observed to increase, bearing pressure will rise

However, the vibration and the PA system data would be enough to diagnose the high probability of surge.

Example case history 7. For a damaged compressor blade:

Vibration: Rises
PA system data: P_2/P_1 drops, T_2/T_1 rises, compressor efficiency drops
Other data: Bleed chamber pressure fluctuates

Again, the vibration and the PA system data would be enough to diagnose the high probability of surge.

For a compressor bearing failure:

Vibration: Rises
PA system data: No change
Other data: Temperature differential across the bearing rises, bearing pressure drops, bleed chamber pressure stays constant

Note that just the vibration reading should be enough to detect incipient bearing failure or bearing failure, even though not supported (even though not negated) by PA data.

These four cases help illustrate that vibration readings and PA analysis should solve most serious problems. Whether or not the other data back up these two systems, it is not essential to these diagnoses. Very often, marketers of expensive expert systems will try to insist these additional data are vital. While the data may be useful for specific problems, they may not be worth the extra initial capital outlay, as well as cost of operator/engineer training data and/or consultants' fees to interpret the data. (As an example, the fee for consultants to interpret data turned

out by the expert system installed on the Canadian Air Force's small F-18 fleet's F404 engines was about $1 million in 1987. Bear in mind that the expert system could be called justifiable on a critical flight engine, despite triple redundancy in its control systems.)

Codes and Specifications

Specifications for PA systems and intelligent expert on-line systems, real time or otherwise, are as plentiful as the number of system designers/manufacturers. The more expensive they are, the more they are likely to be complex, with an intentional tendency to exclude competition.

Codes for enclosures, such as control panels, computers, controls, valves, and so forth, are unchanged from the codes specified in API, ASME, and so forth, for specifications with respect to safety considerations. *See* Some Commonly Used Specifications, Codes, Standards, and Texts.

Operational Optimization Audits

Audits are conducted to assess the efficiency and validity of a plant, a process or any part thereof at a time during the life of that unit. Audits can result in major, expensive modifications that have a good ROI, such as PA systems. When PA systems are retrofit, this is often the result of an audit, broad or limited in scope.

The word audit carries with it the connotation of time unwillingly but dutifully spent on a necessary evil. The audit team and those who provide them with information expect boredom, witch hunting, paper trails, and, worst of all, lost revenue time. The latter factor may not be the case, depending on the circumstances. With careful planning, the time can be used to optimize design, maintenance, and operational conditions to maximize profit margins. Stricter environmental conditions sometimes make an audit a requirement, and, in some cases, suspended operations. The time should be viewed as an opportunity, as environmentally prompted design changes may herald other significant maintenance time or operational efficiency gains.

There are two kinds of audit teams: internal (in-house) and external. On occasion, the team consists of both of these groups. The audit team is trained to look for areas of material breakdown, safety hazards that have arisen as a consequence of deterioration, and items that require change because they fall under recently enacted legislation.

Note that for circumstances where operational conditions are changing, for instance in a combined oil and gas field where relative volumes of gas, oil, and seawater, as well as molecular weights are changing, the two audit types may occur simultaneously if retrofit, optimization, or redesign become an issue.

Preparation for an audit

1. Collect the data.
   a. Sources include maintenance and production management information systems (MIS), automated and manual, current relevant legislation, and relevant labor contracts.

      Comprehensive MIS can help track recurring items that indicate required specification, design, or maintenance practice changes, such as wear plates instead of wear rings, an additional vibration probe–monitoring position, and additional fluid moved through a seal buffer system. Legislation can dictate abandonment of long used cleaning fluids and procedures and redesign of the exhaust system off a plasma spraying booth. Labor contracts, particularly in a union environment, can dictate similar changes.

      When external changes, legal, labor, or otherwise, dictate a major change

in procedure and/or operating and maintenance procedures, an audit should be considered to cover the scope of all affected systems.

b. Maintenance and production personnels' "must have" and "nice to have" lists and equipment literature.

The status of these items changes through the life of a facility. Where wear rings might have sufficed in abrasive service, changes in process flow content may make wear plates necessary.

An audit, then, is something personnel should plan for and collect data for continuously between audits.

c. Latest updates of relevant standards and practices.

d. Format of paperwork to be used.

e. Description of relevant repair procedures, contractor lists, and spare parts brokers if relevant. Questions asked here should include:

- What is the expected remnant life of the production field in question?
- What are the OEM's service intentions with respect to the models used in production?
- What are spares inventories?
- What are inventories of official scrap of critical components?
- Do new repair technologies make salvage of previously scrapped components possible?
- What impact do the answers to these questions have on the profitable life of the existing plant? On the profitability of planned expansions?
- On the design of planned expansions?
- On the choice of OEMs and system design for planned expansions?

f. Quotes on retrofit procedures and installations.

Contractors should also have indicated their completion times for retrofits for minimum impact on shutdown times. Consider penalty clauses, cost plus clauses, and other relevant expense items.

2. Planning process.
   a. Get updates of all information in step 1.
   b. Identify departments that should have audit input.
   c. Identify the extent of input required from different departments.
   d. For each department identify primary and secondary contacts.
   e. Formulate a time-line program. Work backward from the required completion date of the audit.
   f. Review the time-line schedule with the team.
   g. Decide on the interface of audit/regular operations/ongoing maintenance/shutdown.
   h. Finalize the time line. (Time line should be flexible at all times.)
   i. Identify and build special tooling/gauges/instrumentation.
   j. Identify any special heavy lifts required. Arrange all details of safety equipment required. Relevant questions may include:
      - For critical rotor balancing procedures, will specific arbors make fewer operations possible?
      - Will tolerance tightening on specific balance tooling decrease rotor imbalance and increase TBOs?
      - Will digital versus analog readouts affect operational efficiency? TBO?
   k. Identify the tolerance changes required by specific applications.
   l. Identify and collate information learned from previous equipment failures.

Recommendations for conducting the audit

1. Using the information collected during the preparation phases, formulate the checklists to be used during the audit. The lists are only to be used as guides, however, as totally unforeseen circumstances might come to light.

2. Members of the audit team should include representatives from all departments that may be affected by its outcome.
3. Provide audit team members with appropriate training conducted by an external objective party. This party should work in concert with plant personnel and OEMs but not be focused on any specific party's interests.
4. An objective party, preferably the trainer in item 3, should be present during the audit and during analysis of its findings.
5. Arrange for relevant photographic records to be made and filed during the audit for future analysis.

Summary

For life-cycle analysis to be truly successful, it needs to be linked with everyday operations and maintenance at a plant, as well as with periodic audit and shutdown activities. The amount of equipment and instrumentation used for LCA should be tailored strictly to just what is necessary. A great many expensive "bells and whistles" (features) may be unnecessary and just produce mounds of additional data that the customer has to manage.

References and Additional Reading

1. Soares, C. M., "Aspects of Aircraft Gas Turbine Engine Monitoring Systems Experience as Applicable to Ground Based Gas Turbine Engines," *TMC*, 1988.
2. Various service bulletins (various OEMs) used as a guide only.
3. Boyce, turbomachinery notes, 1979.
4. Soares, C. M., Failure analysis reports, C-18 (250 series) Allison engines, 1985.
5. Soares, C. M., Fleet life extension study reports (T55 Avco Lycoming), 1985.
6. Soares, C. M., "Residual Fuel Makes Inroads into Chinese Market," *Modern Power Systems*, May 1997.
7. Soares, C. M., "New Turbines for Old," *Asian Electricity*, 1997.
8. Repair technology literature, various OEMs.
9. Working system data/results from WinGTap on Anchorage power station, Liburdi Engineering.
10. Pistor, "A Generalised Gas Turbine Performance Prediction Method through PC Based Software," *IAGT*, 1997.
11. Little, Wilson, and Liburdi, "Extension of Gas Turbine Disc Life by Retrofitting a Supplemental Cooling System," *IGTI*, 1985.
12. Little and Rives, "Steam Injection of Frame 5 Gas Turbines for Power Augmentation in Cogeneration Service," *IGTI*, 1988.
13. Little, Nikkels, and Smithson, "Incremental Fuel Cost Prediction for a Gas Turbine Combined Cycle Utility," *IGTI*, 1989.
14. Soares, C. M., "Vibration Analysis: Separating the Elements of Machinery, Process and Personnel," *TMC*, 1994.

Liquid Eliminators (*see* Separators)

Liquid Natural Gas (LNG)*

An LNG processing system requires filters and other appropriate accessories to maintain appropriate delivery properties. A basic system is shown in Fig. L-11. This is an area where constant research is being conducted to minimize vessel size and weight. Computational fluid dynamics (CFD) and specialized probes assist in this research and can, when necessary, also be used in operational functions to avoid plant shutdowns (see Figs. L-12 through L-14).

---

* Source: Peerless, USA.

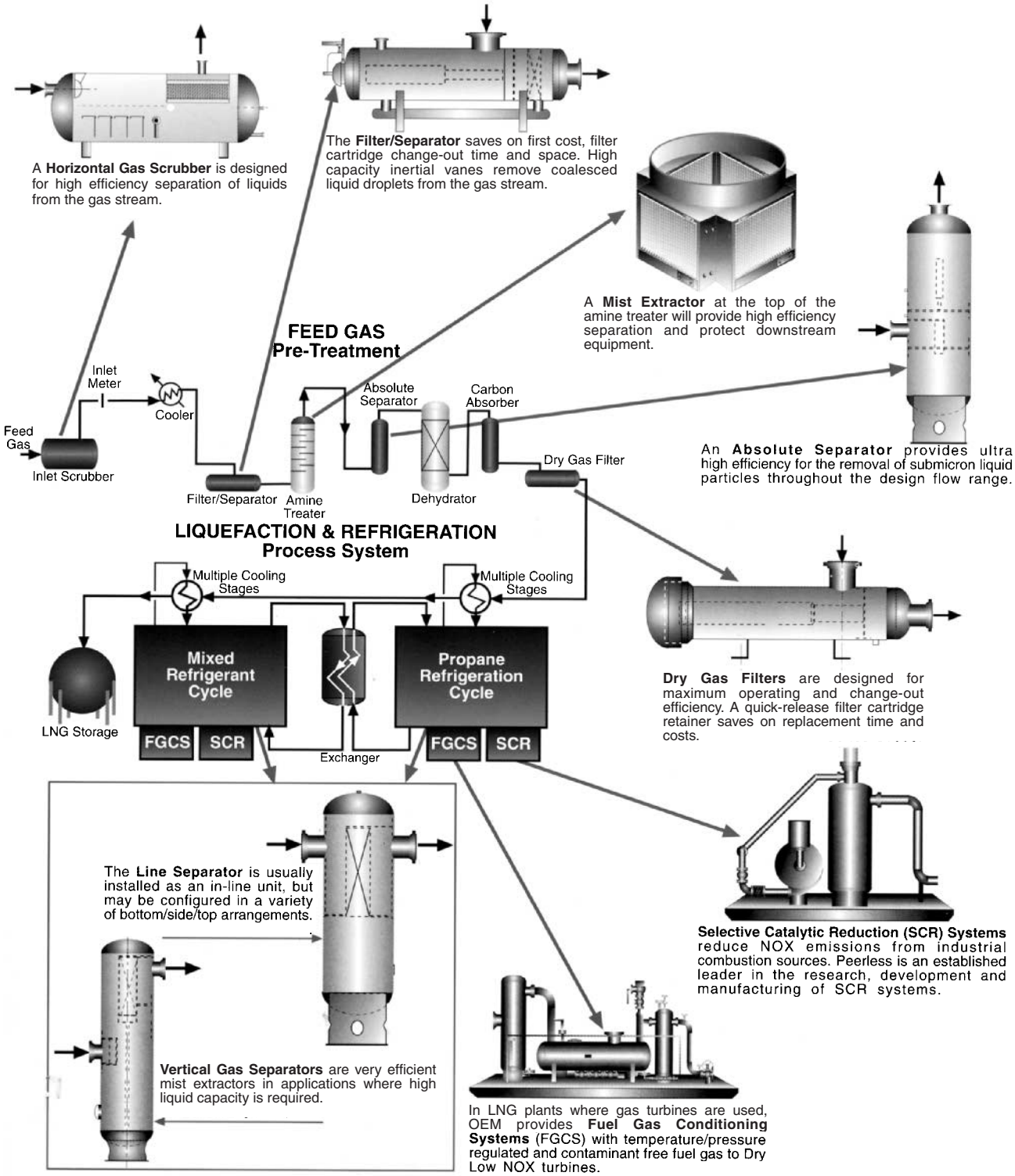


FIG. L-11 An LNG feed, liquefaction, and refrigeration process system. (Source: Peerless.)

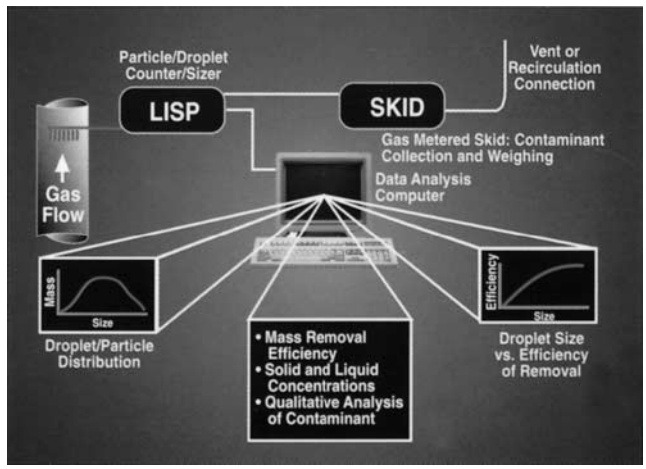


FIG. L-12 Diagram of the Laser Isokinetic Sampling Probe (LISP^{SM}) field test setup and field analysis equipment. (Source: Peerless.)

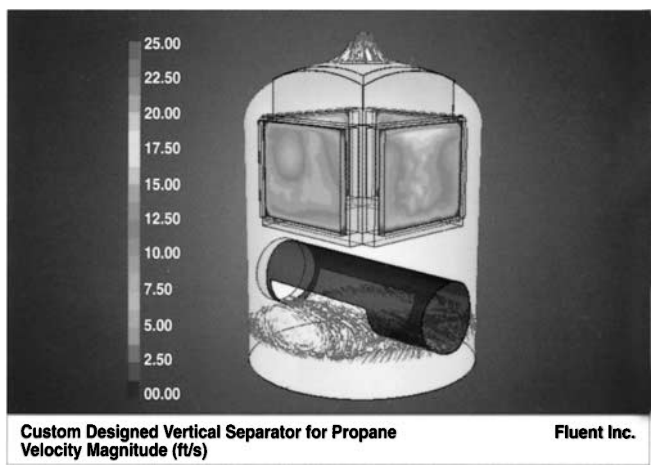


FIG. L-13 Proprietary Sizing^{TM} reduces the vessel by several sizes. Computational fluid dynamics technology contributes to the application solution and ensures all design specifications are met. (Source: Peerless.)

Typical Liquefied Natural Gas Process

Computational fluid dynamics (CFD)

Sophisticated computer models help to reduce the size of separator vessels and ensure that liquid/vapor separation is achieved to specification. The CFD flow model pictured in Fig L-13 depicts the final design of a vertical gas separator for an LNG facility. This graphic provides the engineer with visual confirmation of gas flow paths and that the separator face velocities meet established design criteria.

CFD models use actual vapor properties such as those for propane, ethane, or any of the various mixed refrigerants to determine separation performance and capacity.

In-line testing without plant shutdown

A new field sampling tool for pressurized gas streams, the Laser Isokinetic Sampling Probe (LISP^{SM}) was developed, custom-designed, and built to specifications. It collects

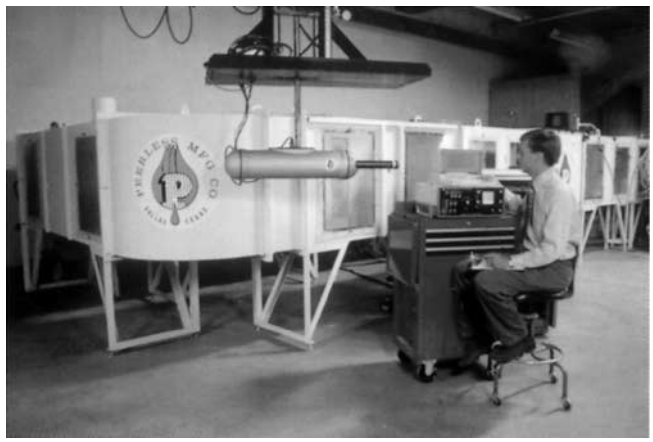


FIG. L-14 An R&D lab is equipped with a computerized forward scattering spectrometer probe (FSSP). The FSSP uses precision optics and a laser to measure liquid droplets down to submicron diameters. This FSSP is being inspected before being placed into the wind tunnel. (Source: Peerless.)

and weighs entrained liquids and solids both up- and downstream of separators or filters at very high system operating pressures.

Thus, samples can be taken of liquids and solids in their pressurized state. And because of the high degree of sensitivity demanded by the LISP, meticulous measurements can be made of particles as small as 0.3 microns in diameter. The result is the most accurate and reliable pressurized, in-line, field sampling of LNG processes without a plant shutdown.

Lubrication*

Lubrication is primarily concerned with reducing resistance between two surfaces moving with relative motion. Any substance introduced on or between the surfaces to change the resistance due to friction is called a *lubricant*. In addition to reducing friction, a lubricant removes excess heat, cleans microscopic wear particles from surfaces, coats surfaces to prevent rust and corrosion, and seals closures to prevent dust and moisture from entering.

The choice of the proper lubricant not only is important to manufacturers in order to enable them to meet their guarantees for performance and reliability but is, of course, of the utmost importance to users of the equipment in keeping their maintenance costs to a minimum and safeguarding machinery against abnormal wear, corrosion, and the effects of contamination. When choosing a lubricant, conditions such as operating speed, load conditions, method of sealing, temperature range, moisture condition, bearing design, and quantity of lubricant all affect the final choice.

It is generally recognized that a specification giving only physical and chemical properties does not guarantee satisfactory performance of any particular lubricant. Manufacturers and users, therefore, must rely on the experience, integrity, and

* Source: Demag Delaval, USA.

reputation of the lubricant supplier and on the record of satisfactory past performance of the particular type of lubricant offered for a given purpose.

The lubricant should be a first-grade branded product that has previously been used and proved to be satisfactory for the continuous lubrication of similar equipment in the same service. Such experience should have proved the lubricant to be satisfactory, particularly with respect to foaming, rusting, sludging, and separation for water and other impurities.

The brand of lubricant decided upon should be continued in use and should not be changed without compelling reason.

Lubrication Methods

Either splash lubrication or forced-feed oil lubrication is commonly used for rotating machinery such as turbines, pumps, compressors, reduction gears, and worm gears. Splash lubrication is used for relatively slow-speed machinery, while high-speed machinery always requires forced-feed lubrication.

The usual form of splash lubrication employs oil rings. In this arrangement a loose ring rides freely on the journal and dips into a sump in the bearing bracket containing oil. The ring rotates because of its contact with the journal, but at a slower speed. The oil adheres to the ring until it reaches the top of the journal, when it flows onto the shaft.

Ring oiling for small machines is used predominantly when the additional cost of a pumping system cannot be justified. The system enjoys the advantage of self-containment, needing no external motivation for its performance. Cooling coils are sometimes added when the sump temperature may become excessive.

The fully forced, or direct-pressure, system, in which the oil is forced into the bearing under pressure, is used in the majority of large circulation systems. Force feeding increases considerably the flow of lubricant to the bearing, thereby removing the heat generated by the bearing. This system is most reliable in high-speed operations with considerable load. (See Figs. L-15 and L-16.)

Grease lubrication is principally used for ball bearings and roller bearings since the housing design and maintenance are simpler than for oil lubrication. As compared with an oil system, there are virtually no leakage problems and no need for a circulation system.

The data in Table L-4 give desirable viscosities and other specifications for oils. The data in Table L-5 give grease recommendations for various applications.

Oil Characteristics

A lubricating oil should be a petroleum oil of high quality having guaranteed uniformity, high lubricating qualities, and adequate protection against rust and oxidation. It should be free from acids, alkalies, asphaltum, pitch, soap, resin, and water. The oil must not contain any solid matter or materials that will injure the oil itself or the parts it contacts or impair its lubricating properties. Lubricating oil should not foam, form permanent emulsions, oxidize rapidly, or form sludge. It may contain additives or inhibitors if their use supplements but does not adversely affect the desirable properties and characteristics of an oil.

Horsepower losses, bearing exit temperatures, and oil-film thicknesses decrease with lower viscosity values and increase with higher viscosity values.

When cold starting is important or a product has ring-oiled bearings, a lubricating oil with a high viscosity index should be used. A high viscosity index means that the rate of change of viscosity of an oil with change of temperature is small.

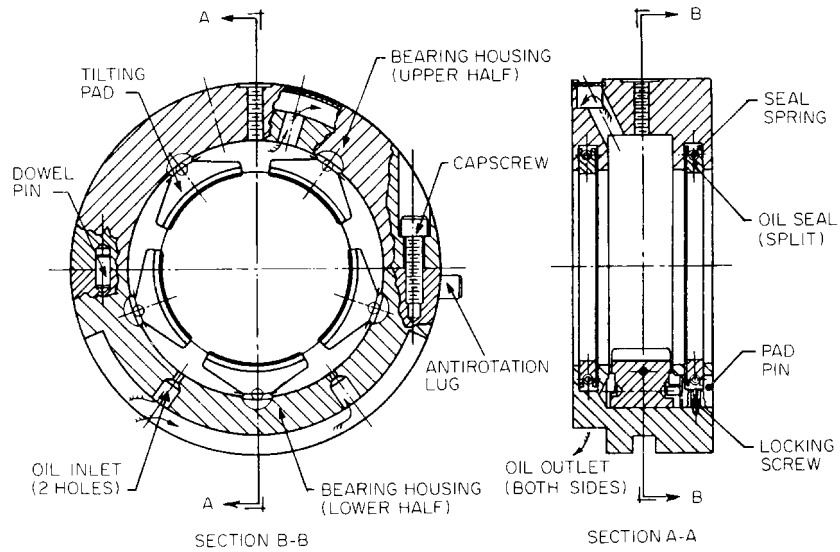


FIG. L-15 Typical tilting-pad bearing. (Source: Demag Delaval.)

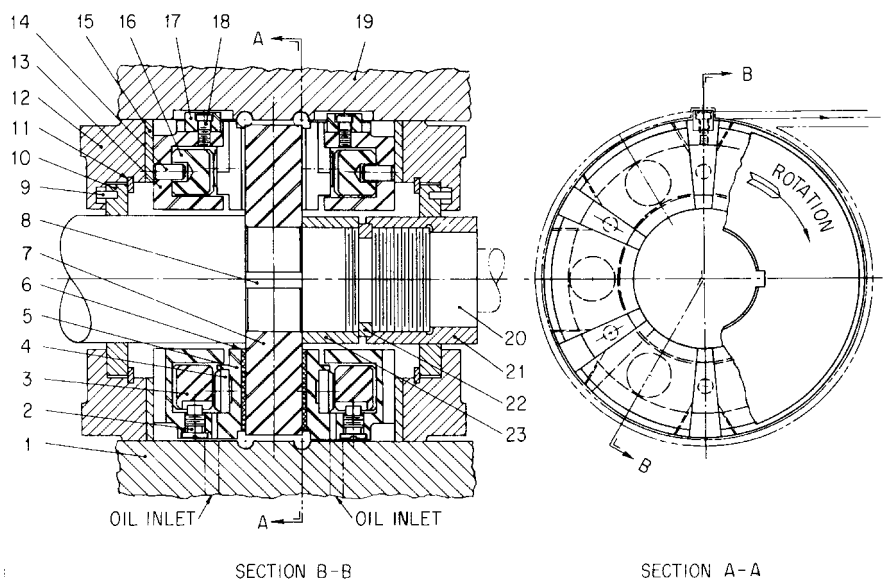


FIG. L-16 Section of tilting-pad thrust bearing. (1) Bearing bracket. (2) Leveling-plate set-screw. (3) Upper leveling plate. (4) Shoe support. (5) Shoe. (6) Shoe babbitt (4, 5, and 6 assembled as a unit). (7) Collar. (8) Key. (9) Pin. (10) Oil guard. (11) Snap ring. (12) Thrust-bearing ring. (13) Base ring (in halves). (14) Leveling-plate dowel. (15) Shim. (16) Lower leveling plate. (17) Base-ring key. (18) Base-ring key screw. (19) Bearing-bracket cap. (20) Shaft. (21) Outer check nut. (22) Retaining ring. (23) Inner check nut. (Source: Demag Delaval.)

Grease Characteristics

Greases should be high-grade, high-temperature lubricants suitable for application by hand, pressure gun, or hand compression cup.

Greases should remain in the solid state at operating temperatures. Grease components should not separate on standing or when heated below their dropping

TABLE L-4 Oil Selection (Source: Demag Delaval)

Product					Type of Oil	Viscosity, SSU ASTM D88			Oil Temperature, °F	
						100°F	130°F	210°F		
						Minimum Maximum	Minimum Maximum	Minimum Maximum	Minimum Operating	Normal to Bearings
Marine propulsion units: turbine-driven					Turbine*	490 625	220 270	62 minimum	90	110 130
Ship's service turbine-generator sets marine auxiliaries: direct or gear drive					Turbine	375 525	180 230	54 minimum	90	110 130
Marine propulsion units: diesel-driven					EP, R&O	630 770	270 320	69 minimum	90	110 130
Centrifugal pumps	Direct drive	Ring-oiled bearings	With water cooling		Turbine	250 350	120 155	47 minimum	. . .	140 160
			Without water cooling	Liquids up to 130°F	Turbine	250 350	120 155	47 minimum	. . .	140 160
				Liquids 131°F and above	Turbine	375 525	180 230	54 minimum	. . . . . .	140 180
			Forced circulation		Turbine	250 350	120 155	47 minimum	. . .	110 120
		Forced feed			Turbine	140 180	85 105	42 minimum	90	110 120
	Gear drive	Forced feed			Turbine	250 350	120 155	47 minimum	90	110 120
Gears	Helical				Turbine	140 180	85 105	42 minimum	90	110 120
	Worm				See Sec. 10, Table P-11					
Turbines	Direct drive	Ring-oiled bearings	With water cooling		Turbine	250 350	120 155	47 minimum	. . .	140 160
			Without water cooling		Turbine	375 525	180 230	54 minimum	. . .	140 180
			Forced circulation		Turbine	250 350	120 155	47 minimum	. . .	110 120
		Forced feed			Turbine	140 180	85 105	42 minimum	90	110 120
	Gear drive	Forced feed			Turbine	250 350	120 155	47 minimum	90	110 120
Centrifugal compressors	Direct drive				Turbine†	140 180	85 105	42 minimum	90	110 120
	Gear drive				Turbine†	140 180	85 105	42 minimum	90	110 120
Turbochargers	Lubricated independently				Turbine	375 525	180 230	54 minimum	. . .	120 160
	Lubricated by engine				See engine manufacturer's specifications	SAE 20 or 30 preferred			. . .	120 160

* Approximately 300 lb/in Ryder gear machine test. † Compressors with oil seals, 190 minimum aniline point.

TABLE L-5 Grease Selection (Source: Demag Delaval)

Component		Type	NLGI No.	Worked Penetration	Drop Point	Corrosion Test
				Minimum ASTM D 217–68	Minimum ASTM D566–76	
Ball bearings, roller bearings, oscillating or sliding plain bearings, sliding pedestal supports*		Sodium, lithium, or sodium-calcium soap base	2	265–295	350	Pass federal test method Standard No. 791 Method 5309.2
Governor valve lifting gear	Steam temperature 600°F maximum	Sodium or lithium soap base	2	265–295	350	Pass federal test method Standard No. 791 Method 5309.2
	Steam temperature 600–825°F	Nonsoap base	1 or 2	265–340	500	Pass federal test method Standard No. 791 Method 5309.2
	Steam temperature over 825°F	Silicone	1 or 2	265–340	520	Pass Military G-23827 A

* An alternative lubricant for sliding-pedestal supports is a mixture of fine graphite and cylinder or turbine oil mixed to a paste consistency.

point, the temperature at which grease changes from a semisolid to a liquid state. They also should not separate under the action of centrifugal force.

Greases should resist oxidation and must not gum, harden, or decompose. They must not contain dirt, fillers, abrasive matter, excessive moisture, free acid, or free lime.

Oil Maintenance

The lubricating system must be kept clean and free from impurities at all times. The accumulation of impurities will cause lubricant failure and damage to the equipment.

Provision should be made for maximum protection against rust during idle periods. The main lubricating system should be operated at intervals to remove condensation from metal surfaces and coat these surfaces with a protective layer of lubricant. This should be done daily when the variation in day and night temperatures is great and weekly when the variation in day and night temperatures is small. In addition, a unit idle for an extended period of time should, if possible, be operated from time to time at the reduced speeds specified under normal starting procedures.

The use of a suitable oil purifier is recommended. Since some purifiers can alter the properties of lubricating oils, especially inhibited oils, the manufacturer should be consulted before the purifier is selected.

Grease Maintenance

Grease housings should be relubricated routinely when the grease in service is unable to satisfy lubrication demands. The housing should be completely flushed

and filled with new grease and any excess worked out before replacing the drain plug. Care should be taken not to overfill the housings, as this will result in a breakdown of the grease to fluid consistency and overheat the bearings.

In some cases, small additions of fresh grease to the housing are sufficient for proper lubrication. When this procedure is followed, the housing should be completely cleaned and new grease added during each major overhaul.

Lubrication Piping

Oil-feed and oil-drain piping is generally of low-carbon steel. Piping used should be pickled (a procedure of cleaning the internal surfaces). If low-carbon steel piping has not been pickled, the following procedure should be followed:

1. Sandblast pipe along the pipe run.
2. Deburr if necessary.
3. Wash all internal surfaces with a petroleum-base cleaning solvent.
4. Air-blast dry.
5. Visually inspect.
6. If piping is to be stored in house, fog all internal surfaces progressively along the pipe run, through all openings, with an oil-soluble preservative compound.

M

Magnetic Bearings (*see* Bearings)

Measurement* (*see also* Condition Monitoring; Control Systems)

Temperature Measurement

Measurement of temperature is generally considered to be one of the simplest and most accurate measurements performed in engineering. The desired accuracy in the measurement can be obtained, however, only by observing suitable precautions in the selection, installation, and use of temperature-measuring instruments and in the proper interpretation of the results obtained with them.

Four phenomena form the basis for most measuring instruments:

- Change in physical dimensions or characteristics of liquids, metals, or gases
- Changes in electrical resistance
- Thermoelectric effect
- Radiant energy

The following types of instruments are available for use under appropriate conditions:

- Liquid-in-glass thermometer
- Resistance thermometer
- Thermocouple thermometer
- Filled-system thermometer
- Bimetallic thermometer
- Radiation thermometer
- Optical pyrometer

Liquid-in-glass thermometer

A liquid-in-glass thermometer consists of a thin-walled glass bulb attached to a glass capillary stem closed at the opposite end, with the bulb and a portion of the stem filled with an expansive liquid, the remaining part of the stem being filled with the vapor of the liquid or a mixture of this vapor and an inert gas. Etched on the stem is a scale in temperature degrees so arranged that, when calibrated, the reading corresponding to the end of the liquid column indicates the temperature of the bulb. The three types of liquid-in-glass thermometers (Fig. M-1) are partial-immersion, total-immersion, and complete-immersion thermometers.

1. A partial-immersion thermometer is designed to indicate temperature correctly when used with the bulb and a specified part of the liquid column in the stem exposed to the temperature being measured; the remainder of the liquid column

* Source: Demag Delaval, USA.

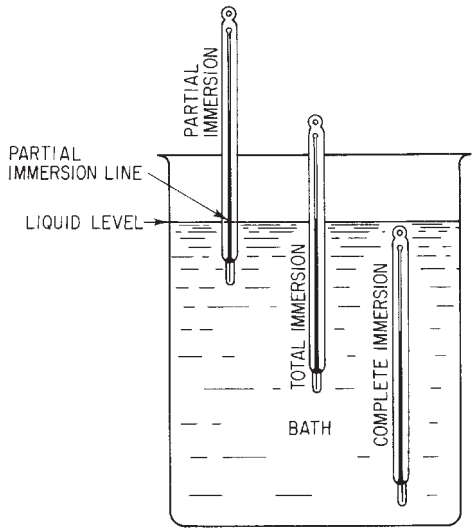


FIG. M-1 Partial-, total-, and complete-immersion thermometers. (Source: Demag Delaval.)

and the gas above the liquid are exposed to a temperature that may or may not be different.

2. A total-immersion thermometer is designed to indicate the temperature correctly when used with the bulb and the entire liquid column in the stem exposed to the temperature being measured and the gas above the liquid exposed to a temperature that may or may not be different.
3. A complete-immersion thermometer is designed to indicate the temperature correctly when used with the bulb, the entire liquid column in the stem, and the gas above the liquid exposed to the temperature being measured.

Tables M-1 and M-2 show National Bureau of Standards (NBS) certification tolerances for laboratory thermometers. The term *tolerance in degrees* means acceptable limits of error of uncertified thermometers. *Accuracy in degrees* is the limit of error to be expected when all necessary precautions are exercised in the use of thermometers. The limits to which NBS certification values are rounded off are shown in the column "Corrections stated to."

The operation of a liquid-in-glass thermometer depends on having the coefficient of expansion of the liquid greater than that of the bulb glass. As a consequence, an increase in temperature of the bulb causes the liquid to be expelled from the bulb, resulting in a rise in position of the end of the liquid column. The capillary stem attached to the bulb serves to magnify this change in volume on a scale.

The most frequently encountered source of error when using liquid-in-glass thermometers is the misuse or complete neglect of the emergent-stem correction. This correction derives from the use of the thermometer with a portion of the stem exposed to a different temperature from that of calibration. A common example is the use of partial immersion of a thermometer calibrated for total immersion. For detailed information on this correction, see the American Society of Mechanical Engineers' *Power Test Codes: Temperature Measurement*.

Resistance thermometer

A resistance thermometer is a temperature-measuring instrument in which electrical resistance is used as a means of temperature measurement. The

TABLE M-1 Tolerances for Fahrenheit Mercurial Total-Immersion Laboratory Thermometers

Temperature Range in Degrees	Graduation Interval in Degrees	Tolerance in Degrees	Accuracy in Degrees	Corrections Stated to
	Thermometers for Low Temperatures			
–35 to 32	1 or 0.5	1	0.1–0.2	0.1
–35 to 32	0.2	0.5	0.05	0.02
	Thermometers Not Graduated above 300°			
32 up to 300	2	1	0.2–0.5	0.2
32 up to 300	1 or 0.5	1	0.1–0.2	0.1
32 up to 212	0.2 or 0.1	0.5	0.02–0.05	0.02
	Thermometers Not Graduated above 600°			
32 up to 212	2 or 1	1	0.2–0.5	0.2
Above 212 up to 600		2	0.5	0.2
	Thermometers Graduated above 600°			
32 up to 600	5	4	0.5–1.0	0.5
Above 600 up to 950		7	1–2	0.5
32 up to 600	2 or 1	3	0.2–1.0	0.2
Above 600 up to 950		6	0.5–1.0	0.2

TABLE M-2 Tolerances for Fahrenheit Mercurial Partial-Immersion Laboratory Thermometers

Temperature Range in Degrees	Graduation Interval in Degrees	Tolerance in Degrees	Accuracy in Degrees	Corrections Stated to
	Thermometers for Low Temperatures			
–35 to 32	1	1	0.3–0.5	0.1
	Thermometers Not Graduated above 300°			
32 up to 300	2 or 1	2	0.2–1.0	0.2
	Thermometers Not Graduated above 600°			
32 up to 212	2 or 1	2	0.2–0.5	0.2
Above 212 up to 600	2 or 1	3	1–2	0.5
	Thermometers Graduated above 600°			
32 up to 600	5 or 2	5.0	1–2	1
Above 600 up to 950		10	2–3	1

instrument consists of a resistor, a resistance-measuring instrument, and electrical conductors connecting the two. The resistor may be metallic (usually in wire form) or a thermistor (a thermally sensitive variable resistor made of ceramiclike semiconducting material).

The basis for resistance thermometry is the fact that most metals and some semiconductors change in resistivity with temperature in a known, reproducible manner. Several materials are commonly employed for resistance thermometers,

the choice depending on the compromises that may be accepted. Although the actual resistance-temperature relation must be determined experimentally, for most metals the following empirical equation holds very closely:

$$R_T = R_0(1 + AT + BT^2) \tag{M-1}$$

where R_T = resistance at temperature T; R_0 = resistance at 0°C; T = temperature, K; and A and B = constants depending on material.

The temperature-resistance function for a thermistor is given by the following relationships:

$$R = R_0 e^k \tag{M-2}$$

$$k = \beta\left(\frac{1}{T} - \frac{1}{T_0}\right) \tag{M-3}$$

where R = resistance at any temperature T, K; R_0 = resistance at reference temperature T_0, K; e = base of napierian logarithms; and β = a constant (which usually has a value between 3400 and 3900, depending on the thermistor formulation or grade).

Types of resistance thermometers

Platinum thermometer. This thermometer is known for its high accuracy, stability, resistance to corrosion, and other characteristics. It has a simple relation between resistivity and temperature, shown in Eq. (M-1).

Precision platinum thermometer. This thermometer is used to define the International Practical Temperature Scale from −297.3 to 1168.3°F. The purity and physical properties of the platinum of which the thermometer is made are prescribed to meet close specifications. Different procedures are used for making precision thermometers to cover different temperature ranges.

Industrial platinum resistance thermometer. The requirements for reproducibility and limit of error for thermometers of this type are lower than those for standard thermometers; so are the manufacturing precautions lowered for these thermometers.

Nickel resistance thermometer. This thermometer has been adapted satisfactorily in industrial applications for a temperature range from −100 to 300°F. The nickel resistance thermometer is less stable than platinum thermometers, but its low cost favors its usage.

Copper resistance thermometer. Copper is an excellent material for resistance thermometers. Its availability in a pure state makes it easy to match with established standards. The resistivity curve of copper is a straight-line function of temperature between −60 and 400°F, and that makes copper resistance thermometers suitable for the measurement of temperature differences with high accuracy. Copper resistance thermometers are reliable and accurate means of temperature measurement at moderate temperature levels.

Thermistors (nonmetallic resistance thermometers). Thermistors are characterized by a negative coefficient of resistivity, and their temperature-resistivity curve is exponential. Modern thermistors are very stable; they have high-temperature sensitivity and very fast response. Because thermistors are high-resistance circuits, the effect of the lead wires is minimized, and regular copper wires can be used throughout the circuit. Noninterchangeability owing to the difficulty of reproducing

TABLE M-3 Typical Characteristics of Resistance Thermometers

	Noble Metal		Nonmetallic
	Precision	Industrial	Thermistor
Sensitivity	0.1Ω/°C	0.22Ω/°F	Varies with units
Precision	±0.001°C	±0.3°F	±0.02°F up to 200°F
Accuracy	±0.01°C	±3.0°F standard ±1.5°F special	±0.5°F standard ±0.2°F special
Response: bare		15 s	Fast
Response: with well		30 s	
Resistance	25.5Ω at 0°C	25Ω at 32°F	Varies with units
Linearity	70.1°C/50°C span	70.1°C/50°C span	Exponential
Range	–452.2 to 1,168.3°F (–269 to 630.74°C)	–297.3 to 1,950°F (–182.96 to 1,064°C)	–100 to 500°F (–75 to 260°C)

	Base Metal		
	10 Ω	Copper, 100 Ω	Nickel, 100 Ω
Sensitivity	0.22Ω/°F	0.22Ω/°F	0.186Ω/°F (0.213Ω/°F)
Precision	±0.1°F	±0.1°F	±0.1°F
Accuracy	±0.5°F standard ±0.2°F special	±0.5°F standard ±0.2°F special	±0.5°F standard ±0.2°F special
Response: bare	20 s	40 s	40 s
Response: with well	60 s	90 s	90 s
Resistance	10 Ω at 77°F	100Ω at 77°F	100 Ω at 77°F
Linearity	Excellent	Excellent	Excellent
Range	–100 to 300°F (–75 to 150°C)	–325 to 300°F (–200 to 150°C)	–100 to 300°F (–75 to 140°C)

resistance properties and the nonlinearity of the resistivity curve limits the use of thermistors.

Information on important characteristics of different classes of resistance thermometers is included in Table M-3.

Accessories. Some forms of Wheatstone-bridge circuits are used for the measurement of temperature with base-metal or industrial platinum resistance thermometers, while the Mueller bridge is used with precision platinum resistance thermometers.

Thermocouple thermometer

A thermocouple thermometer is a temperature-measuring instrument in which the electromotive force developed in a circuit composed of two dissimilar metals is used as a means of temperature measurement. It consists of a device for measuring electromotive force, a sensing element (thermocouple), and electrical conductors operatively connecting the two.

The thermocouple thermometer operates on the principle that an electric current will flow in a closed circuit of two dissimilar metals when the junctions of the metals are at two different temperatures. Thermocouple materials are available for use within the approximate limits of –300 to 3200°F. Platinum is the generally accepted standard material to which the thermoelectric characteristics of other materials are referred. The emf-temperature relations of conventional thermoelements versus platinum are shown in Fig. M-2. Reference tables of

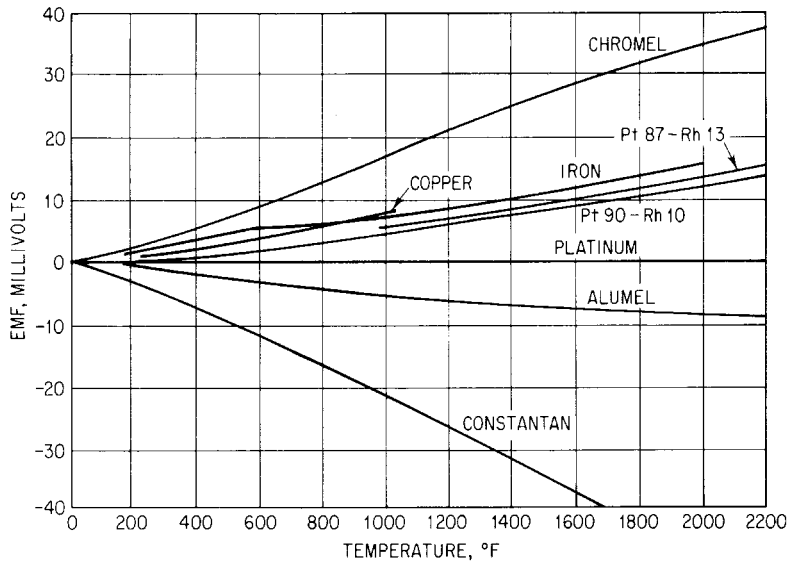


FIG. M-2 emfs of various materials versus platinum. (Source: Demag Delaval.)

TABLE M-4 Limits of Error of Thermocouples

Thermocouple Type	Temperature Range, °C	Limits of Error	
		Standard	Special
T (copper-Constantan)	–184 to –59	. . .	±1%
	–101 to –59	±2%	±1%
	–59 to +93	±0.8°C	±0.4°C
	+93 to +371	±¾%	±⅜%
J (iron-Constantan)	0 to 277	±2.2°C	±1.1°C
	277 to 760	±¾%	±⅜%
E (Chromel-Constantan)	0 to 316	±1.7°C	. . .
	316 to 871	±½%	. . .
K (Chromel-Alumel)	0 to 277	±2.2°C	±1.1°C
	277 to 1260	±¾%	±⅜%
R and S (platinum vs. 13% rhodium-platinum)	0 to 538	±1.4°C	. . .
(platinum vs. 10% rhodium-platinum)	538 to 1482	±¼%	. . .
B (platinum vs. 30% rhodium-platinum vs. 6% rhodium)	871 to 1705	±½%	. . .

temperature versus electromotive force as well as polynomial equations expressing the temperature-voltage relationship for different types of thermocouples are available in technical literature.

The iron-Constantan thermocouple is used most widely in industrial applications. The copper-Constantan thermocouple is used widely in industrial and laboratory thermometry.

The platinum –10 percent rhodium versus platinum (Type S) thermocouple serves as an instrument for defining the International Practical Temperature Scale from 630.74 to 1064.43°C. It is being used in industrial laboratories as a standard for base-metal thermocouples and other temperature-sensing devices.

Table M-4 lists the seven commonly used thermocouples and some of their characteristics.

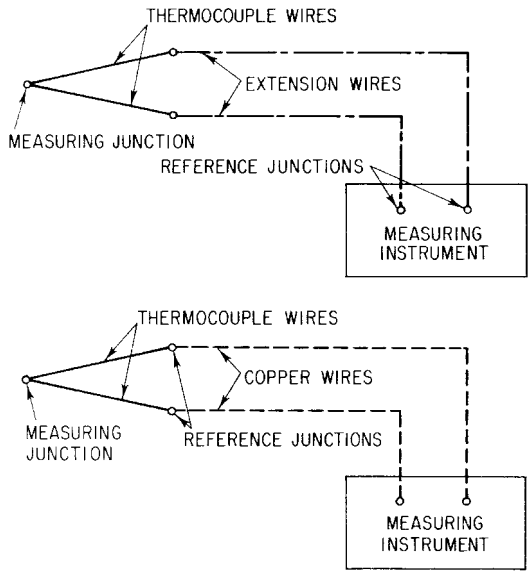


FIG. M-3 Thermocouple thermometer systems. (Source: Demag Delaval.)

The electrical conductors connecting the thermocouple and the measuring instrument may use the actual thermocouple wires, extension wires, or connecting wires (see Fig. M-3). When it is not possible to run the thermocouple wires to the reference junction or to the measuring instrument, extension wires can be used. To assure a high degree of accuracy, extension wires should have the same thermoelectric properties as the thermocouple wires with which they are used. Significant uncertainties are introduced when extension wires are not matched properly. Calibration of the instrument with extension wires helps to minimize these uncertainties. Connecting wires are a pair of conductors that connect the reference junction to the switch or potentiometer. They are usually made of copper. They do not cause uncertainty in measurements when the reference junction is kept at constant temperature, for example, the ice point.

Indicating potentiometers are recommended by the *ASME Power Test Codes* for performance-test work, although recording potentiometers are used for industrial-process temperature measurement.

Thermocouples may be joined in series. The series connection, in which the output is the arithmetic sum of the emfs of the individual thermocouples, may be used to obtain greater measurement sensitivity and accuracy. A series-connected thermocouple assembly is generally referred to as a *thermopile* and is used primarily in measuring small temperature differences. A schematic diagram of a series-connected thermocouple is shown in Fig. M-4.

Thermocouples may also be joined in parallel. In the parallel-connected thermocouple circuit, a mean value of the individual thermocouples is indicated, and it will be the true arithmetic mean if all thermocouple circuits are of equal resistance. A schematic diagram of a parallel-connected thermocouple circuit is shown in Fig. M-5.

The installation of extensive thermocouple equipment requires the services of qualified instrument technicians, and special attention should be given to extension wires, reference junctions, switches, and terminal assemblies.

Opposed thermocouple circuits are sometimes used to obtain a direct reading of a temperature difference between two sets of thermocouples reading two levels of

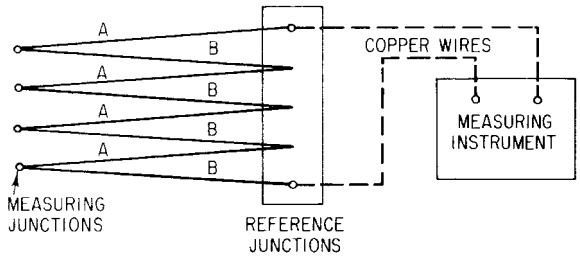


FIG. M-4 Thermocouples connected in series. (Source: Demag Delaval.)

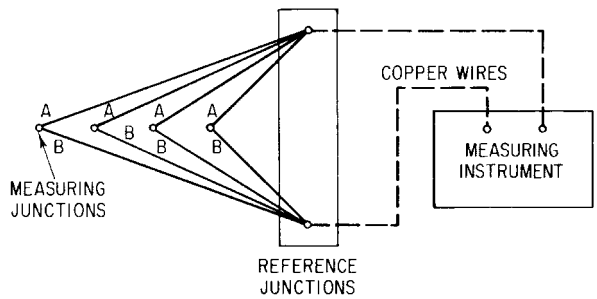


FIG. M-5 Thermocouples connected in parallel. (Source: Demag Delaval.)

temperature. The number of thermocouples in each set is the same. This method is considered to provide the highest degree of accuracy in the measurement of the critical temperature difference.

Filled-system thermometer

A filled-system thermometer (Fig. M-6) is an all-metal assembly consisting of a bulb, a capillary tube, and a Bourdon tube and containing a temperature-responsive fill. Associated with the Bourdon is a mechanical device that is designed to provide an indication or record of temperature.

The sensing element (bulb) contains a fluid that changes in physical characteristics with temperature. This change is communicated to the Bourdon through a capillary tube. The Bourdon provides an essentially linear motion in response to an internally impressed pressure or volume change.

Filled-system thermometers may be separated into two types: those in which the Bourdon responds to volume changes and those that respond to pressure changes. The systems that respond to volume changes are completely filled with mercury or other liquid, and the system that responds to pressure changes is either filled with a gas or partially filled with a volatile liquid.

Bimetallic thermometer

A bimetallic thermometer (Fig. M-7) consists of an indicating or recording device, a sensing element called a bimetallic-thermometer bulb, and a means for operatively connecting the two. Operation depends upon the difference in thermal expansion of two metals. The most common type of bimetallic thermometer used in industrial applications is one in which a strip of composite material is wound in the form of a helix or helices. The composite material consists of dissimilar metals that

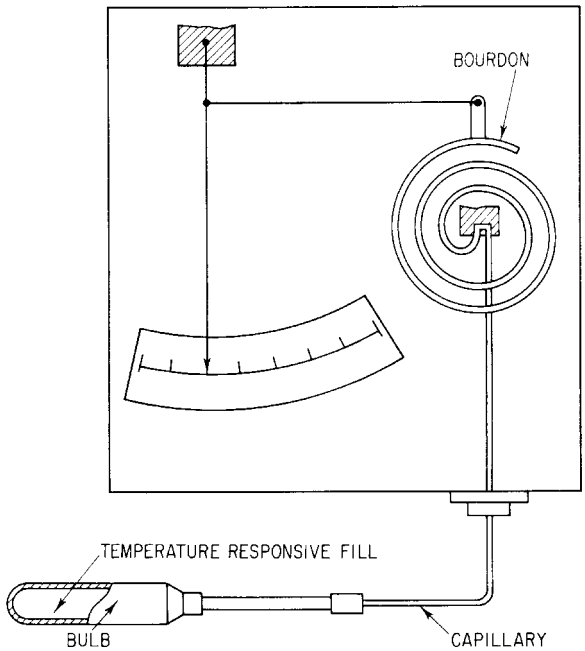


FIG. M-6 Filled-system thermometer. (Source: Demag Delaval.)

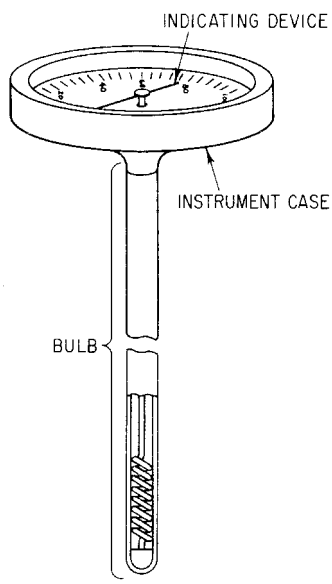


FIG. M-7 Bimetallic thermometer. (Source: Demag Delaval.)

have been fused together to form a laminate. The difference in thermal expansion of the two metals produces a change in curvature of the strip with changes in temperature. The helical construction is used to translate this change in curvature to rotary motion of a shaft connected to the indicating or recording device.

A bimetallic thermometer is a relatively simple and convenient instrument. It comes in industrial and laboratory versions.

Pyrometry

There are two distinct pyrometric instruments, the radiation thermometer and the optical pyrometer, which are described in greater detail in the following subsections. Both pyrometers utilize radiation energy in their operation. Some of the basic laws of radiation transfer of energy will be described briefly.

All bodies above absolute-zero temperature radiate energy. This energy is transmitted as electromagnetic waves. Waves striking the surface of a substance are partially absorbed, partially reflected, and partially transmitted. These portions are measured in terms of absorptivity α, reflectivity ρ, and transmissivity τ, where

$$\alpha + \rho + \tau = 1 \tag{M-4}$$

For an ideal reflector, a condition approached by a highly polished surface, $\rho \rightarrow 1$. Many gases represent substances of high transmissivity, for which $\tau \rightarrow 1$, and a blackbody approaches the ideal absorber, for which $\alpha \rightarrow 1$.

A good absorber is also a good radiator, and it may be concluded that the ideal radiator is one for which the value of α is equal to unity. In referring to radiation as distinguished from absorption, the term *emissivity* ε is used rather than *absorptivity* α. The Stefan-Boltzmann law for the net rate of exchange of energy between two ideal radiators A and B is

$$q = \sigma(T^4{}_A - T^4{}_B) \tag{M-5}$$

where q = radiant-heat transfer, Btu/h·ft^2; σ = Stefan-Boltzmann constant; and T_A, T_B = absolute temperature of two radiators.

If we assume that one of the radiators is a receiver, the Stefan-Boltzmann law makes it possible to measure the temperature of a source by measuring the intensity of the radiation that it emits. This is accomplished in a radiation thermometer.

Wien's law, which is an approximation of Planck's law, states that

$$N_{b\lambda} = C_1 \lambda^{-5} e^{-C_2/\lambda T} \tag{M-6}$$

where $N_{b\lambda}$ = spectral radiance of a blackbody at wavelength λ and temperature T; C_1, C_2 = constants; λ = wavelength of radiant energy; and T = absolute temperature.

The intensity of radiation $N_{b\lambda}$ can be determined by an optical pyrometer at a specific wavelength as a function of temperature, and then it becomes a measure of the temperature of a source.

Radiation thermometer

A radiation thermometer consists of an optical system used to intercept and concentrate a definite portion of the radiation emitted from the body whose temperature is being measured, a temperature-sensitive element, usually a thermocouple or thermopile, and an emf-measuring instrument. A balance is quickly established between the energy absorbed by the receiver and that dissipated by conduction through leads, convection, and emission to surroundings. The receiver equilibrium temperature then becomes the measure of source temperature, with the scale established by calibration. An increase in the temperature of the source is accompanied by an increase in the temperature of the receiver that is proportional to the difference of the fourth powers of the final and initial temperatures of the source.

The radiation thermometer is generally designated as a total-radiation thermometer that utilizes, as an index of the temperature of a body, all the energy

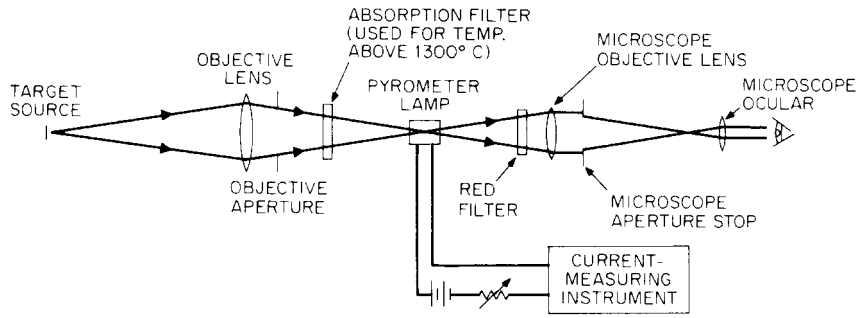


FIG. M-8 Schematic diagram of an optical pyrometer. (Source: Demag Delaval.)

(all wavelengths) per unit area per unit time radiated by the body. Radiation thermometers are classified according to the method of collecting the radiation and focusing it on the receiver: single mirror, double mirror, and lens.

The radiation thermometer can be classified not as a primary laboratory instrument but rather as an industrial instrument. Its practical useful range extends from ambient temperature to 7500°F, although different thermometers must be used to cover this range.

Optical pyrometer

Optical pyrometers use a method of matching as the basis of operation. Generally, a reference temperature is provided in the form of an electrically heated lamp filament, and a measure of temperature is obtained by optically comparing the visual radiation from the filament with that from the unknown source. In principle, the radiation from one of the sources, as viewed by the observer, is adjusted to match that from the other source. Two methods are employed: (1) the current through the filament may be controlled electrically, through a resistance adjustment; or (2) the radiation accepted by the pyrometer from the unknown source may be adjusted optically by means of some absorbing device such as an optical wedge, a polarizing filter, or an iris diaphragm. The two methods are referred to, respectively, as the method using the variable-intensity comparison lamp and the method using the constant-intensity comparison lamp. In both cases the adjustment required is used as the means for temperature readout. Figure M-8 illustrates schematically an arrangement of a variable-intensity pyrometer.

A typical optical pyrometer consists of a power supply and an optical system. The optical system incorporates a telescope, a calibrated lamp, a filter for viewing nearly monochromatic radiation, and an absorption glass filter (see Fig. M-8). The filament of the lamp and the test body are viewed simultaneously. The filament current is adjusted until the filament image disappears in the image of the test body.

Visual optical pyrometers should not be used for the measurement of temperatures below 1400 °F. Automatic optical pyrometers can be used for the measurement of lower temperatures, and they are of great value in the measurement of very high temperatures.

Calibration

To compare or to measure temperature, a temperature scale is necessary. Two ideal temperature scales were proposed: the thermodynamic scale of Kelvin and the ideal-gas scale. The International Committee on Weights and Measures came up with a

more practical temperature scale, the International Practical Temperature Scale of 1968 (IPTS-68), which is based on 11 fixed, reproducible temperature points.

There are two widely used temperature scales in engineering practice. The first, the Celsius scale, derives directly from IPTS-68; it has 100 units (degrees) between the ice point and the steam point of water. The second, the Fahrenheit scale, has 180 units (degrees) between these two fixed temperature points. In the first case the freezing point is marked 0, while in the second case this point is marked 32. The relationship between the two scales is as follows:

$$F = {}^{9}/_{5}\,C + 32, \text{ degrees Fahrenheit}$$

$$C = {}^{5}/_{9}\,(F - 32), \text{ degrees Celsius}$$

Calibration at fixed points is a complex process. Standard platinum resistance thermometers and standard platinum-rhodium-platinum thermocouples are calibrated at fixed points for use as primary standards. It is recommended that calibration be done by the NBS or other qualified laboratory. The narrow-band optical pyrometer is another primary standard; its range over the freezing point of gold is obtained through extrapolation. Ordinary calibration of temperature-measuring instruments is effected by comparison of their readings with those of primary or secondary standards at temperatures other than fixed points. Comparators are used to produce those temperatures.

Secondary standards are liquid-in-glass thermometers and base-metal thermocouples. They are calibrated by comparing them with primary-standard platinum-resistance thermometers or standard platinum-rhodium versus platinum thermocouples at temperatures generated in comparators. These secondary standards are used in turn for the calibration of other devices, such as liquid-in-glass thermometers, bimetallic thermometers, filled-system thermometers, and base-metal thermocouples, in which the highest degree of accuracy is not required. Optical pyrometers as secondary standards are compared with primary-standard optical pyrometers, and they are then used for calibration of regular test pyrometers.

There is ample literature by the American Society of Mechanical Engineers (ASME), the American Society for Testing and Materials, the NBS, and others that deals with calibration methods, specifications for construction and usage of measuring instruments and temperature comparators, and processing of calibration data. It is advisable in each case to have the major components of the system (primary and secondary standards), potentiometers, and Mueller bridges calibrated periodically by the NBS or other qualified laboratory.

Other considerations

The preceding presentation on temperature measurement shows clearly how complex the subject is and what precautions must be taken to obtain a meaningful temperature measurement. The proper use of the right temperature-measuring instruments is very important. Calibration for instrumental errors is mandatory for temperature-sensing devices and other temperature-measurement-system components; periodic checking of the calibration is also very important.

If for reasons of protection of the sensitive temperature-measuring element against corrosive atmosphere or excessive mechanical stress, the use of thermometer wells is prescribed, such wells should be designed and installed with the utmost care to avoid damage and the introduction of additional errors. The *ASME Power Test Codes* should be followed in this respect. The most important precautions in using a thermometer well are to keep the sensing element in

intimate contact with the well and to have the exposed parts of the well as small as possible and insulated from their surroundings.

The nature of heat transfer between the medium, the temperature of which is being measured, and the sensing element and the sources of temperature errors due to conduction, radiation, and aerodynamic heating are described below. The temperature-sensing element indicates its own temperature, which may not be the exact temperature of the fluid in which it is inserted. The indicated temperature is established as a result of heat-flow equilibrium of convective heat transfer between the sensing element and the fluid on one side and heat flow through conduction and radiation between the element and its surroundings on the other side. This applies closely to fluids at rest or to fluids moving with low velocities. The conditions are more complex for fluids moving at higher velocities (corresponding to a Mach number greater than 0.3), in which the aerodynamic heating effect plays a greater part in heat balance.

Conduction error. Conduction error, or immersion error, is caused by temperature gradients between the sensing element and the measuring junction. This error can be minimized by high heat convection between fluid and sensor and low heat conduction between sensor and measuring junction. In the thermocouple this would mean a small diameter, low conductivity, and long immersion length of the wires.

Radiation error. When the sensing element (other than radiation thermometer) is placed so that it can "see" surfaces at a much lower temperature (a sink) or at a much higher temperature (a source), a radiant-heat interchange will result between the two, causing the sensing element to read an erroneous temperature.

Radiation error may be largely eliminated through the proper use of thermal shielding. This consists in placing barriers to thermal radiation around the probe, which prevent the probe from seeing the radiant source or sink, as the case may be. For low-temperature work, such shields may simply be made of sheet metal appropriately formed to provide the necessary protection. At higher temperatures, metal or ceramic sleeves or tubes may be employed. In applications in which gas temperatures are desired, care must be exercised so as not to cause stagnation of flow around the probe.

Measurement of temperature in a rapidly moving gas. When a probe is placed in a stream of gas, the flow will be partially stopped by the presence of the probe. The lost kinetic energy will be converted to heat, which will have some bearing on the indicated temperature. Two "ideal" states may be defined for such a condition. A true state would be that observed by instruments moving with the stream, and a stagnation state would be that obtained if the gas were brought to rest and its kinetic energy completely converted to heat, resulting in a temperature rise. A fixed probe inserted into the moving stream will indicate conditions lying between the two states.

An expression relating stagnation and true temperatures for a moving gas, with adiabatic conditions assumed, may be written as follows:

$$t_s - t_t = \frac{V^2}{2g_c J c_p} \tag{M-7}$$

This relation may also be written as

$$\frac{t_s}{t_t} = 1 + \frac{(k-1)M^2}{2} \tag{M-8}$$

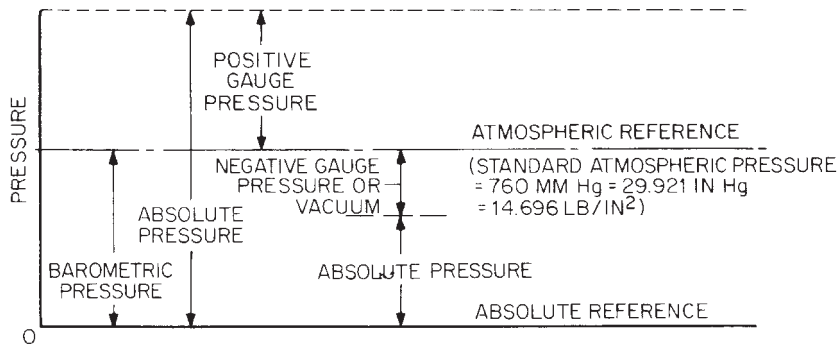


FIG. M-9 Relations between absolute, gauge, and barometric pressure. (Source: Demag Delaval.)

where t_s = stagnation or total temperature, °F; t_t = true or static temperature, °F; V = velocity of flow, ft/s; g_c = gravitational constant, 32.2 ft/s^2; J = mechanical equivalent of heat, ft · lb/Btu; c_p = mean specific heat at constant pressure, Btu/lb · °F; k = ratio of specific heats; and M = Mach number.

A measure of effectiveness of the probe in bringing about kinetic-energy conversion may be expressed by the relation

$$r = \frac{t_i - t_t}{t_s - t_t} \tag{M-9}$$

where t_i = temperature indicated by the probe, °F, and r = recovery factor.

If $r = 1$, the probe would measure the stagnation temperature, and if $r = 0$, it would measure the true temperature.

By combining Eqs. (M-7) and (M-9), the following relationships are obtained:

$$t_t = t_i - \frac{rV^2}{2g_c J c_p} \tag{M-10}$$

or

$$t_s = t_i + \frac{(1-r)V^2}{2g_c J c_p} \tag{M-11}$$

The value of the recovery factor r depends on the type and design of the temperature-measuring probe; it can be anywhere between 0 and 1.0. Often it is specified by the manufacturer for the specific designs of the temperature probes, or it should be determined experimentally. The difference between stagnation and static temperature increases rapidly as the flow Mach number increases. It is important therefore to know the value of the recovery factor in order to get an as accurate as possible evaluation of the temperatures of the moving gas.

Pressure Measurement

General principles and definitions

1. *Pressure* is defined as the force per unit area exerted by a fluid on a containing wall.

2. *Pressure relationships* (Fig. M-9). *Differential pressure* is the difference between any two pressures. *Absolute pressure* is the force per unit area exerted by

a fluid on a containing wall. *Gauge pressure* is the difference between absolute pressure and ambient-atmospheric pressure. *Vacuum pressure* is negative gauge pressure.

3. *Flow-stream pressures*. *Static pressure* is pressure measured perpendicularly to the direction of flow. This is the pressure that one would sense when moving downstream with the fluid. *Total pressure* is pressure in the direction of flow, where pressure as a function of direction is at a maximum. Total pressure would be sensed if the stream were brought to rest isentropically. *Velocity pressure* is the difference between static and total pressure measured at a specific region in the direction of flow. It is called *velocity head* when measured in height of fluid. Velocity pressure is equal to $^1/_2\rho V^2$, where ρ is the fluid density and V is the fluid velocity.

Pressure connections

1. *Sources of errors. Flow errors:* Leakage errors can be eliminated by proper sealing of connections. Errors due to friction, inertia, and lag errors in the gauge piping, encountered in dynamic flow, can be minimized by using short large-diameter connecting tubes. *Turbulence errors:* The static-pressure tap on the wall parallel to the flow should not be too large in order to prevent a disturbance in the flow that would cause an inaccurate static reading; the tap, however, should be large enough to give a proper response. The area surrounding the pressure tap should be smooth to ensure that a burr or other obstruction will not affect the reading. The edge of the hole should be sharp and square. When the pressure is fluctuating, a damping device can be used to improve readability, although the accurate method would be to use a suitable recording instrument and determine the average pressure over a period of time.

2. *Static taps*. Static taps (Fig. M-10*a*) should be at least 5 diameters downstream from symmetrical pipe fittings and 10 diameters downstream from unsymmetrical fittings, according to the *ASME Power Test Codes*. When possible, a Weldolet or pipe coupling should be welded to the outside of the pipe and the hole then drilled through to the main pipe. Since the error increases with velocity pressures, care must be taken in high-velocity areas to ensure sharp, square holes that are as small as possible (down to $^1/_{16}$ in) to keep disturbance and error to a minimum. In low-velocity areas, larger holes should be used to improve dynamic-pressure response and prevent clogging. When flow is nonuniform, several taps should be used along the periphery of the pipe.

3. *Static tubes*. Static tubes (Fig. M-10*b*) are used for measurement of static pressure in a free stream as on a moving plane. Static taps in the wall are preferable, since static tubes disturb the flow, making calibration necessary for accurate measurement. Unless one expects a static-pressure distribution, wall taps should be used.

4. *Impact tubes or pitot tubes*. An impact tube (Fig. M-10*c*) faces directly into the flow, giving a total-pressure reading. Velocity pressure is determined by taking a static reading, preferably along a wall, and taking the difference; impact tubes can be used to get a velocity profile by traversing. Maximum-velocity direction can be determined by rotating the tube.

5. *Piping arrangement*. Connecting piping (Fig. M-10*f*) should be arranged to avoid liquid pockets in gas-filled lines and air pockets in liquid-filled lines. This is accomplished by having gas-filled lines sloping up to the measuring instrument and liquid-filled lines sloping downward to the instrument. Both types should have vents close to the instrument to bleed lines. More vents might be needed if lines must have dips or twists in them. For vacuum pressures an air bleed allowing

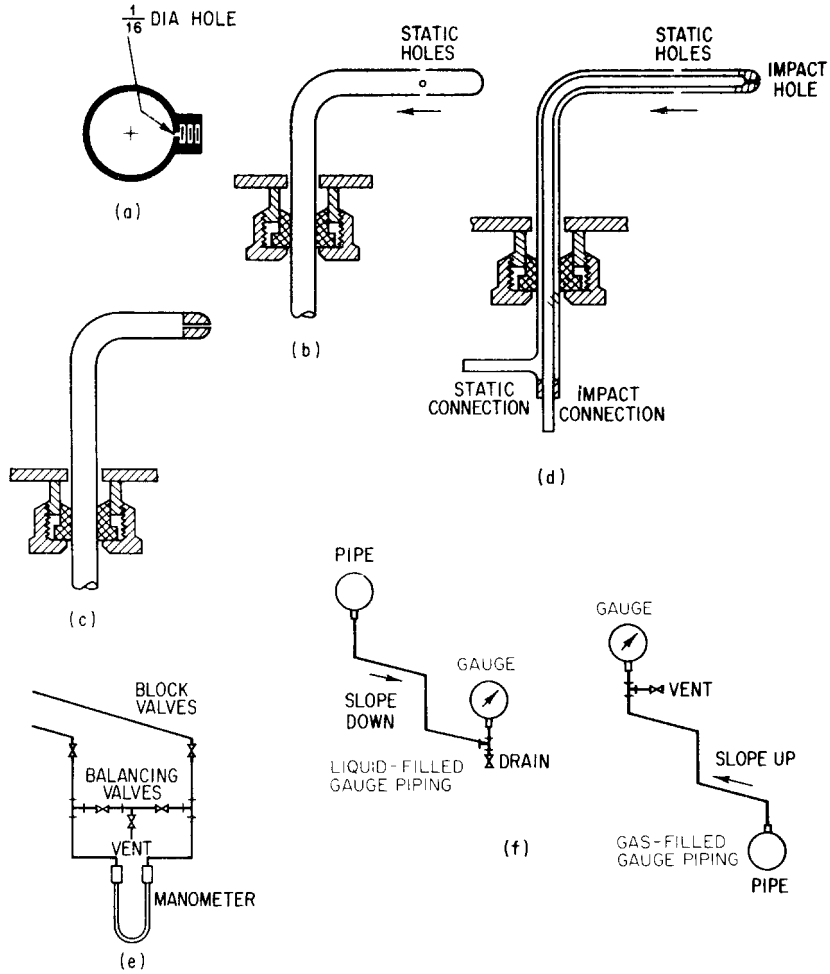


FIG. M-10 (*a*) Static-pressure connection. (*b*) Static tube. (*c*) Impact tube. (*d*) Combination pitot-static tube. (*e*) Cross connection. (*f*) Typical pressure-gauge piping arrangements. (Source: Demag Delaval.)

very small flow should be provided near the instrument to keep lines purged of condensate, etc., between readings. When using a manometer, this can be accomplished by a valve, or a very small hole can be drilled near the top of the manometer, which would be closed or covered when taking a reading. For mechanical or electrical transducers as measuring instruments, the same procedures hold true, but in hot-steam lines it might be necessary to loop the line and fill with water close to the instrument to protect the instrument from the high temperature. For differential measurement the arrangement shown in Fig. M-10*e* should be used to prevent and detect leakage.

6. *Calibration and error analysis*. Pressure measurements are referred to primary standards of pressure, which can be calibrated in terms of mass, length, and time. All pressure-measuring devices have an associated error that must be considered in making a pressure measurement. In a field environment of noise, vibration, moisture, temperature fluctuation, pressure fluctuation, pressure-tap geometry, connecting tubing, etc., other errors or uncertainties must be considered in evaluating the pressure measurement.

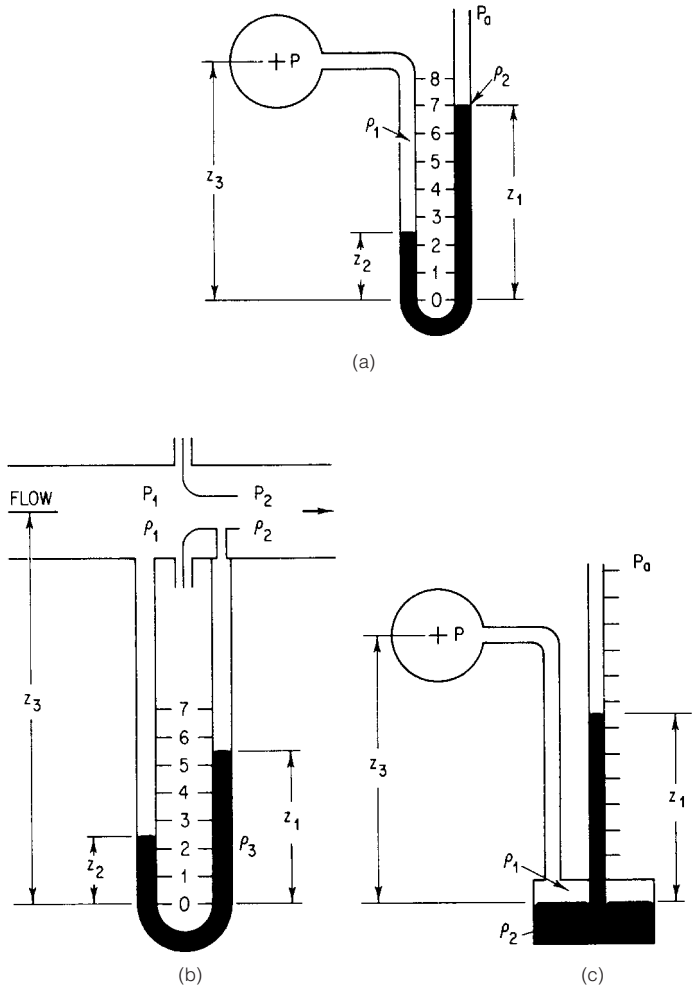


FIG. M-11 Manometer types. (*a*) U-tube manometer, open to the atmosphere. (*b*) Differential U-tube manometer. (*c*) Cistern manometer. (Source: Demag Delaval.)

Liquid-level gauge

1. A manometer measures pressure by balancing it against a column of liquid with a known density and height. Selection of the liquid depends on test conditions; however, the liquid must always be denser than the flowing fluid and immiscible with it. Other factors to consider are the specific gravity, the useful temperature range, the flash point, the viscosity, and the vapor pressure. The basic manometer liquids used are water (specific gravity, 1), mercury (specific gravity, 13.57), red oil (Meriam; specific gravity, 0.827), tetrabromoethylene (specific gravity, 2.95), and carbon tetrachloride (specific gravity, 1.595). Special fluids are also available with specific gravities of 1.20 and 1.75. The manometer fluid used must be kept pure to ensure that the specific gravity remains constant.

2. The basic types of manometers are U-tube and cistern (Fig. M-11).

In the U-tube manometer the pressure on one leg balances the pressure on the other leg. By performing a fluid balance and knowing the density of all fluids and their height, one can calculate the pounds per square inch difference between the two. Often the second leg is open to atmospheric pressure so that the pressure difference represents gauge pressure and must be added to barometric pressure to

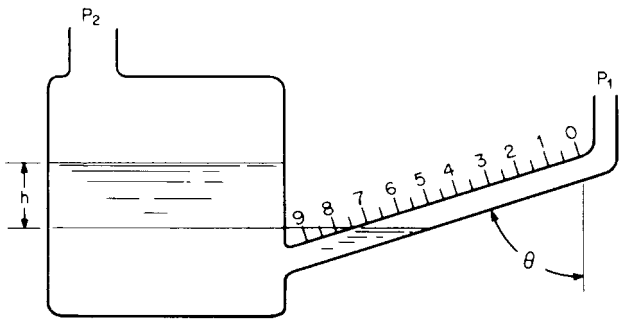


FIG. M-12 Inclined manometer. (Source: Demag Delaval.)

find the total pressure. When the second leg is connected to a pressure other than atmospheric, it is called a *differential pressure* and represents the direct difference between the two pressures.

In the well- or cistern-type manometer one leg has a cross section much larger than that of the other leg. The zero adjustment in the cistern is usually made manually with an adjusting screw. Then the pressure is found by the following formula

$$P = P_a + Z_1 g\rho_2 - Z_3 g\rho_1$$

where g = acceleration due to gravity and ρ = density of liquid.

Special types of manometers sometimes used for more accurate measurement include the inclined manometer, the micromanometer, and U tubes installed with hook gauges, as well as many special types of manometers for vacuum measurement, which will be mentioned later.

3. An inclined manometer (Fig. M-12) is a manometer inclined at an angle with the vertical. Although the vertical displacement is still the same, the movement of liquid along the tube is greater in proportion to the secant of the angle. The common form of inclined manometer is made with a cistern, as shown in Fig. M-12.

The scale can be graduated to take account of the liquid density, inclination, and cistern-level shift so that readings will be in convenient pressure units such as equivalent vertical inches or centimeters of water. A spirit level and leveling screws are usually provided so that the designed angle can be reproduced in installation.

This form of manometer is useful for gas pressures, as for draft gauges. The graduation intervals are commonly 0.01 in of water (0.25 mm of water) with spans up to about 10 in (25 cm).

4. Barometers are a special case of manometers to measure atmospheric pressure. A primary barometer is a U tube with one end open to the atmosphere and the other end connected to a continuously operating vacuum pump.

In many cases a Fortin-type barometer (Fig. M-13) is suitable. In this case the mercury in the well is exposed to the atmosphere with the other end evacuated and sealed. All barometer readings should be corrected for temperature, local gravity, and capillary effect. Atmospheric pressure can also be measured by an aneroid barometer, which is a special type of elastic gauge. It is sometimes used in place of a manometer-type barometer because of the ease of transportation.

Deadweight tester and gauges

1. *Principle, design, and operation testers.* Deadweight testers are the most common instrument for calibrating elastic gauges with pressures in the range of 15 to about 10,000 lb/in^2 or higher.

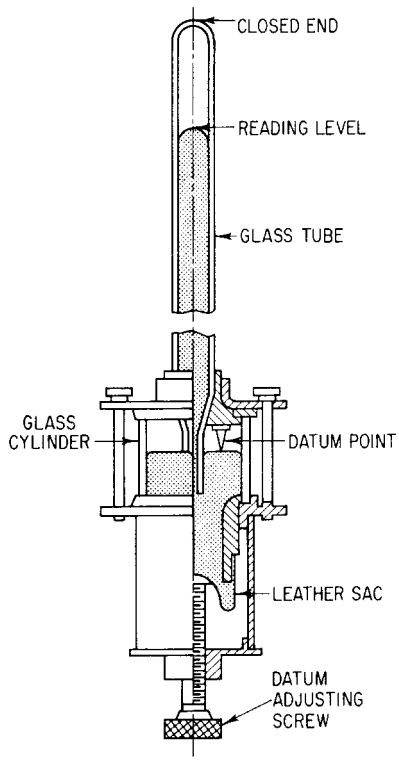


FIG. M-13 Fortin barometer. (Source: Demag Delaval.)

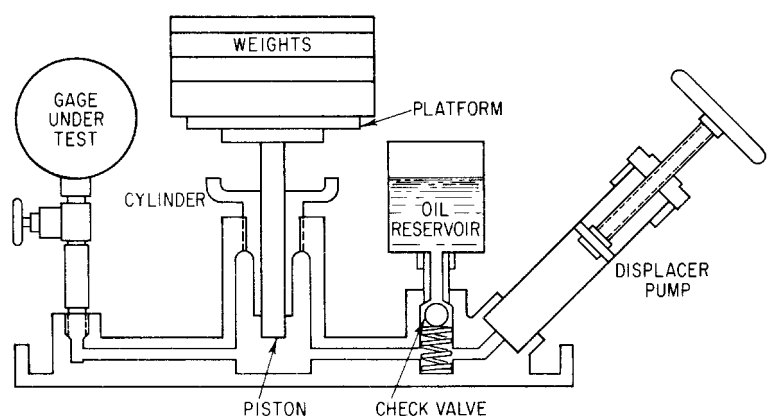


FIG. M-14 Deadweight tester. (Source: Demag Delaval.)

Deadweight testers (Fig. M-14) have a piston riding in a cylinder with a close clearance. The total weight on the piston including that of the platform and the piston itself and any additional weights, divided by the cross-section area of the piston (which is usually an even fraction of an inch such as $^1/_8$ in^2), determine the pressure on the gauge being tested. The piston must be in a vertical position and spinning freely when the measurement is taken. The inertia created by spinning minimizes the viscous drag on the piston by spreading oil around the diameter. Maximum error is usually 0.1 percent of the pressure measured.

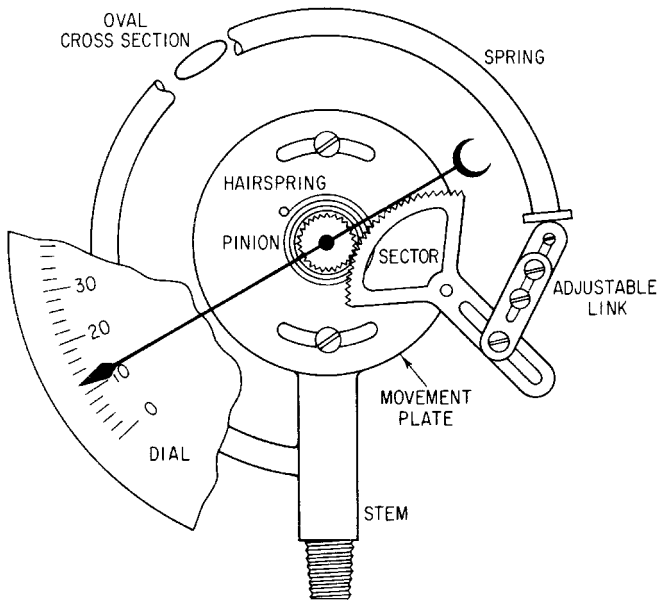


FIG. M-15 Bourdon gauge. (Source: Demag Delaval.)

To operate, put the desired weight on the piston, close the pressure-release valve, and pressurize the tester's fluid with the displacer pump or screw-type ram until the weights are lifted and the piston is floating. Then slowly spin the piston, and take the gauge reading and compare it with the equivalent pressure created by the piston and weights. The gauge reading must then be corrected accordingly.

Special testers include high-pressure, low-pressure, and lever types. For very high pressure (above 10,000 lb/in^2) it is necessary to use a tester that makes adjustments to minimize the leakage and to correct for deformation of the piston and cylinder. Low-pressure testers (0.3 to 50 lb/in^2 are covered) use air as the working fluid for a more accurate measurement. Lever-type testers use a force-amplifying linkage to apply weight to the piston with an inertial wheel on a motor to keep the piston spinning freely.

2. *Deadweight gauges*. Deadweight gauges are mainly used to measure a relatively stable pressure so that it can be maintained. These gauges give very precise measurements but are not practical for a test with a wide range of pressures since many weight changes would be necessary.

3. *Corrections.* Corrections include those necessary for local gravity, weight measurement, effective area, head, and buoyancy adjustments. The head correction is usually the only one necessary when accuracy of $^1/_4$ percent is satisfactory.

Elastic gauges

1. In elastic gauges, an elastic member is caused to stretch or move by a given pressure. The movement is amplified through a linkage and usually is employed to rotate a pointer indicating the pressure reading in relation to atmospheric pressure.

2. Bourdon gauges (Fig. M-15) contain a hollow tube curved in an arc that tends to straighten as internal pressure is applied, moving the linkage and pointer to indicate the pressure reading. Differential as well as compound, vacuum, and straight-pressure Bourdon gauges are available. Differential-pressure gauges have

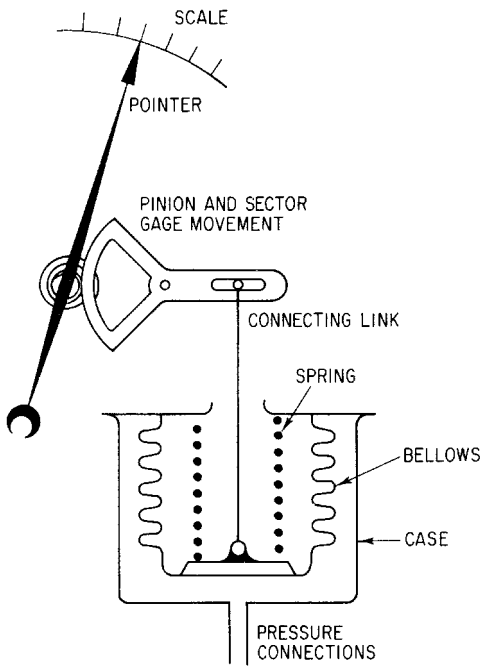


FIG. M-16 Bellows gauge. (Source: Demag Delaval.)

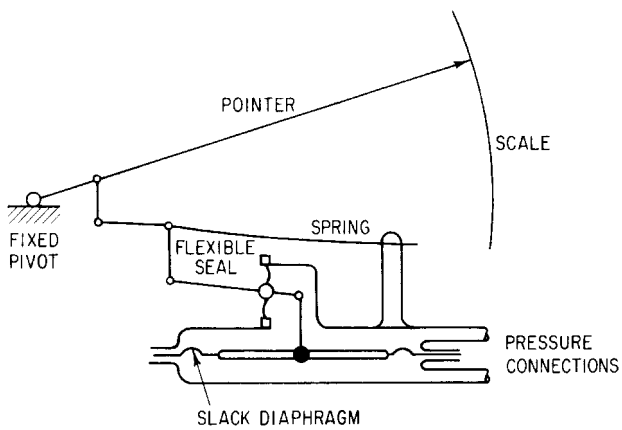


FIG. M-17 Slack-diaphragm gauge. (Source: Demag Delaval.)

either the Bourdon tube enclosed in a seal-pressurized case or two Bourdon gauges, one subtracting from the other. Ranges go from 0 to 15 psig to 0 to 100,000 psig as well as the vacuum range.

3. Bellows gauges (Fig. M-16) have a bellows or elastic chamber expanding to actuate the gauge. They are usually used in low-pressure applications with a maximum reading of about 50 psig.

4. Diaphragm gauges (Fig. M-17) use a flexible diaphragm as the inducer. This type is suitable for ranges from 0 to 1 inHg up to 200 lb/in^2. Variations of this gauge are valuable in special cases in which the process fluid must be kept separate from the gauge, as when the fluid is very hot (up to 1500°F with special modifications) or corrosive, or when the fluid would tend to clog other gauges.

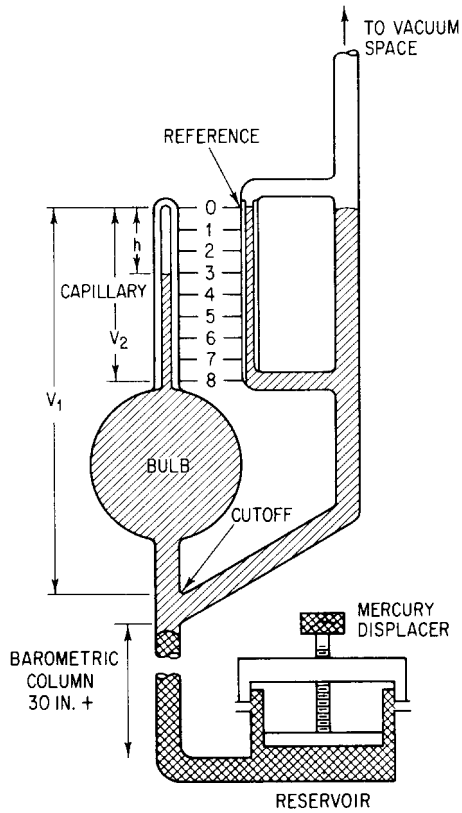


FIG. M-18 McLeod gauge. (Source: Demag Delaval.)

5. All elastic gauges must be calibrated continually to ensure accuracy. Accuracy to 0.5 percent or better of full scale can be obtained.

6. Gauges must be bled for assurance that neither air nor water bubbles are present in the lines. To obtain a gauge reading, first make sure that the linkage is free. This is done with a light tap to the gauge. When damping the gauge needle by closing down on the inlet line, the needle is left fluctuating slightly to indicate that the line is still open.

Special measuring devices

For low-pressure measurement the McLeod gauge (Fig. M-18) is a primary measuring device. The calibration depends only upon dimensional measurements. Other direct-reading pressure gauges, for low pressure, are the mercury micromanometer, the Hickman butyl phthalate manometer, and the consolidated diaphragm comparator. Gauges measuring properties directly convertible to pressure are the thermal-conductivity gauges (thermocouple gauge, Pirani gauge), ionization gauges (Philips-Penning gauge, alphatron gauge), and the molecular-vacuum gauge.

Electric transducers. Devices that convert a pressure into a mechanical analog of that pressure, such as a manometer, which exhibits a difference in the height of a liquid column, were discussed in the preceding subsections. Practical reasons make it difficult to transmit these mechanical signals over large distances, but modern control systems require this capability. Transmitting information over great

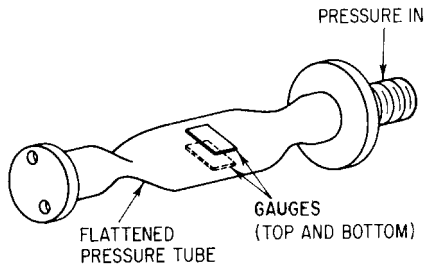


FIG. M-19 Bonded strain gauge. (Source: Demag Delaval.)

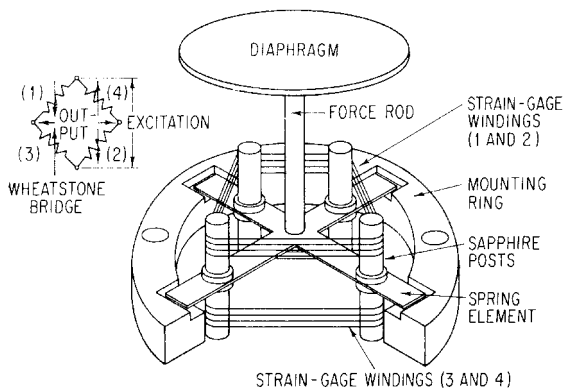


FIG. M-20 Unbonded strain gauge. (Source: Demag Delaval.)

distances is easily accomplished by electronic instrumentation. Transmission of signals representing measured pressure can be accomplished by varying an electric current through wires to the remote location.

The device used to obtain an electronic signal that is related to a pressure is an electric transducer. An electric transducer consists of the following:

1. *Sensing element*. A device that receives a pressure signal and converts it to a force or displacement.
2. *Transmitter*. A device that contains a sensing element, detects the force or displacement in the sensing element, and sends an electric signal (related to the force or displacement) to a receiver.

Implicit in the use of an electric transducer is a receiver that detects the electric signal and indicates the pressure.

Many of these transducers measure the resistance change of a wire or strain gauge deformed by pressure. These instruments can be calibrated to measure pressure directly. There are two basic ways of mounting these gauges. With bonded strain gauges (Fig. M-19) they are usually mounted on a diaphragm or tube that will deform as pressure is applied, changing the resistance of the gauge. With unbonded strain gauges (Fig. M-20) a thin wire is usually wrapped around a sensing element that deforms and stretches the wire, changing its resistance, which can also be converted to pressure. Another special type of resistance gauge for high-pressure reading is the bulk-modulus pressure gauge (Fig. M-21), which uses direct pressures on a loosely wound coil of fine wire to get a resistance change. The sensing mechanism is separated from the process fluid by a bellows. Other specialized types use differences in inductance or capacitance and correlate them to pressure.

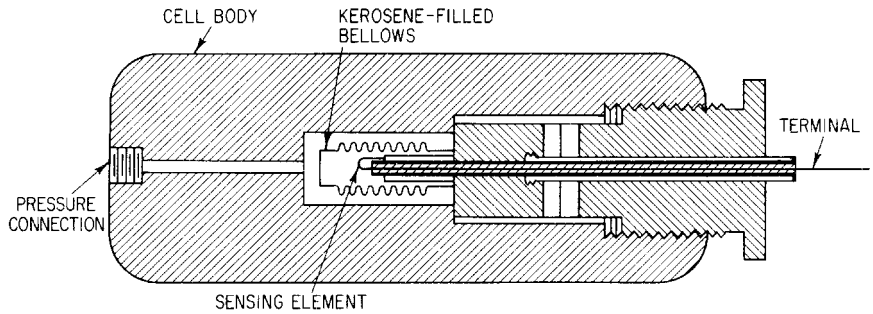


FIG. M-21 Section through a bulk-modulus pressure gauge. (Source: Demag Delaval.)

Flow Measurement

General

The three most extensively used types of flow-metering devices are the thin-plate square-edged orifice, the flow nozzle, and the venturi tube. They are differential-head instruments and require secondary elements for measurement of the differential pressure produced by the primary element. The Supplement to *ASME Power Test Codes: Instruments and Apparatus*, describes construction of the above primary flow-measuring elements and their installation as well as installation of the secondary elements. The method of flow measurement, the equations for flow computation, and the limitations and accuracy of measurements are discussed. Diagrams and tables showing the necessary flow coefficients as a function of Reynolds number and diameter ratio β are included in the standards. Diagrams of the expansion factor for compressible fluids are given.

Some characteristic features of various types of primary elements are listed in the following:

- *Orifice*. Simple, inexpensive, well-established coefficient of discharge, high head loss, low capacity for given pipe size, danger of suspended-matter accumulation; requires careful installation of pressure connections.
- *Flow nozzle*. High capacity, more expensive, loss comparable with that of the orifice; requires careful installation of pressure connections.
- *Venturi tube*. High capacity, low head loss, most expensive, greater weight and size; has integral pressure connection.

Nomenclature

a = throat area of primary element, in^2
C = coefficient of discharge
d = throat diameter of primary element, in
D = pipe diameter, in
E = $1/\sqrt{1-\beta^4}$, velocity-of-approach factor
F_a = thermal-expansion factor
h = manometer differential pressure, in
h_w = manometer differential pressure, in H_2O at 68°F
k = ratio of specific heats
K = CE = combined-flow coefficient for orifices, velocity-of-approach factor included
n = numerical factor dependent upon units used
q = capacity of flow, gal/min

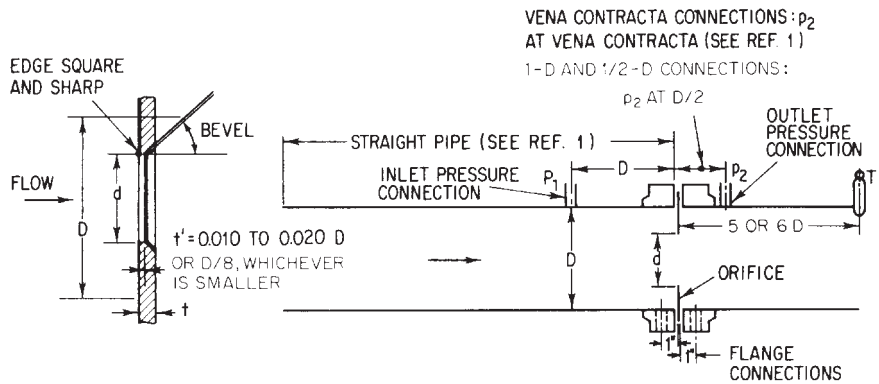


FIG. M-22 Orifice construction and installation. (Source: Demag Delaval.)

Q_i = capacity of flow, ft^3/min, at conditions i
R_d = Reynolds number based on d
$r = \dfrac{P_2}{P_1}$ = pressure ratio across flow nozzle, where P_1 and P_2 are absolute pressures
M = rate of flow, lb/s
M_h = rate of flow, lb/h
M_m = rate of flow, lb/min
Y = net-expansion factor for square-edged orifices
Y_a = adiabatic-expansion factor for flow nozzles and venturi tubes
$\beta = \dfrac{d}{D}$ = diameter ratio
ρ = specific weight of flowing fluid at inlet side of primary element, lb/ft^3
ρ_i = specific weight of flowing fluid at conditions i

Primary-element construction and installation

The primary element may be installed within a continuous section of pipe flowing full or at the inlet or exit of a pipe or a plenum chamber. Orifice and venturi tube are installed within the pipe in a closed-loop test. The flow nozzle may be installed within, at inlet, or at outlet of the pipe.

It is normal practice to use a venturi tube installed within a continuous section of pipe in pump-acceptance tests and a flow nozzle at the exit of the discharge pipe in compressor-acceptance tests. More closed-loop testing has recently been required in compressor testing. Industry normally uses the nozzle configuration shown in Fig. M-26 with a closed loop. The construction of the primary elements and examples of their installation are given in the following paragraphs.

Orifice (Fig. M-22). The recommended diameter ratio $\beta = d/D$ is from 0.20 to 0.75. The thickness of the orifice plate shall be not less than shown in Table M-5.

Three types of pressure connections may be used: vena contracta taps, 1-D and $^1/_2$-D taps, and flange taps. Appropriate discharge coefficients have to be used in each case.

Flow nozzle. The flow nozzle should be of either the low- or the high-ratio long-radius type as shown in Fig. M-23. The diameter ratio β should be from 0.15 to 0.80, although *ASME Power Test Codes: Compressors and Exhausters* (page 22) recommends the β range between 0.40 and 0.60.

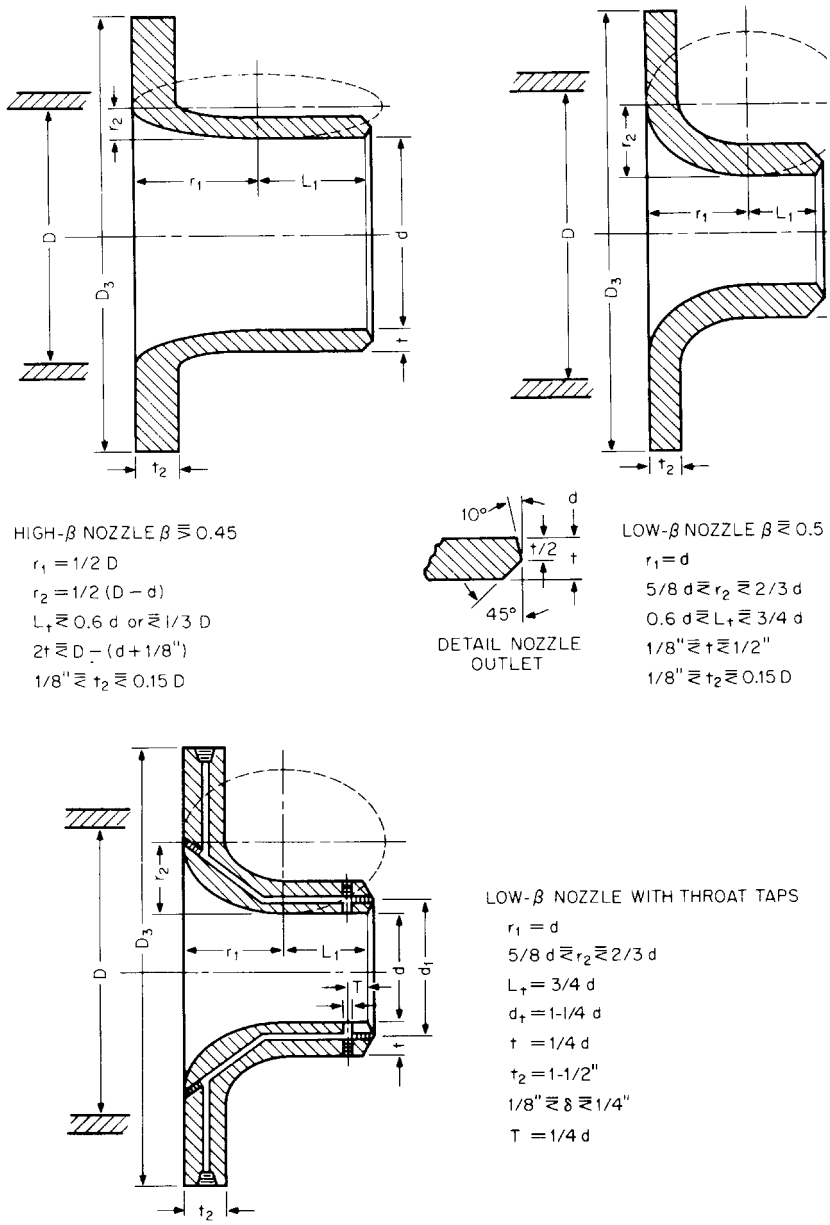


FIG. M-23 Recommended proportions of ASME long-radius flow nozzles. (Source: Demag Delaval.)

Different test arrangements for the measurement of subcritical flow with the flow nozzle are shown in Figs. M-24 through M-26.

Venturi tube. The proportions of the standard-form (Herschel-type) venturi tube and its installation are shown in Fig. M-27. The diameter ratio $\beta = d/D$ should be between 0.4 and 0.75 for the best results. Special forms (venturi-nozzle tube for high-pressure feedwater application, venturi-insert nozzle) may be used. They need individual calibration, however, while graphs may be used for the standard form.

TABLE M-5 Minimum Recommended Thicknesses of Orifice Plates

Differential Pressure, in H_2O	Internal Diameter of Pipe, in				
	3 and Less	6	10	20	30
		$\beta < 0.5$			
<1000	1/8	1/8	3/16	3/8	1/2*
<200	1/8	1/8	1/8	1/4	3/8
<100	1/8	1/8	1/8	1/4	3/8
		$\beta > 0.5$			
<1000	1/8	1/8	3/16	3/8	1/2
<200	1/8	1/8	1/8	3/16	3/8
<100	1/8	1/8	1/8	3/16	1/4

* For 1/2-in plate in 30-in line, maximum differential = 500 in.

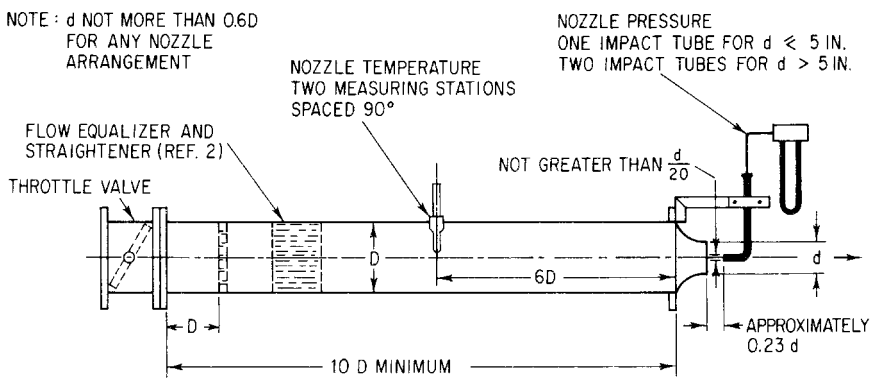


FIG. M-24 Test arrangement *A*. (Source: Demag Delaval.)

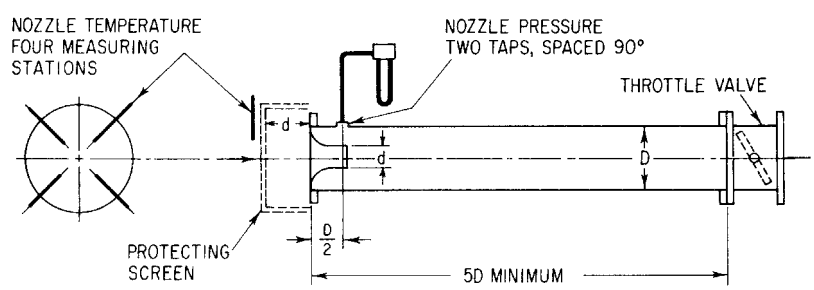


FIG. M-25 Test arrangement *B*. (Source: Demag Delaval.)

Calculation of flow rates

The basic relationships for flow computation were obtained from PTC 19.5, 4-1959, Supplement to *ASME Power Test Codes*, "Flow Measurement of Quantity of Materials." The nomenclature in these relationships was, however, adapted to the more recent terminology of Interim Supplement 19.5, *Fluid Meters*.

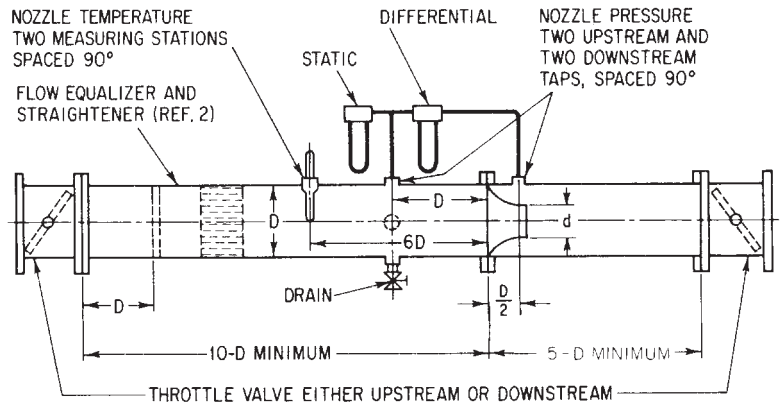


FIG. M-26 Test arrangement *C*. (Source: Demag Delaval.)

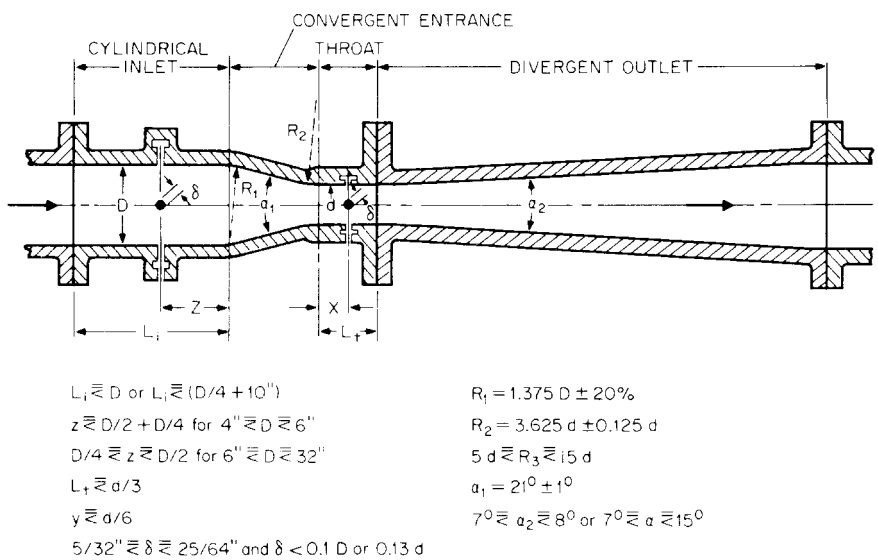


FIG. M-27 Dimensional proportions of classical (Herschel) venturi tubes with a rough-cast convergent-inlet cone. (Source: Demag Delaval.)

Incompressible fluids. The flow of any liquid through an orifice, flow nozzle, or venturi tube is determined by the following equation:

$$M = \frac{CaE_a n\rho}{\sqrt{1-\beta^4}}\sqrt{2gh} \tag{M-12}$$

With the units in American practice, Eq. (M-12) is written

$$M_h = 359CEd^2F_a\sqrt{h_w\rho} \tag{M-13}$$

Compressible fluids. To compensate for the change in specific weight as the fluid passes through the primary element, the equation must be modified by the expansion factor Y for orifices or Y_a for nozzles.

Orifices

$$M_h = 359CEd^2F_aY\sqrt{h_w\rho} \tag{M-14}$$

Flow nozzles and venturi tubes

$$M_h = 359CEd^2F_aY_a\sqrt{h_w\rho} \tag{M-15}$$

Quite often the flow coefficient $K = CE$ is specified for use in Eqs. (M-13) and (M-14) for orifices.

Delaval applications. With particular reference to the methods used at the information source, the above equations assume the forms shown next.

Venturi tube. For measurement of the flow rate of water at 68°F with a *venturi tube* with mercury used as the manometer liquid and a value of 0.984 used for the flow coefficient, Eq. (M-13) becomes

$$q = 19.79\frac{d^2}{\sqrt{1-\beta^4}}\sqrt{h_0} \tag{M-16}$$

where h_0 = manometer pressure, inHg at 68°F

If water at a temperature other than 68°F is metered, a temperature correction should be made.

The coefficient of 0.984 is applicable for β values from 0.3- to 0.75-in pipes of 4 to 32 in, provided the Reynolds number is greater than 200,000. The *Hydraulic Institute Standards* require a minimum velocity of 20 ft/s in the throat for the rated flow of the pump.

Flow nozzle. For measurement of airflow with *flow-nozzle* arrangement *A* (see Fig. M-24),

$$M_m = 5.983CEd^2Y_a\sqrt{h_w\rho} \tag{M-17}$$

and

$$Q_i = \frac{M_m}{\rho_i} \tag{M-18}$$

where

$$Y_a = r^{1/k}\left[\frac{k}{k-1}\frac{1-r^{(k-1)/k}}{1-r}\right]^{1/2} = \text{adiabatic-expansion factor} \tag{M-19}$$

The formula applies to subcritical flow with pressure ratio P_2/P_1 preferably greater than 0.7.

Unless the nozzle in this arrangement has been individually calibrated, the value of the flow coefficient should be taken as 0.99, provided that the throat Reynolds number exceeds 200,000. For nozzles within the pipe and using pipe taps, the coefficient of discharge for pipe Reynolds numbers above 100,000 varies from 0.980 to 0.993, depending on β. Exact coefficients can also be obtained from Tables II–III-5 in ASME Supplement 19.5 on *Fluid Meters*.

For a critical flow measurement, an arrangement similar to that of Fig. M-24 with static-pressure taps 1 *D* upstream from the pipe exit but without an impact tube is used. Rules of the *ASME Power Test Codes: Compressors and Exhausters* should be used for flow computation.

Table M-6 gives approximate flow rates for different nozzle sizes. It is useful for air tests with flow-nozzle arrangements *A* and *B*, when using a long-radius low-β-series nozzle (see Fig. M-23).

TABLE M-6 Long-Radius Low-Ratio Nozzle

	Approximate Flow Rates, ft^3/min	
d	10 in H_2O	40 in H_2O
2.000	250	500
2.500	400	800
3.000	575	1,150
4.000	1,000	2,000
5.000	1,600	3,200
6.000	2,250	4,500
8.000	4,000	8,000
10.000	6,250	12,500
12.000	9,000	18,000
18.000	20,000	40,000
24.000	36,000	72,000

SOURCE: *ASME Power Test Codes*, PTC 10-1965, reaffirmed 1973, *Compressors and Exhausters*.

TABLE M-7 Values of Discharge Coefficient *C* for 8-in Pipe (Velocity-of-Approach Factor Not Included)

R_d / β	Reynolds Number					
	14,000	25,000	50,000	100,000	500,000	1,000,000
	Flange Taps					
0.250	0.6190	0.6091	0.6028	0.5997	0.5971	0.5968
0.500	0.6443	0.6259	0.6142	0.6084	0.6037	0.6032
0.750	. . . . .	0.6749	0.6351	0.6153	0.5993	0.5973
	Taps at 1 D and $\frac{1}{2}$ D					
0.250	0.6036	0.6012	0.5990	0.5975	0.5955	0.5950
0.500	0.6150	0.6112	0.6079	0.6055	0.6024	0.6017
0.750	. . . . .	0.6337	0.6254	0.6196	0.6118	0.6099
	Vena Contracta Taps					
0.250	0.6071	0.6035	0.6003	0.5981	0.5952	0.5945
0.500	0.6185	0.6136	0.6094	0.6064	0.6024	0.6014
0.750	. . . . .	0.6318	0.6240	0.6185	0.6112	0.6095

Orifice. Equation (M-13) is used for measuring the flow rate of water with orifice and Eq. (M-14) for measuring the rate of flow of air. Interim Supplement 19.5 on *Fluid Meters* gives the tables of the values of discharge coefficient C for various pipe diameters as a function of diameter ratio β and pipe Reynolds number R_d based on orifice diameter d. Excerpts from the tables in the Supplement are given in Table M-7 for the three types of pressure-tap locations. The values below the stepped line are extrapolations and are subject to larger tolerance as given in Table II–V-1 in the Supplement.

References and Additional Reading

Temperature

1. *ASME Power Test Codes*, part 3, PTC 19.3-1974, *Temperature Measurement*.
2. Benedict, R. P., *Fundamentals of Temperature, Pressure, and Flow Measurements*, 2d ed., John Wiley & Sons, Inc., New York, 1977.

Pressure

1. *Meriam Standard Indicating Fluid Bulletin*, IM-11.
2. *ASME Power Test Codes: Instrument and Apparatus*, part 2, *Pressure Measurement*, American Society of Mechanical Engineers, New York, 1964.
3. Beckwith, T. G., and N. Lewis Buck, *Mechanical Measurements*, Addison-Wesley Publishing Company, Inc., Reading, Mass., 1961.
4. Benedict, R. P., *Fundamentals of Temperature, Pressure, and Flow Measurements*, 2d ed., John Wiley & Sons, Inc., New York, 1977.
5. Consolidated Electrodynamics, *Transducer*, General Catalog Bulletin 1322, December 1965.

Flow

1. *Fluid Meters*, part II, *Application*, Interim Supplement 19.5, *ASME Power Test Codes: Instruments and Apparatus*, 6th ed., 1971, pp. 179–256.
2. *ASME Power Test Codes*, PTC 10-1965, reaffirmed 1973, *Compressors and Exhausters*.
3. Supplement to *ASME Power Test Codes: Instruments and Apparatus*, PTC 19.5, 4-1959, chap. 4, "Flow Measurement of Quantity of Materials."
4. *Hydraulic Institute Standards*, 13th ed., 1975, pp. 64–68.

Metallurgy; Metallurgical Repair; Metallurgical Refurbishment

This is another of those fields that merits its own set of handbooks: one each for welding, brazing, coatings, laser repair work, and so on. What follows is just enough information to give the process engineers some idea of how much might be possible in this field, including when they could question component lives as specified by an original equipment manufacturer (OEM) and ask for improvements in their life-cycle assessments (LCAs) (*see* Life-Cycle Assessment). Process engineers can also use the material here for direction on how to proceed with questions to ask a metallurgical repair facility. This is an area that offers considerable room for cost economics optimization. The cost of a standard overhaul for a 20,000-hp gas turbine can easily top $1,000,000, especially if the customer does not maintain a parts pool (storage of critical parts) and recondition or repair parts as much as possible. Over 50 percent of repair costs can be saved with careful and correct use of modern metallurgical repair technology.

At one point in the gas turbine's history, a keen demarcation existed between industrial (be they for power generation or mechanical drive) and aircraft gas turbines. That division gets very hazy with some models, particularly those of manufacturers who make both aviation and land-based machines, such as General Electric and Rolls Royce. Metallurgy becomes particularly quite similar between, for instance, a GE LM2500 (aeroderivative) and a GE Frame 9E (industrial) and a GE CF6-80C2 (aircraft). Rolls Royce makes land-based Trents for power generation and aviation Trents and so forth. While there are differences, there are more important similarities, metallurgically speaking. Typical aeroderivative engines are illustrated in Figs. M-28 (Rolls Royce Avon) and M-29 (GE LM2500). The use of cooling passages (technology transferred from aircraft to aeroderivative engines) helps raise the turbines' TITs (see Fig. M-30).

There is also a sort of hybrid turbine category with manufacturers, such as Siemens, that are officially a land-based machinery manufacturer, but have a technology-sharing agreement with Pratt and Whitney. Pratt, which manufactures aviation turbines as well, also uses its aviation technology where the company believes it might benefit users (and economize on design costs) in their land-based models. The consequences for repair technology are that overhaul shops acquire expertise and equipment at a far higher level of sophistication than they did only 10 years ago. Also, now most refurbishment facilities do both aviation and land-based work to optimize the size of their client base.

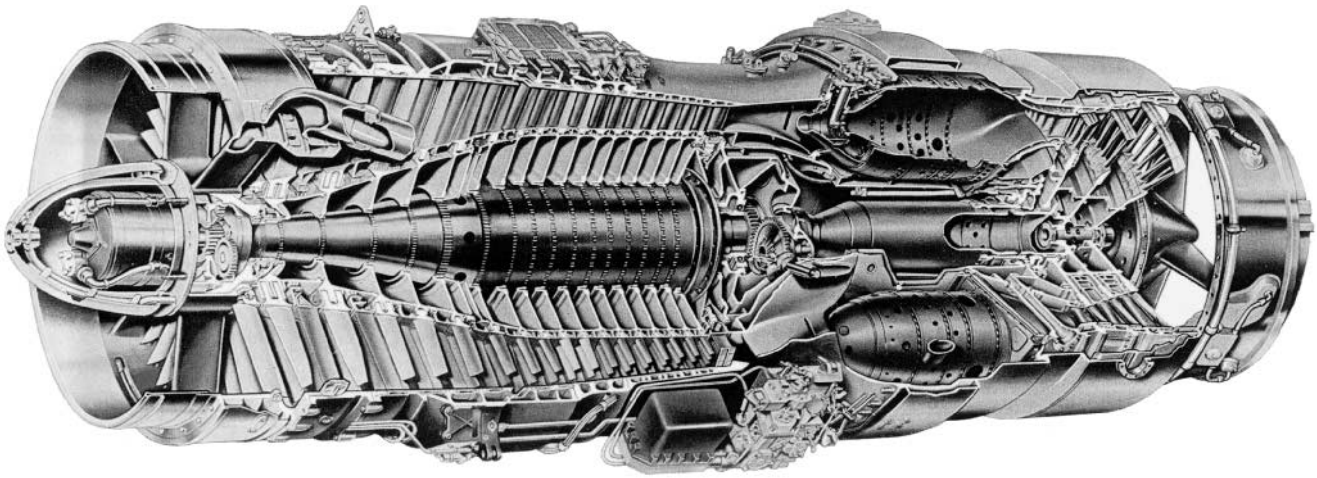


FIG. M-28 Rolls Royce avon aeroderivative gas generator (1535). (Source: Rolls Royce.)

The implications for turbine maintenance are that if the shops use their new technology well, the longevity of parts can be extended, as can time between overhauls. Properly used, sophisticated repair technology can be used to promote the on-condition philosophy that end users are learning to value increasingly. This fact is best illustrated by looking at the nature of some of these repair processes.

The new consciousness regarding environmentally friendly processes sometimes adds initial capital cost, and end users have to bear their share. However, their overall costs per fired hour drop over a longer period, as better repair techniques mean that the cost of "repeat" repairs (done when a process doesn't work right the first time) drops sharply.

What follows is a summary of some of the most basic techniques available.

Repair/Refurbishment Processes*

Brazing

Activated diffusion healing (ADH): This involves manipulating the mix powdered alloys and repair process temperature.

Partitioned allow component healing (PACH): This uses higher temperatures than ADH and is used on single crystal and directionally solidified materials.

Cleaning and stripping

"Environmentally friendly" is a key term for this stage of repair, which formerly had problems with carcinogen content and other types of hazardous chemical cleaners.

---

* Source: Liburdi Engineering, Canada, or as stated.

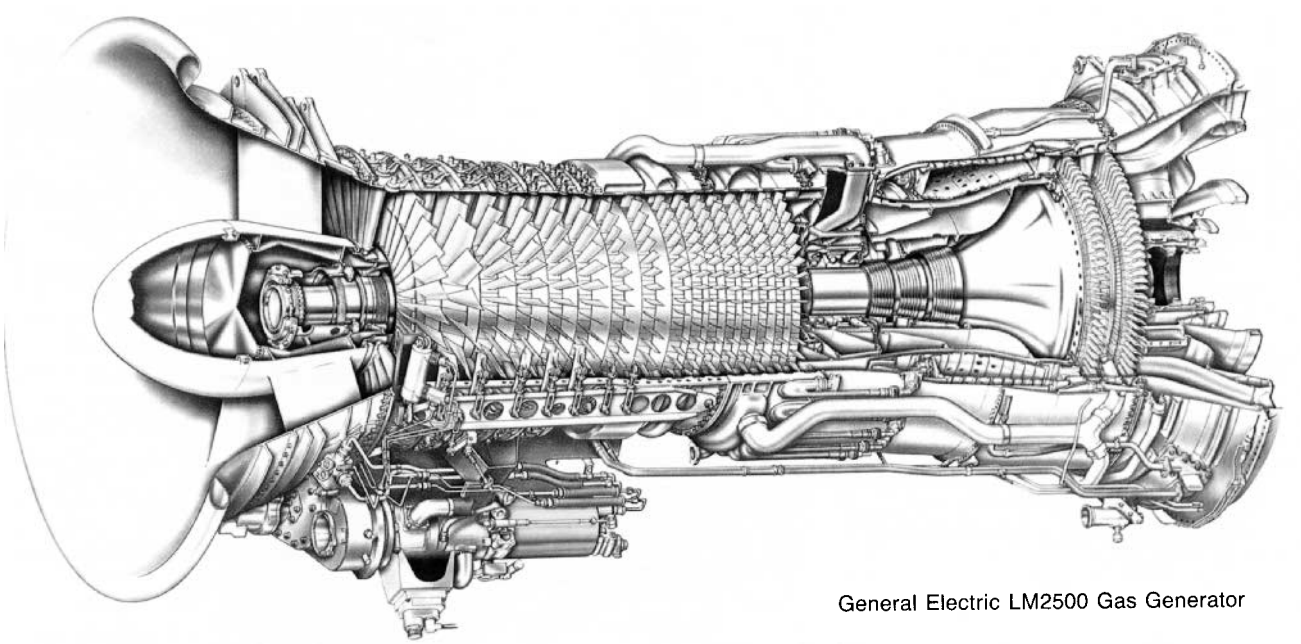


FIG. M-29 General Electric LM2500 gas generator. (Source: General Electric.)

DESIGN FEATURES

Compressor
Type: Axial
No. Stages: 17
Pressure Ratio: 9 to 1
Air Flow: 76 kg per sec ec
Design Speed: 7,550 rpm

Combustor
Type: Cannular
No. Combustors: 8
Ignition: High energy igniters
No. Igniters: 2
No. Fuel Nozzles: 8

Compressor Turbine
Type: Impulse/Reaction
No. Stages: 3
Cooling: 1st & 2nd vanes
1st blades

Bearings
Journal: Roller and ball
Thrust: Ball
Starter
Basic: External drive, motor supplied by packager
Optional: G.G. mounted electric motor

Control System
Supplied by packager

Dimensions
Dry Weight: 1,600 kg
Volume: 3.5 × 1 × 1 m

MATERIALS OF CONSTRUCTION

Compressor
Casing, Front . . . Aluminum Liner . . . Nimonic 263
Casing, Rear . . . Carbon steel
Rotor Blades Stage 00 . . . Titanium alloy
Stage 3–15 . . . Titanium alloy
Stage 0–2 . . . Aluminum alloy
Stator Vanes Stage 00 . . . 12% Cr steel
Stage 5–15 . . . 12% Cr steel
Stage 0–4 . . . Aluminum alloy
Discs . . . Ferritic heat-resistant steel

Combustor
Casing . . . Medium carbon steel

Compressor Turbine
Discs . . . Ferritic heat-resistant steel
1st Vane . . . X-40
2nd & 3rd Vane . . . Nimonic 242
1st Blade . . . Inco 738
2nd Blade . . . Nimonic 115
3rd Blade . . . Nimonic 105

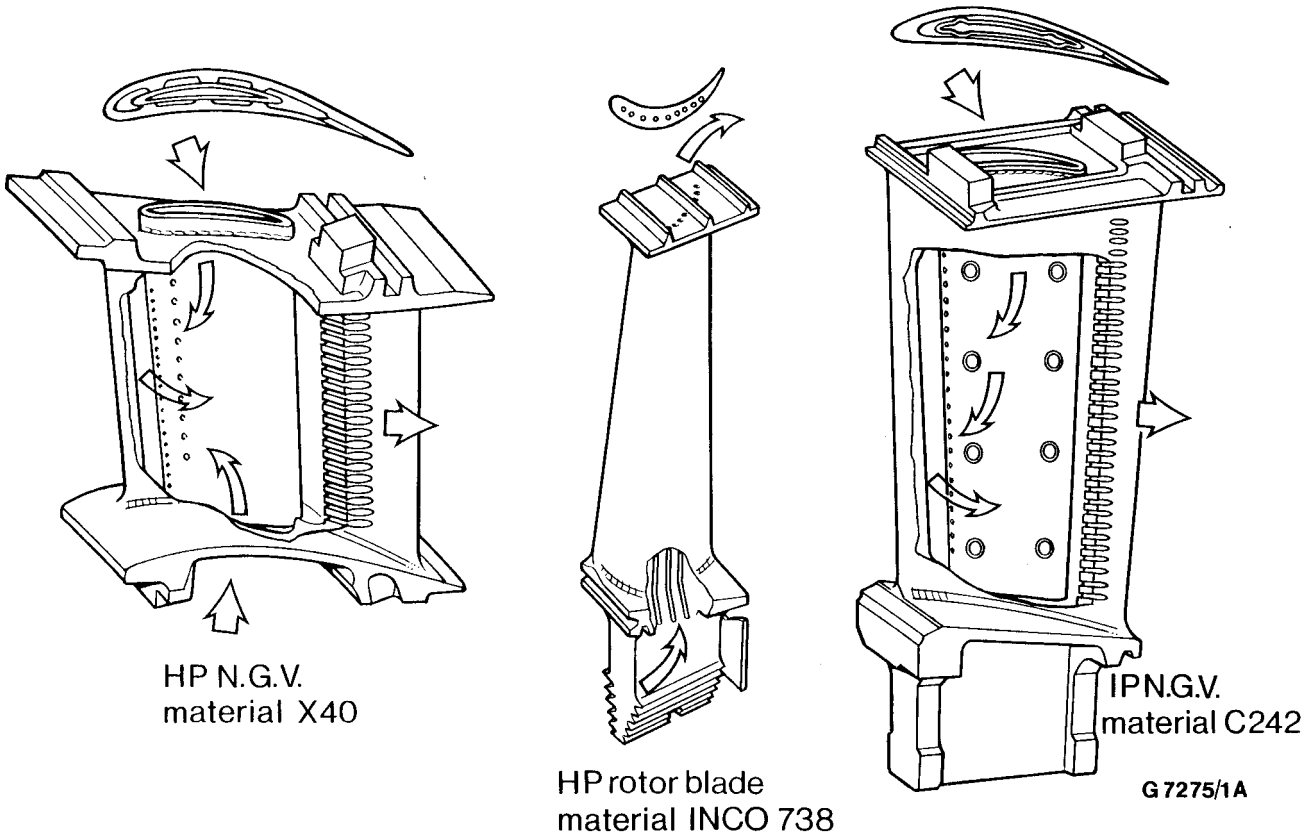


FIG. M-30 Avon Mk. 1535 turbine cooling air. HP = high pressure, N.G.V. = inlet guide vane, IP = intermediate pressure. (Source: Rolls Royce.)

- Plastic cleaners: Clean, recyclable physical abrasives strip contaminants without damaging the parent metal. Note that shot peening ("old" technology) did, especially if the shot was metallic. Glass-bead peening was less damaging, but both "old" peen processes inserted some amount of temporary residual stress, desirable or otherwise. New metallurgy and more sophisticated repair processes can replace conventional peening with beneficial effects on parts' integrity and longevity.
- Autoclaving: Parts are subjected to high pressures and temperatures for superior cleaning.
- Fluoride ion process: This process removes aluminum oxides from parts' surfaces.
- Laser cleaning: Lasers are very accurate and can remove coating or paint layers with great precision and relatively little temperature increase.
- Water jet: This process is also very precise for removing items such as coatings. It can also cut through 10 cm of steel if necessary.

Coatings

Typical examples include:

- Physical vapor deposition (PVD): This is a ceramic coating process and is not required except in the case of high turbine-inlet temperature (TIT) designs. It requires a vacuum.

- Air plasma: Like PVD, except a vacuum is not required.
- Thermally densified coatings (TDC). Generally used to protect shrouds. PVD and TDC are currently used mainly on aircraft engine components, although that will change as OEMs increase the market share of aeroderivatives in land-based use.
- High-velocity oxygen fuel (HVOF): This can be used on bearing surfaces and has replaced hazardous chromates.
- Vapor coatings: Platinum bond coats that provide surfaces that can meet aircraft airfoil and therefore land-based standards. These are nonenvironmental polluting processes.

Machining

Mills, lathes, and grinders have been the conventional workhorses for this process for decades. They are slow and can build up heat in the component if mismanaged. They often need expensive computer numerical control (CNC) to ensure adequate accuracy.

- Water jet: See the same process under Cleaning above.
- Laser machining: See the same process under Cleaning. The time saved is enormous. A component that might have taken 2 to 3 days to machine conventionally can be done with laser technology in half an hour. The initial cost of a laser machining rig is high and end users will bear some part of the initial capital cost, but with the cost of being off-line while they wait for their parts, they generally don't care about any increase in repair cost. There may be none once the time factor is considered.
- Laser drilling: The most accurate kind of drilling, generally used for manufacture of cooling holes on internally cooled turbine blades. However, it is accurate enough that it can be used in overhaul to "redrill" (open up clogged holes) in the same components.
- Electrical discharge machining (EDM): Somewhat older technology than laser drilling and much cheaper, but still fairly accurate for processes such as drilling small holes in components.

Nondestructive test/inspection (NDT/NDI)

Radiographic inspection. Radiographic inspection uses radiation such as X rays or gamma rays to penetrate the material and produce an image of the component on photographic film that can be evaluated and permanently recorded. The exposure time is carefully controlled to produce an image or silhouette of the component that varies in density or shading according to the amount of radiation that reaches each area of the film. When defects are present, the amount of radiation reaching the film is altered resulting in a change in film density directly below the defect location. This enables the radiographer to determine the location and size of the defect. However, narrow defects that are oriented perpendicular to the radiation beam will not be detected because they do not significantly alter the beam transmission.

Turbine manufacturers generally rely on radiographic inspection to control the internal quality of critical castings and welds. Radiography can locate internal defects such as gas porosity, nonmetallic inclusions, cracks, hot tears, shrinkage cavities, lack of fusion, and core shifts. Some of the components inspected include precision cast blades, vane segments, structural welds, and large housing castings.

Ultrasonic inspection. Ultrasonic inspection relies on the use of sound waves to locate discontinuities in a manner similar to radar. The sound waves are generated by a piezoelectric crystal and transmitted through a coupling medium such as oil or water into the component being tested. The high-frequency sound waves penetrate through the material and behave in a manner similar to light waves. When the beam strikes a defect or the opposite wall of the component, the waves are reflected back and re-collected by the crystal. The sound energy is transmitted back into an electrical signal and the information displayed as impulses on a cathode ray screen. The operator must use his skill and knowledge to properly calibrate and interpret the signals and report on the approximate size and location of any defects.

Ultrasonic inspection is capable of locating both internal and surface-connected defects; however, it will not detect narrow cracks oriented parallel to the beam. Turbine manufacturers employ ultrasonics to qualify the internal soundness of critical forgings such as shafts and discs and to determine wall thickness for cooling passages.

Magnetic particle inspection. Magnetic particle inspection is used to locate surface defects in ferromagnetic materials such as steels. A magnetic field is established in the component by either passing electricity directly through prongs or indirectly by wrapping cables around the part. When the lines of force are interrupted by a surface crack, a magnetic leakage field is produced that will attract magnetic particles. Defects can be located by spreading finely divided ferromagnetic material (either dry or in a liquid fluorescent medium) over the surface of the component and observing where the particles are attracted and held in place. The particles form an outline of the defect revealing its relative size and shape. Defects are best resolved when they are oriented at 90° to the direction of the induced magnetic field and, therefore, components should be inspected in at least two directions. The parts must be demagnetized after inspection.

Magnetic particle inspection is used extensively in the turbine industry to inspect forgings, castings, and welds for surface or near surface defects such as cracks, laps, seams, and grinding marks. Components inspected include steel discs, shafts, compressor blades, and vanes and welds on casings.

Eddy current testing. Eddy current testing involves the interaction between the external magnetic field of the test coil or probe and the internal magnetic field resulting from the induced currents in the components. When a part is scanned by the probe, defects will cause a change in the impedance of the test coil that is displayed as a signal on the cathode ray screen. However, the evaluation and interpretation of the signals require considerable skill. The equipment must be calibrated using a reference sample of the same material with a known defect and the probe must be specially designed for each application to suit the geometry of the part.

Eddy current testing can be used on any conducting material including non-ferromagnetic materials such as superalloys and stainless steels. The process can be very effective in detecting surface and subsurface cracks in blades and discs and is gaining wider acceptance in the turbine industry. Eddy current can also be used to sort materials and measure coating thickness.

Liquid penetrant inspection (LPI). Liquid penetrant inspection is widely used throughout the turbine industry to detect surface cracks. The process appears to be

deceptively simple and is often misused. Surface preparation is particularly important because defects can easily be masked by overpeening.

In the analysis of service-exposed components such as discs and blades, a judicious compromise is often required on the surface preparation. Light glass bead cleaning is often effective in removing surface deposits with minimal loss in sensitivity; however, care must be taken not to overpeen the surface. Where optimum sensitivity is required, such as in rotating blades, the parts should be chemically etched after glass bead cleaning to remove metal peened over the discontinuities.

LPI is designed to locate only discontinuities that are open to the surface. When a penetrant is applied to the surface of a component, it is drawn into surface discontinuities by capillary forces. After the excess penetrant has been removed from the surface, the penetrant trapped by the defects is drawn back out by capillary action and forms a detectable outline of the defect.

Over the years, many penetrants have been developed that vary in sensitivity and application. As indicated in the enclosed information sheets from the manufacturers, the penetrants are divided into water washable and post-emulsifiable (solvent remover) types. Commonly used penetrants include ZL-17B or Ardrox 970-P10 (water washable) and ZL-22A (solvent type). Care must be taken to ensure that the correct combinations of penetrants, removers, and developers are used. In addition, a test should never be repeated using a different penetrant type because complete masking of the defects will occur.

Field inspections are particularly challenging because the inspection conditions are generally not ideal and often the operator does not have all the equipment desired. Although some compromises are often required, the inspector must ensure that the sensitivity and validity of the test are not jeopardized.

Laser techniques. These can be used for inspection, calibration, and highest-resolution surface mapping.

Computer-aided topography (CAT) scan. CAT scanners locate internal defects in engine components. It is particularly useful for sophisticated components such as blades and airfoils with internal passages and cavities.

Powder metallurgy

This involves molding of powdered alloy flux (somewhat like plasticine) to damaged areas of a component (blades and vanes typically) and use of elevated temperatures to make that flux integral with the parent material. Some sophisticated repair shops develop their own equivalent process and do the work under license to OEMs or sell the OEM (and other facilities) the right to use the process.

Welding

Traditional weld repairs could result in component warpage, which then requires specialized heat treatment and associated jigs, which, in turn, extends repair times and costs, with severe consequences for overall turbine maintenance.

- Laser welding: Extremely accurate and much less heat intensive than conventional repair, laser welding is particularly useful for turbine blades, compressor blade leading edges, and other sensitive components.

- Weld repair using robotics*: The automation of turbine blade welding provides both metallurgical benefits and production advantages. Heat-affected zone cracking in sensitive superalloys, such as IN738 and IN100, can be eliminated or greatly reduced by optimizing process control, and higher production yields can be achieved when welding jet engine blades. However, the successful implementation of automated processes requires careful consideration and engineering of the technology package. In particular, the equipment packager must be experienced in the technology associated with turbine blade welding and incorporate appropriate tooling, measurement system, power source and robotic controls.
- Superalloy welding at elevated temperatures (SWET): OEMs often develop their own proprietary process for this technology that is commonly used on superalloy or directionally solidified materials such as turbine blades.
- Dabber TIG (tungsten inert gas): A slightly older process that uses TIG to rebuild knife edge seals with minimal heat warpage.
- Plasma transfer arc: Similar to dabber TIG and used for the same components.

The exception to the "land-based turbine design approaches aircraft engine technology standards" is evident in certain OEM models, such as Alstom's GT11N2 and GT35. However, that is because both these models are designed to take the punishment meted out by vastly inferior fuels or just be conservative enough to require less training for end-user operators and maintenance staff. Alstom also makes sophisticated models with metallurgy that will match those of the "dual" (both aircraft engine and land-based engine) OEMs for users with different requirements and less punishing fuels. However, Alstom also contracts powder metallurgy repair, for instance.

All OEMs can enhance their "benefits to end user" objectives from some for the preceding techniques, such as the ultimate time-saver laser machining. It ultimately depends on the specifics of an end user's application. The potentially usable repair techniques on any end user's selection play a huge role in determining the turbine model's overall maintenance costs and therefore the ultimate crux of gas turbine selection, called "total costs per fired hour."

Basic Fundamentals of Materials

The properties of the materials used in gas turbines are determined by their composition and their prior processing and service history. To understand how these factors work to govern alloy behavior, a basic understanding of some fundamental principles of materials engineering is useful. This is largely a question of understanding some of the terms used by metallurgists in describing material behavior.

Turbine materials are governed by the laws of thermodynamics, which basically means that changes that take place in the materials result in a reduction of the energy state in the material. We often speak of the *equilibrium* or *stable* condition of a material; this simply means the condition of lowest energy.

Given infinite time, all materials would end up in their equilibrium condition. In

* Source: Adapted from extracts from Lownden, Pilcher, and Liburdi, "Integrated Weld Automation for Gas Turbine Blades," Liburdi Engineering, Canada, ASME paper 91-GT-159.

practice, there are *kinetic* barriers to achieving equilibrium and most materials are used in a *metastable* condition. The most common kinetic barrier is the rate of *diffusion* (i.e., the speed at which atoms in a solid material can rearrange themselves). Almost all of the metallurgical reactions that occur in turbine materials occur at rates governed by the speed of diffusion. Examples include the rate at which a coating interdiffuses with the base metal or the rate at which strengthening particles grow in an alloy.

All of the materials used in gas turbines are *crystalline* in nature. This means that the atoms of the elements that make up the alloy are arranged in regular periodic arrays or lattices, with each atom occupying a site in the array. When we refer to grain or crystal *orientation*, we are referring to the direction relative to this crystal lattice. The mechanical and physical properties of materials depend on their orientation.

In real materials, the crystals are not perfectly periodic, but contain various lattice defects. Two of the most important of these are *dislocations* and *grain boundaries*. Dislocations are an important class of planar defects, since their presence within crystals leads to plastic deformation behavior. Most materials are not used as a single crystal, but as polycrystals that consist of many individual crystals with different orientation called *grains*. The interfaces between the individual grains are *grain boundaries*. The size and degree of orientation between grains and the nature of the grain boundaries are important in determining the properties of a material.

Metallurgists control the nature of grains by the processing performed during manufacture. In a polycrystalline material, grains will increase in size at elevated temperatures and thus *grain growth* will occur during high temperature heat treatments. Grain sizes can be reduced by introducing plastic work into a material. At high temperatures, the resulting strain energy drives the process of *recrystallization*, which results in the formation of smaller grains during heat treatments or hot-working operations.

Engineering materials are almost exclusively mixtures of two or more elements, which are called *alloys*. Alloying elements can dissolve in the matrix of the principal elements to form a *solid solution*, in which the dissolved element is randomly distributed in the crystal lattice. The alloying elements can also react with the matrix to form a *compound* that has a specific arrangement of atoms of each element.

Commonly both solid solutions and compounds will coexist within the same material as different *phases*. The stability of specific phases within a given alloy system varies with the composition and the temperature. Kinetics also determine which phases form within an alloy. Many reactions are sluggish enough that the stable phase may not form initially and the alloy may exist in metastable condition for some length of time. By using chemistry and heat treatment to control the phases formed by an alloy, metallurgists can alter the strength of materials.

Three principal types of deformation take place upon the application of loads to turbine materials. *Elastic* deformation is instantaneous reversible deformation that results from the distortion of the crystal lattice. *Plastic* deformation is the irreversible deformation that takes place instantaneously through the movement of dislocations through the crystal matrix. *Creep* deformation takes place by a variety of diffusion-controlled processes over time, resulting in continuing strain under the applied load.

At sufficiently high loads or after a critical amount of deformation has taken place, *fracture* of a material will occur. Fracture can be broadly classed as ductile or brittle. In turbine materials under most conditions, fracture occurs by the

formation and linkage of internal cavities formed either by creep or plastic deformation.

When cyclic loads are applied to a material, cracking and fracture may occur by the process of *fatigue*. On a microscopic scale, fatigue occurs by localized plastic deformation, resulting in the initiation and growth of macroscopically brittle cracks.

The resistance of a material to deformation and fracture depends on its composition and *microstructure*. The size of grains, the size and distribution of second phases, and the effects of alloying elements on the crystal lattice of the matrix all influence the mechanical behavior of the alloy.

Exposure of alloy surfaces to operating conditions results in surface reactions between the alloy and the environment. *Oxidation* and *hot corrosion* occur at elevated temperatures through direct reaction with oxygen and other environmental contaminants. *Aqueous corrosion* occurs in wet environments through dissolution reactions.

Material Selection for Gas Turbines

This subsection is with specific reference to gas turbine materials, the most severe thermal application in a plant.

The materials used in gas turbines or jet engines span the range of metallurgical alloys from high-strength steel, to lightweight aluminum or titanium, to temperature-resistant nickel or cobalt superalloys. In a gas turbine, the temperatures can vary from ambient to gas temperatures in excess of the melting point of superalloys and, therefore, the materials in the different sections must be selected on the basis of their capability to withstand the corresponding levels of stress and temperature. The following summary outlines the materials used in the different components of the gas turbine, along with a rationale for their selection.

Compressor rotor

The temperature in a typical compressor will range from ambient to approximately 800°F (425°C). The discs and blades rotate at high-speed and are, therefore, highly stressed and subjected to aerodynamic buffeting or fatigue. In industrial turbines, the discs are generally made from high strength alloy steel and the blades from martensitic stainless steel. However, in jet engine derivatives, lighter materials such as aluminum and titanium are used for the blades and vanes in the front of the compressor. In some cases, the last stages of the compressor can run significantly hotter and more creep-resistant materials must be used such as A286 and IN718.

Turbine discs

Turbine discs are highly stressed in the rim area where the blade root attachment occurs and in the hub of bored discs where high burst strength is required. The discs are forged from high-strength steels in advanced industrial turbines and iron or nickel base superalloys such as A286 and Inconel 718 for the jet engines. The disc rim is generally isolated from the hot gas path and cooled to as low as 600°F (315°C) for alloy steel discs to ensure adequate material strength and creep resistance.

Combustion cans

The flame temperature in a burner generally exceeds 3000°F (1650°C). The temperature is moderated by mixing with cooler compressor discharge air that flows

around the combustion chamber and through the slots in the walls to keep metal relatively cool [approximately 1500°F (815°C)]. The combustor cans are generally fabricated from nickel base sheet superalloys such as Hastelloy X, Nimonic C263, and Inconel 617. These alloys have good weldability and oxidation resistance.

Turbine vanes

The stationary vanes in a turbine act as guides for the hot gas to ensure that it enters the blade's airfoil at the right angle and with minimal pressure loss. The applied stresses are generally low; however, they are subjected to turbine inlet gas temperatures which, in some engines, exceed the melting point of the material [2500°F (1370°C)]. The vanes are generally made from cobalt base alloys such as FSX414 and X45, which have good castability and excellent oxidation and thermal shock resistance. In advanced designs, the vanes are cast with integral cooling passages to reduce the metal temperature. The cobalt base alloys are generally weldable and minor weld repairs are often allowed. In some designs, the cobalt base alloys have been replaced with more creep-resistant nickel base alloys such as IN738, Rene 80, and IN939, which are significantly harder to weld repair.

Turbine blades

Turbine blade airfoils are subjected to the most severe combination of applied stresses due to centrifugal and bending loads and high temperatures. The blade materials must have excellent strength and creep resistance, as well as oxidation resistance. In advanced units, the blades are cooled by internal passages to moderate the metal temperature and improve blade life. The blades are generally precision forged or cast from nickel base alloys such as Udimet 520, IN738, Rene 80, and Mar M247 and can be manufactured as either equiaxed, directionally solidified, or single crystal castings. These materials have poor weldability and repairs must be approached with extreme caution.

Service life for turbine components

Once in service, critical gas path components require care and attention to optimize their life potential. Often, turbine users choose to abdicate their responsibility in this critical area and elect to rely solely on the manufacturer for guidance. However, this can lead to premature component replacement of failures if components are neglected or improperly assessed.

A gas path component management program basically involves the detailed characterization of components at established intervals. For example, during major overhauls, representative blades are removed and destructively tested for corrosion, microstructure, and remaining creep life. The data are tabulated and a life trend curve established for the material. This will provide the user with specific information on how the engines are standing up rather than rely on someone else's data and provide advance warning of impending problems with corrosion or creep.

Creep damage can be detected in turbine blades by judicious testing of samples from the airfoil and by metallurgical inspection. With prolonged service exposure at high temperature and stress, cavities are formed at the grain boundaries of the material that, with time, will grow in number and size and eventually join to form a crack. However, if creep voiding can be detected prior to surface crack formation, the parts can be rejuvenated by hot isostatic pressing (HIP).

During HIP, the parts are subjected to a combination of pressure and temperature that collapses any internal cavities and diffusion bonds the surfaces back together. Tests have shown that, in most materials, HIP is capable of fully restoring the properties and, therefore, offers the opportunity to recycle and extend the life of serviced blades.

Steam-turbine metallurgy

Steam turbines operate in far less demanding service than gas turbines as the following list of typical steam turbine materials illustrates.

Typical steam turbine materials

Part Identification	Material	Commerical Equivalent
Steam rings — 650 psig/750°F	Carbon steel	ASTM A216 grade WCB
— 900 psig/950°F	Alloy steel	ASTM A217 grade WC6
— 1500 psig/950°F	Alloy steel	ASTM A217 grade WC9
Valve chests — 650 psig/750°F	Carbon steel	ASTM A216 grade WCB
— 900 psig/950°F	Alloy steel	ASTM A217 grade WC6
— 1500 psig/950°F	Alloy steel	ASTM A217 grade WC9
Cylinders	Carbon steel	ASTM A216 grade WCB
Nozzle blocks	Carbon steel	ASTM A212 grade A
	Stainless steel	AISI 416
Sovernor valve	Stainless steel	AISI 410
Sovernor valve stem	Stainless steel	AISI 410
Shafts	Alloy steel	AISI E4340
Discs — standard forged	Alloy steel	ASTM A294 class 5
— integral with shaft	Alloy steel	ASIS E4340
Blading	Stainless steel	AISI 403
Bearing brackets	Carbon steel	ASTM A216 grade WCB

Manufacture

Superalloys

The term superalloy is popularly used to define a material having structural strength above 1000°F (540°C). The development of these temperature-resistant materials in the early 1940s was a primary catalyst for jet engine development. The two technologies have been interrelated ever since and, as the temperature capability of materials was improved, so did the turbine efficiency. The enclosed graph illustrates the strength improvements obtained through alloy development and how the chemistry and microstructure became more complex with each evolution.

Basically, superalloys can be categorized into three chemical families: iron base, nickel base, and cobalt base. All alloys contain additions of chromium and aluminum for oxidation resistance and variations of aluminum, titanium, molybdenum, and tungsten for strength. The nickel base superalloys offer the highest strength and are generally chosen for rotating blade applications. The cobalt base alloys, although weaker, offer better environmental resistance and are generally chosen for stationary vane applications.

From a metallurgical point of view, all superalloys exhibit an austenitic or γ microstructure that, unlike steel, does not undergo any transformation as it is heated to its melting point. The alloys derive their strength from three principal mechanisms:

1. Solid solution hardening of the γ matrix resulting from the coherency strains or stiffening effect or larger atom elements such as chromium, molybdenum, and tungsten.
2. Precipitation of small γ' particles Ni_3 (Al, Ti) throughout the matrix act as obstacles to dislocations or strain flow. The γ' phases precipitate on cooling from the solution temperature and during aging treatments.
3. The formation of blocky carbides at the grain boundaries act to pin the grain boundary and prevent grain boundary sliding during creep.

Manufacture: Castings and Forgings

There are two main routes by which superalloys are manufactured into turbine components: casting and forging.

Forged parts are made by hot-working cast ingots or powder metallurgy compacts into billet or bar and eventually into the final shape. Since it involves significant plastic deformation, considerable refinement of grain size can be achieved through recrystallization; however, the alloy must also have good ductility at the forging temperature. Consequently the higher strength superalloys cannot be manufactured by forging. Turbine discs, some turbine blades, and compressor components are made by forging.

Cast parts are formed by pouring molten alloy into near-net-shape molds and allowing them to solidify. The investment casting process used allows extremely complex shapes, including cooling passages, to be cast in. Molds are made by depositing a ceramic layer around a wax form and melting the wax to form a cavity, while ceramic cores are used to cast internal passages. Because the process does not involve the deformation of the alloy, ductility is not an important consideration. Turbine blades and vanes are commonly made by casting.

A modification of the investment casting process is used to produce directionally solidified and single crystal components. To produce components in these forms, alloy is cast into molds that are subsequently withdrawn from a furnace at a controlled rate. By controlling the solidification of the casting, the grains are forced to grow in one direction and by using a crystal selector at the base of the casting, the casting can be made as a single crystal. Such components have improved creep and thermal fatigue resistance because there are no grain boundaries oriented perpendicular to the principal stress direction. Because of their high cost, such parts are typically limited to first-stage turbine blading.

Metering, Fluids; Metering Pumps (*see* Fuel Systems)

Mist Eliminators (*see* Separators)

Mixers (*see* Agitators; Centrifuges)

Monitoring (*see* Condition Monitoring)

Motors (*see* Electric Motors)

N

Noise and Noise Measurement (*see* Acoustic Enclosures, Turbine)

Noise Silencing and Abatement (*see* Acoustic Enclosures, Turbine)

Nondestructive Testing (FP1, MP1, X Ray) (*see* Metallurgy)

Nozzles

Nozzles can mean nozzles in the airfoil sense, i.e., inlet guide vanes on gas turbines or steam turbines. *See* Metallurgy.

Nozzles can also be used to eject (*see* Ejectors) or spray. Spray-nozzle applications are too numerous to itemize and must be customized for each application. Spray nozzles in gas-turbine fuel systems, for instance, are typically for one-, two-, or dual-phase fuel streams (gas; gas and liquid; or gas, liquid, and a mixture of gas and liquid). Spray nozzles can also be used extensively in metallurgical processes such as plasma coating.

Increasingly, for uniform flow distribution, spray patterns are CNC system controlled. A robot that sprays plasma is one such example. This robotic CNC or PLC control system is generally customized for most applications.

O

Oil Analysis

Some plants have oil-sample analysis done on oil samples taken from oil drains on their turbomachinery packages. Metallic particulate content is trended for a clue as to what problems may occur. For instance, rising content of babitt may indicate bearing wear and/or incipient bearing failure. The problem of using this technique with rotating machinery is that most of this machinery turns so fast, the machine may fail between sampling analyses. Oil analysis has a far better chance of detecting deterioration in slower reciprocating machines, provided the samples are analyzed expeditiously.

Oil Sands; Synthetic Crude; Tar Sands; Shale

Oil sands and tar sands are synonyms for the same material. Synthetic crude results from processing oil sands. Shale is similar to oil sand in that it is a category of soil/rock that contains oil that can be extracted.

Certain areas of the world have large deposits of oil sands (northern Alberta, Canada) or shale (China and the United States) that oil can be extracted from, either by mining the soil and processing it or directing leaching steam into the ground. The latter process recovers only about 60 percent of the oil. The former process can recover more oil but is expensive to design and build because of the high level of corrosion and erosion problems experienced.

This technology is significant to process engineers in that it provides useful information on what equipment can survive the harshness of this process: such equipment would be suitable for similarly demanding processes elsewhere. Figure O-1 is a simplified line diagram of synthetic crude manufacture from processing oil sands.

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.

Oxygen Analysis

Oxygen analyzers used to have applications in turbine and boiler design; they monitored fuel/air ratios. Now the zirconium oxide probe for oxygen analysis is found to have applications in process operations as well, where turbines are involved. These applications give the probe some predictive solving potential, which most rotating machinery engineers might depend on other indicators (such as vibration analysis) for.

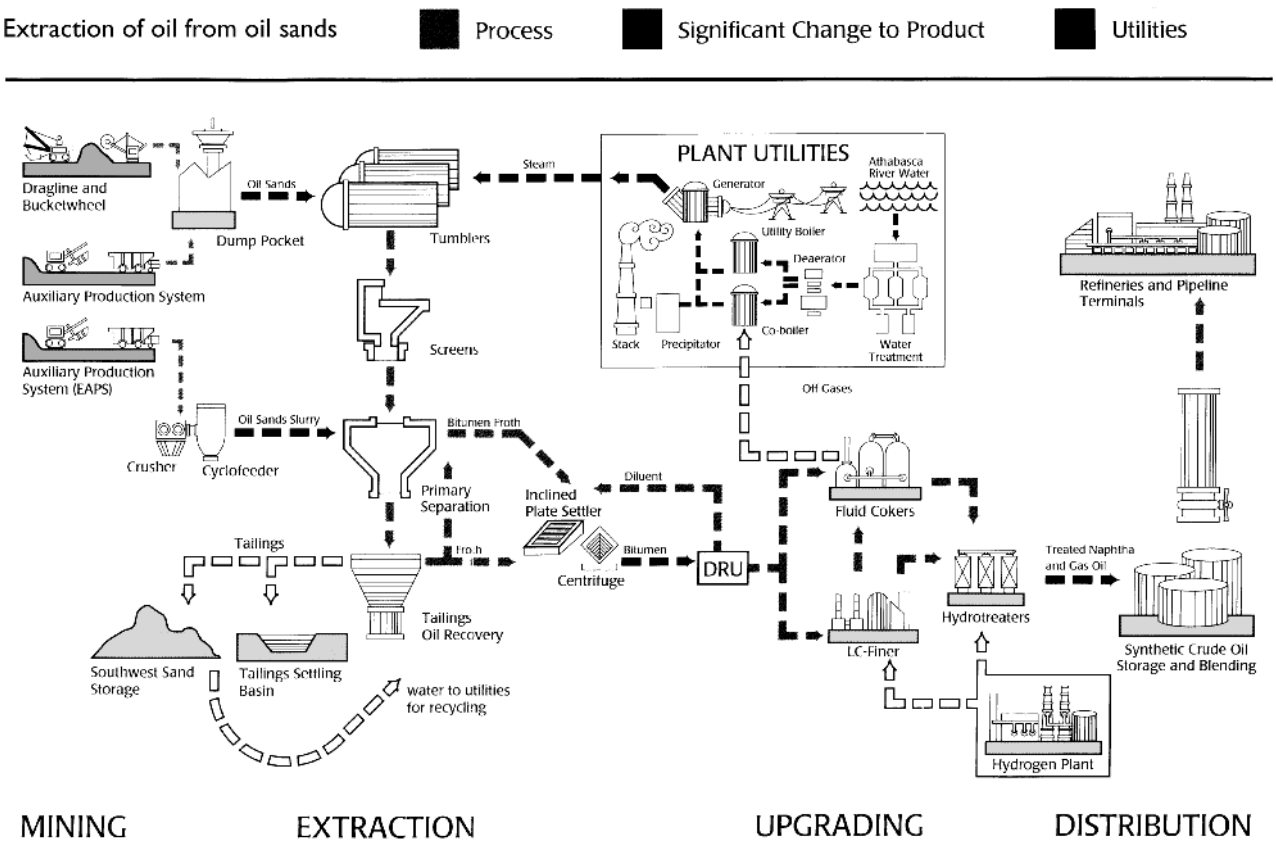


FIG. O-1 The production of synthetic crude oil. (Source: Syncrude Canada Limited.)

Oxygen Analysis Produces Profitable Power Plants*

A recent survey of instrument engineers from European power companies has indicated that many have expanded the utility of the ZrO_2 oxygen probe beyond its traditional use as a tool to optimize fuel/air ratios.

Most operators of large boilers have come to realize that significant stratification can exist in the ductwork carrying flue gases. (See Fig. O-2.) By installing an array of O_2 probes, and averaging the outputs, an average signal is generated that is suitable for use in automatically trimming fuel/air ratios.

While averaging for control purposes is increasingly common, many operators demand a trend display of each individual probe. By watching the relative O_2 values, many operators feel that they can detect problems with fouling at individual burners, slag accumulation, or even problems with coal fines at a pulverizer. The O_2 measurement is increasingly taking on a "predictive maintenance" function.

- *Air heater leakage.* An increasing number of customers are utilizing the O_2 measurement to detect seal leakage at air preheaters. This is particularly effective when utilized with rotating units. Air leakage at the preheater is not only a strong indicator that maintenance is required, it negatively affects thermal heat rate efficiency. See Fig. O-3.

*Source: Adapted from extracts from Simmers, "Oxygen Analysis Produces Profitable Power Plants," *IPG*, January 1998.

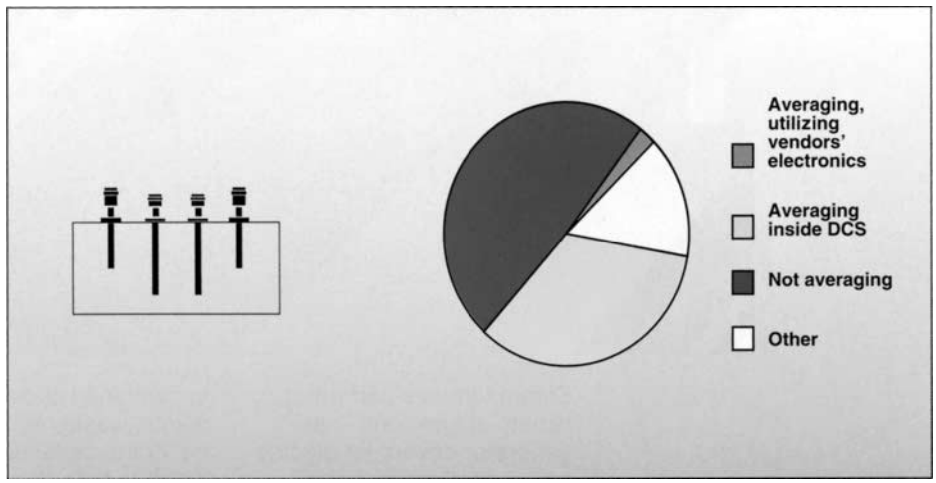


FIG. O-2 Most operators realize that significant stratification can exist in ductwork carrying flue gases. (Source: Simmers.)

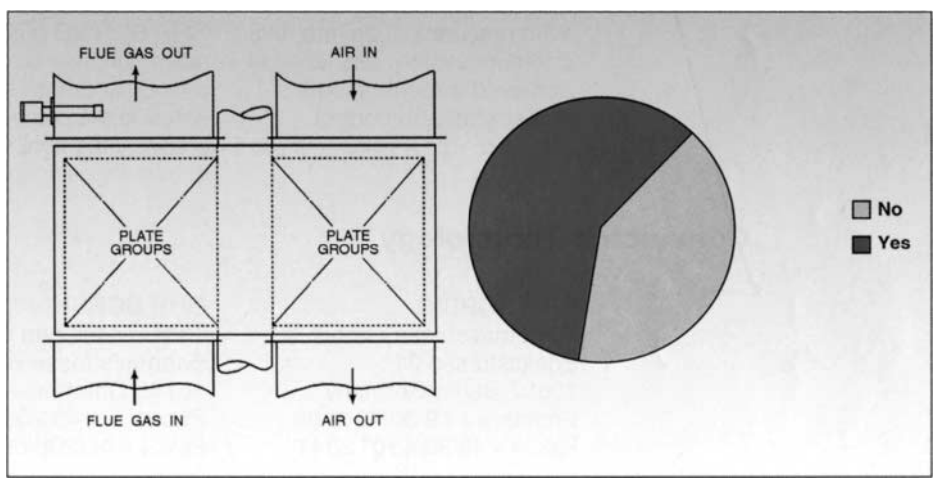


FIG. O-3 Oxygen measurement can be used to detect seal leakage in air heaters. (Source: Simmers.)

- *Flue gas recirculation.* One NO_x reduction strategy is to mix some flue gas into the combustion air prior to the burner, preventing the formation of thermal NO_x. An O_2 analyzer placed downstream of the mixing point can be utilized to maintain a specific final O_2 set-point. (See Fig. O-4.)

  It should be noted that the ZrO_2 sensing cell is mildly sensitive to pressure changes, whereby each 10 mm of H_2O pressure = 1 percent change in reading (not a 1 percent change in O_2). Some windboxes may experience pressures high enough to require a pressure balancing accommodation to compensate.

- *NO_x predictor.* Thermal NO_x is dependent in great part on the amount of O_2 available, as well as temperature at the burner. Increasing numbers of stations are watching O_2 levels as an indication of NO_x production. (See Fig. O-5.)

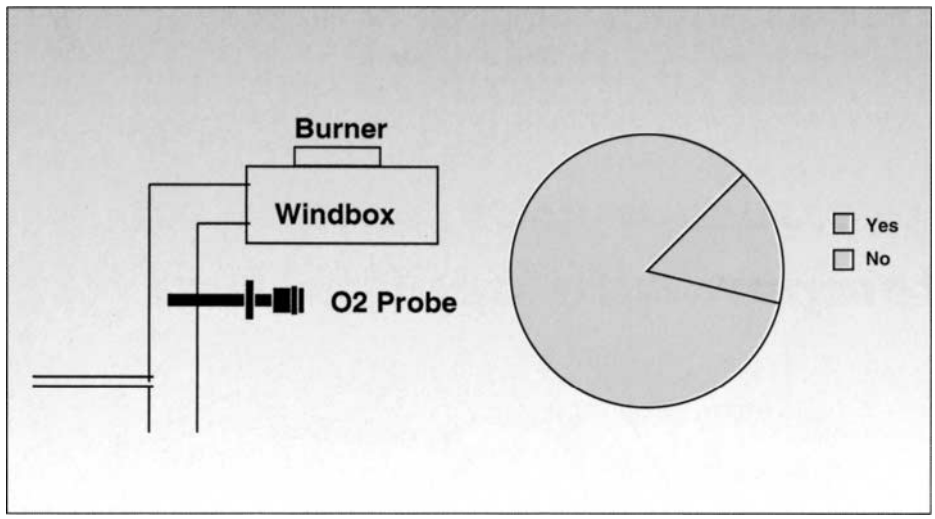


FIG. O-4 Using O_2 measurement to control NO_x formation. (Source: Simmers.)

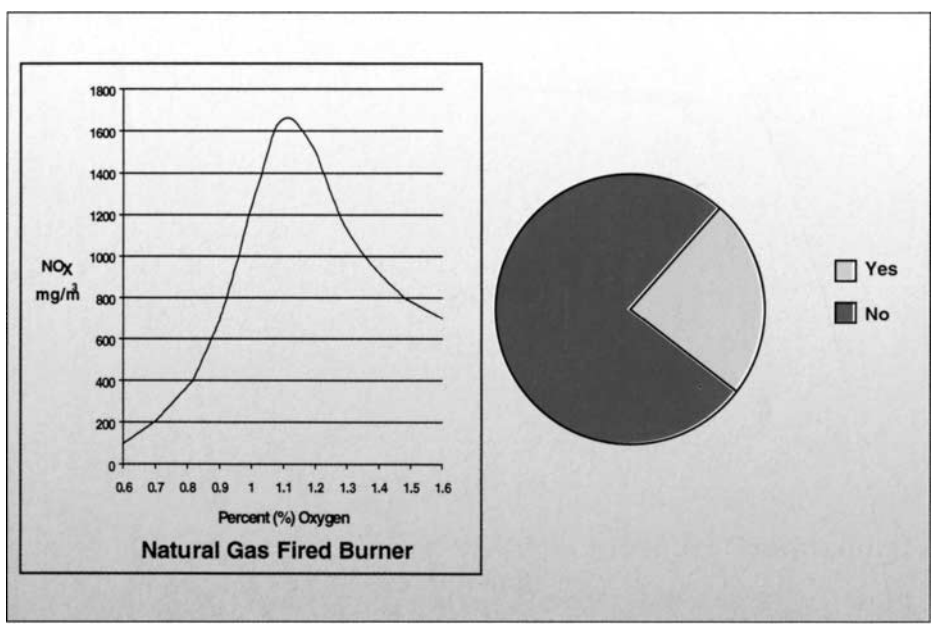


FIG. O-5 Using O_2 to predict NO_x formation. (Source: Simmers.)

Ozone*

Industrial activity and motor vehicle emissions produce ozone, a greenhouse gas. As desired ozone levels are legislated toward, its effects need to be familiar to process and industrial engineers.

Two problems to be considered are ground-level and stratospheric ozone. (As a sample country for data comparisons, Canada has been selected.)

It is important not to confuse the problem of ground-level ozone pollution with

* Source: Environment Canada. Adapted with permission.



FIG. O-6 Urban smog. (Source: Environment Canada.)

the thinning of the stratospheric ozone layer. About 90 percent of atmospheric ozone occurs in the stratosphere, a layer that extends from about 15 to 50 km above the Earth's surface. There it performs the critical function of absorbing harmful ultraviolet (UV) radiation emitted by the sun. At present the stratospheric ozone is thinning and providing plants and animals with less protection from the harmful effects of excess UV radiation than in the past. This is an urgent problem; however, it is not the topic of the fact sheet.

The ozone problem at the Earth's surface is accumulation rather than depletion. The normal state of affairs at ground level is for ozone to form and almost immediately break down, at the same rate at which it is being produced, by releasing one oxygen atom. Figure O-1 shows the chemical cycle involving oxygen (O_2) and two of the nitrogen oxides (NO_x) (in this case, nitric oxide and nitrogen dioxide), sunlight, and high temperatures that governs the formation and breakdown of ground-level ozone. Problems arise when volatile organic compounds (VOCs) (see "Chemical precursors of ground-level ozone" for a description of VOCs) are added to the mix (Fig. O-6).

Because the buildup of ozone at ground level depends upon the concentration of other pollutants, as well as temperature and sunlight, ozone levels usually peak in the late afternoon of hot summer days and can persist into the evening and night. Just how much ozone builds up varies considerably from year to year and from region to region, but summers that are hotter than normal generally produce more episodes of ozone pollution.

Chemical Precursors of Ground-Level Ozone

Ozone is a secondary pollutant in that it is not emitted directly to the atmosphere. Nitrogen oxides and VOCs, both of which are emitted by natural processes and human activities, are called ozone precursors because they must be present for ozone to form (Fig. O-7).

Nitrogen oxides

This group of nitrogen-oxygen compounds includes the gases nitric oxide, nitrogen dioxide, and nitrous oxide. Natural sources of nitrogen oxides are forest fires, lightning,

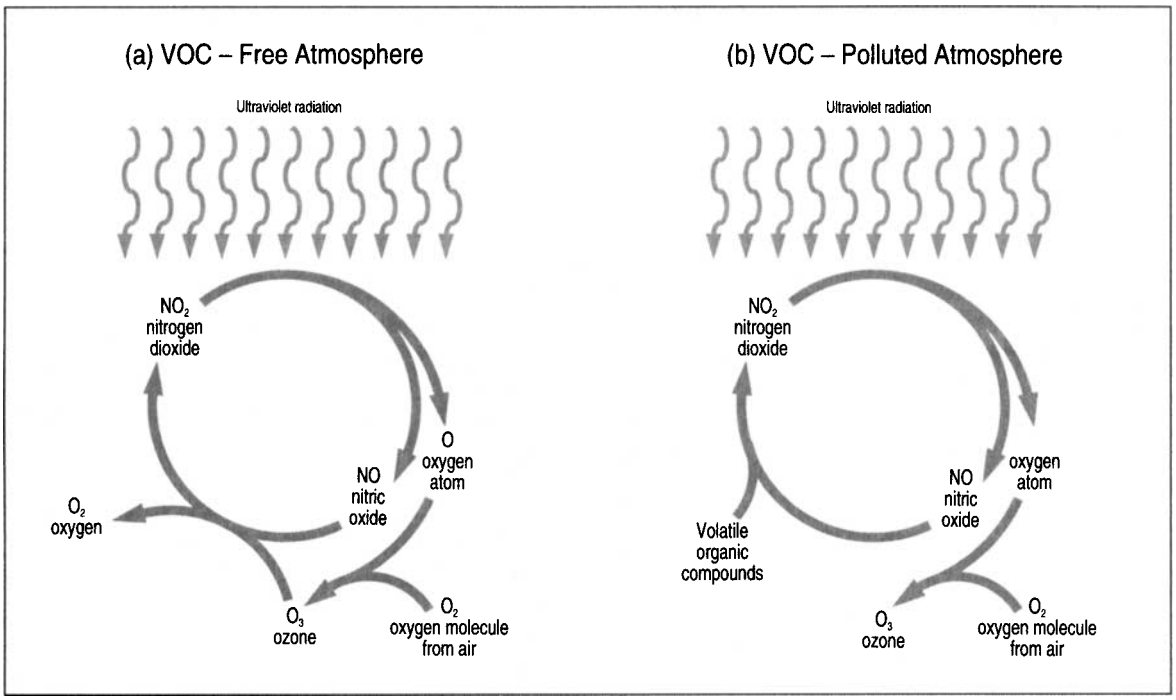


FIG. O-7 In unpolluted air, ground-level ozone forms and breaks down in a steady cycle. Scenario *b* shows one way that pollutants disrupt the natural equilibrium. (Source: Environment Canada.)

and bacterial action in the soil. About 95 percent of human-caused emissions of nitrogen oxides come from the combustion of gasoline, diesel fuel, heavy fuel oil, natural gas, coal, and other fuels, notably in motor vehicles, heavy equipment, turbines, industrial boilers, and power plants (Fig. O-8).

Volatile organic compounds

The term *volatile organic compounds* (VOCs) is used to describe carbon-containing gases and vapors that are present in the air, with the major exceptions of carbon dioxide, carbon monoxide, methane, and chlorofluorocarbons. VOCs are given off by trees and other vegetation, particularly in heavily forested areas. The combustion of fossil fuels, especially in cars and trucks; certain industrial processes; and the evaporation of some liquid fuels and solvents found in cleaning solvents, oil-based paints, varnishes, stains, and thinners are important sources of human-caused VOCs (Fig. O-9). Releases of VOCs lead to ground-level ozone pollution when these emissions occur in the presence of nitrogen oxides (Fig. O-7).

Effects of Ground-Level Ozone

Effects on human health

Ozone is a very irritating and harmful gas. It adversely affected lung function in young, normal subjects who exercised for 6 h in concentrations as low as the present Canadian 1-h objective of 82 parts per billion (ppb). (A part per billion is a unit of measure used to describe the concentration of atmospheric gases. In this case, the unit represents one molecule of ozone in one billion molecules of all gases in the

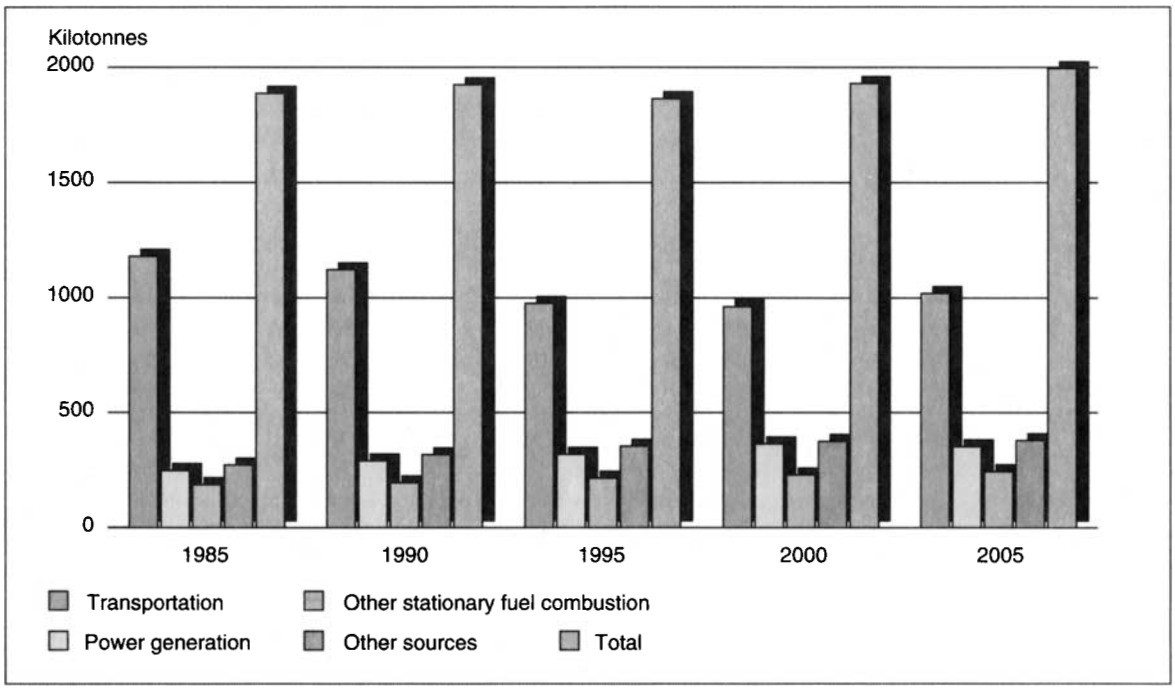


FIG. O-8 Estimates of nitrogen oxide emissions due to human activities in Canada during 1985–2005. (Source: Environment Canada.)

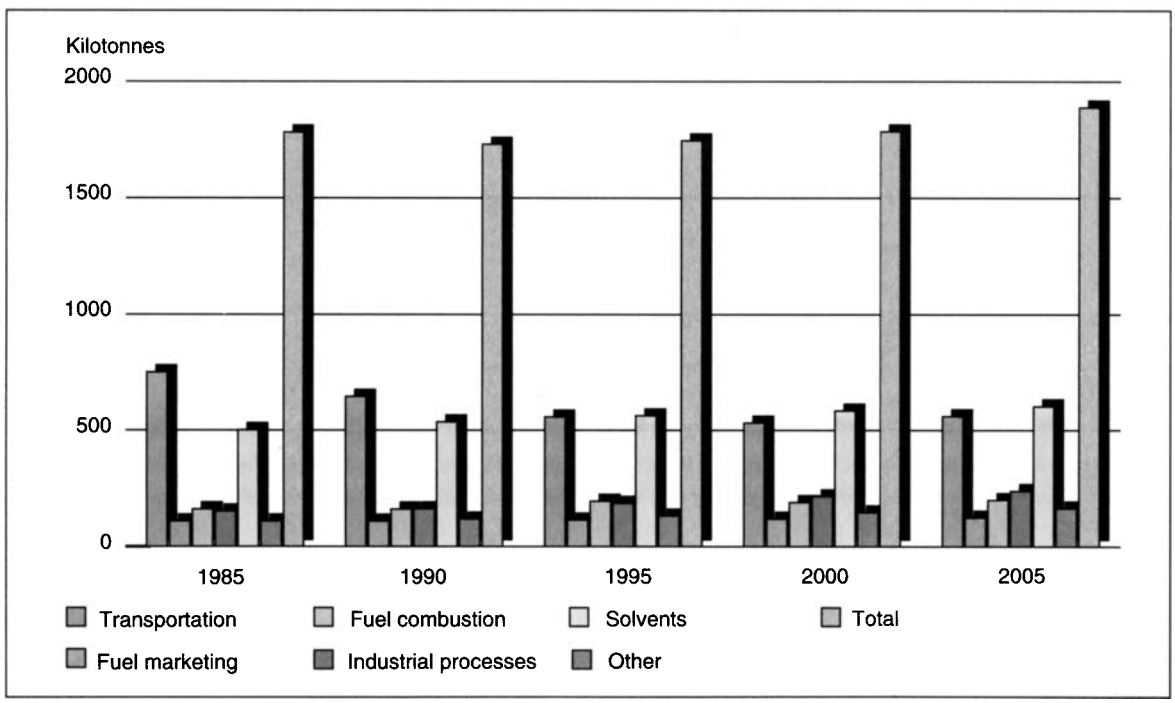


FIG. O-9 Estimates of VOC emissions due to human activities in Canada during 1985–2005. (Source: Environment Canada.)

air.) When lung function is affected, ozone has probably caused inflammation in the lung.

Scientific studies show that after a few days of continuous exposure to ozone, respiratory discomfort disappears. However, although little is known of the long-term effects of repeated ozone exposure on humans, recent research on animals suggests that it may lead to irreversible changes in lung function.

When ozone levels exceed 82 ppb, there is evidence that more people are admitted to hospitals with acute respiratory diseases. In 1987 it was reported that high ozone levels coincided with increased admission of patients with acute respiratory disease to 79 hospitals in southern Ontario. However, it is difficult to separate the effects of ozone from those of sulfate in these epidemiological studies. Furthermore, the health effects of individual pollutants may be intensified when two or more pollutants occur together.

High concentrations of ozone may affect the health of people and vegetation and corrode materials.

Heavy exercise for 2 hours at an exposure of 120 ppb may lead to coughing, shortness of breath, and pain on deep inhalation in healthy adults. Exposures above 120 ppb have resulted in dryness of the throat, shortness of breath, chest tightness and pain, wheezing, fatigue, headache, and nausea.

People working or exercising outdoors inhale larger quantities of air and may suffer more during episodes of ozone pollution. Children are more susceptible because they spend more time outside than adults. Studies showed that children at summer camps in Canada and the United States where they were exposed to a typical summer mix of pollutants, including ozone, experienced a measurable decline in lung function.

Ozone causes similar decreases in lung function in people who have asthma as in those who do not, but the loss is more likely to be serious in those whose lungs are already unhealthy. In clinical studies, people with asthma do not respond to ozone differently than any other population. However, there is recent evidence that when asthmatics are exposed to ozone their sensitivity to allergens is heightened.

Lung function is known to decline with age. Studies of the exposure of human populations to ozone have noted an increase in the rate at which lung function declines. Scientists are researching whether long-term exposure is causing changes in human cells and tissues.

The savings that could be achieved by cutting ground-level ozone pollution are likely considerable.

Effects on vegetation

Ozone is now viewed as the most important pollutant affecting vegetation. Canadian research on the impact that ozone is having on farming has focused mainly on southern Ontario, where ozone levels are typically highest. Estimates of the cost of reduced yields in southern Ontario range from $17 to $70 million, depending on the number of ozone events. Ozone damage to crops also occurs in other regions. Value of lost production in the Fraser Valley has been estimated as $8.8 million annually.

Ozone damages leaf tissue. Leaves may become mottled with yellow, exhibit small black or white spots, develop larger bronze-colored, paper-thin areas, or exhibit other visible symptoms. Inside the leaf, ozone can inhibit metabolic activity, destroy the walls of cells, damage chlorophyll, and reduce photosynthesis. The plant as a whole may grow 10–40 percent more slowly, age prematurely, lose its leaves during the growing season, and produce pollen with a shorter life span.

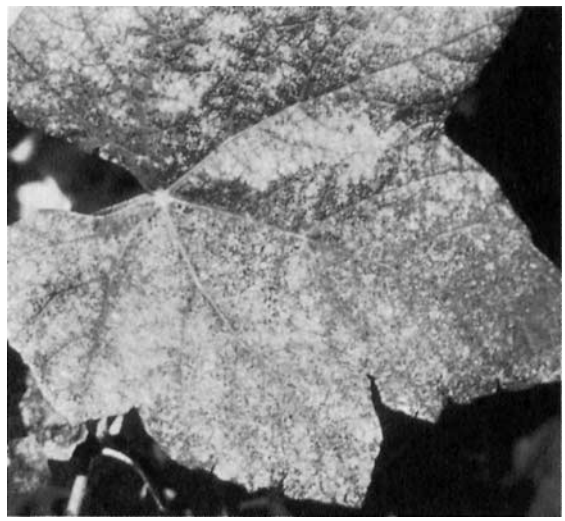


FIG. O-10 Grape leaf with ozone exposure damage. (Source: Environment Canada.)

The effects of ozone on ecosystems are difficult to measure, because species vary in their susceptibility. In forest ecosystems, exposure to ozone may lead to increased vulnerability to disease and other stresses, increased mortality of individuals, and eventually to overall decline of affected species. Both the degree of, and reasons for, the decline in forest health in eastern North America are still debatable, but ozone is believed to be partly responsible for the reported decline of red spruce, sugar maple, and white birch. (See Fig. O-10.)

Damage to materials

Ozone can lead to the development of cracks in products made of rubber or synthetic rubber, such as tires, boots, gloves, and hoses. Continued exposure to high levels of ozone can cause these products to disintegrate completely. Ozone accelerates the fading of dyes; damages cotton, acetate, nylon, polyester, and other textiles; and accelerates the deterioration of some paints and coatings.

It is difficult to pin down the costs of this type of ozone damage. The economic impact in the United States has been estimated at $1 billion, but a similar estimate has yet to be prepared for Canada.

Ambient Air Quality Objective for Ozone

An air quality objective is a statement of the concentration of a specified air pollutant that should not be exceeded beyond a specified length of time, in order to provide adequate protection against adverse effects on humans, animals, plants, and materials. Pollution control agencies routinely monitor the levels of air pollutants and compare the levels with air quality objectives. This allows them to measure their progress in controlling air pollution.

The maximum acceptable level for ground-level ozone in Canada is set at 82 ppb over a 1-h period (see Fig. O-11). An “episode” occurs when the average ozone concentration exceeds 82 ppb for 1 h or more. Ozone episodes in Canada typically last from one to a few days. It is considered that natural levels of ozone in unpolluted conditions would range between 15 and 25 ppb.

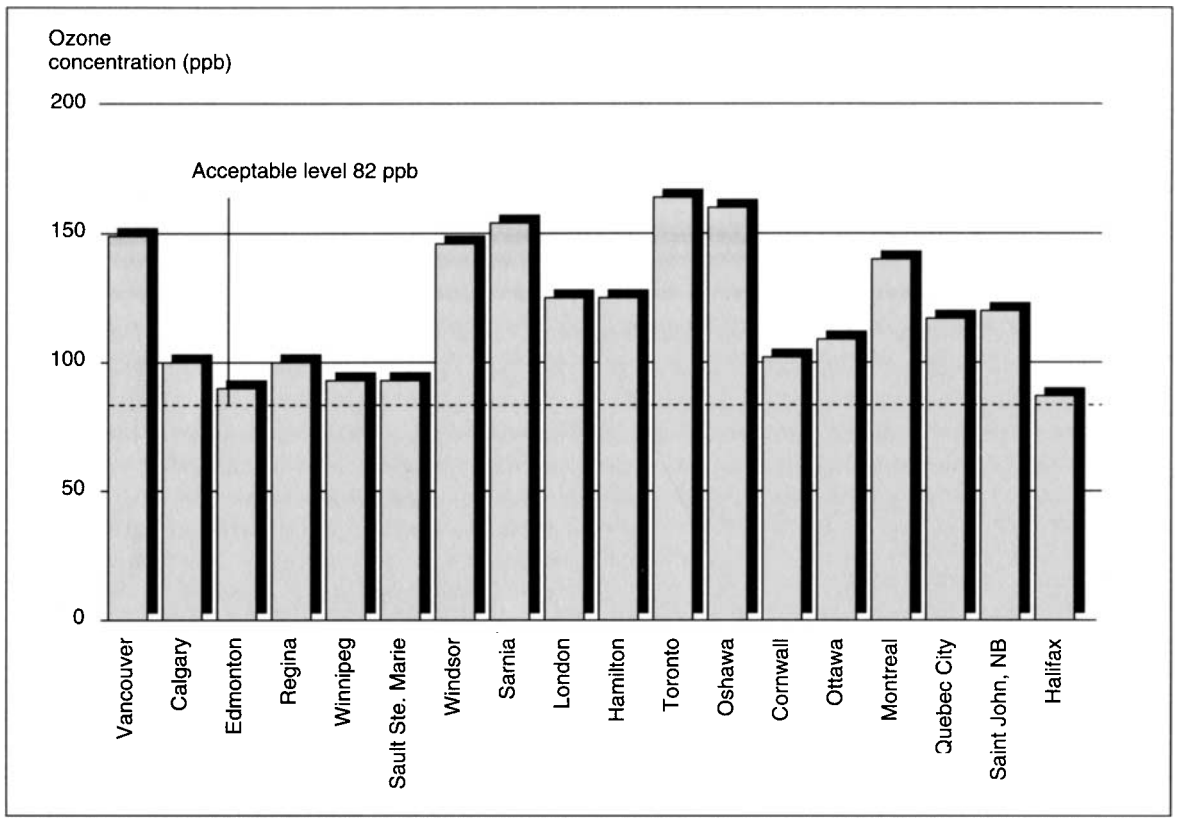


FIG. O-11 Maximum 1-h ozone concentrations for Canadian cities, based on an average of the three highest years during 1983–90. (Source: Environment Canada.)

The federal government has provisionally adopted the number of days per year when ozone concentrations exceed the 1-h air quality objective as its indicator for ground-level ozone (Environment Canada, Indicators Task Force 1991). The objective of the National Environmental Indicators Project is to develop credible, understandable indicators of environmental conditions. These numbers will help decision-makers to make choices consistent with sustainable development and to evaluate the country's progress toward that goal.

Emission Control Options

Measures to control ground-level ozone concentrations focus on the reduction of emissions of nitrogen oxides and VOCs.

Because ground-level ozone is a secondary pollutant, formed by the reaction of primary pollutants, measures to control ground-level ozone concentrations focus on the reduction of emissions of nitrogen oxides and VOCs. The amount of ozone formed depends on the ratio of nitrogen oxides to VOCs in the atmospheric mixture. Under certain conditions, ozone formation could be limited more effectively by controlling nitrogen oxides more than VOCs, and under other conditions the reverse could be true. The complex nature of the problem has made evaluation of control strategies difficult. Computer models are needed to predict the degree of ozone formation based on particular atmospheric conditions. As warm temperatures and

sunlight are needed for ozone formation, it is especially important to reduce summer daytime emissions.

International focus for the control of nitrogen oxides and VOCs

International protocols: In 1988, Canada and 24 other countries signed a protocol to stabilize emissions of nitrogen oxides at 1987 levels by 1994. Canada, the United States, and 19 European countries signed another protocol in November 1991 to reduce the emission of VOCs and their transport across international boundaries. The protocol commits Canada to a 30 percent reduction in annual VOC emissions in the Lower Fraser Valley and Windsor–Quebec Corridor by 1999 based on 1988 levels. Canada is also committed to a national freeze on VOC emissions at 1988 levels by 1999 (United Nations Economic Commission for Europe 1991).

The Canada–U.S. Air Quality Accord: In March 1991, Canada and the United States signed an Air Quality Accord. This agreement addresses the acid rain problem and provides for the study and control of those air pollutants that commonly move across the Canada–U.S. border. Annexes will be developed to specifically address urban smog.

International Joint Commission recommendations on air quality in the Detroit–Windsor–Port Huron–Sarnia Region: In March 1992, the International Joint Commission (IJC) highlighted the need for governments to phase out emissions of air toxics in the region. Among 19 recommendations, the IJC promoted development of a joint regional ozone control strategy that includes emission controls for mobile and stationary sources, including coke ovens. A common ground-level ozone standard has also been recommended for the region.

Canada's management plan for nitrogen oxides and VOCs

A national plan has been developed for the management of nitrogen oxides and VOCs.

A national plan for the management of nitrogen oxides and VOCs has been developed by federal and provincial governments through the Canadian Council of Ministers of the Environment. Initiated in 1988 as a coordinated approach to reducing levels of ground-level ozone throughout the country, the plan was developed in consultation with industry, public interest groups, and environmental groups. It aims for consistent attainment of the 1-h ground-level ozone air quality objective of 82 ppb by the year 2005. Implementation is occurring in several phases:

Phase I (in place by 1995):

a. National Prevention Program: The program outlines 31 initiatives that will reduce emissions of nitrogen oxides and VOCs, including the following:
   - *Energy-conservation and product-improvement measures*:
     - Energy efficiency standards in equipment and appliances
     - Energy audits by industry
     - Reductions in emissions when surface coatings are applied and when adhesives, sealants, and general solvents are used
   - *Public education to promote informed consumer choice and an environmentally responsible lifestyle including*:
     - Energy-conserving driving habits and alternative transportation modes, such as cycling, walking, and ridesharing
     - Energy conservation
     - The use of energy-efficient products
     - The construction of energy-efficient homes and businesses
     - Improved solvent use and recycling

- *Source control initiatives:*
  - New emission standards for cars and light trucks
  - Caps on emissions of nitrogen oxides from trains
  - Emission guidelines for new sources, i.e., power plants, industrial boilers, and compressor engines, as well as for storage tanks for volatile liquids, chemical processes used by industry, commercial and industrial coating applications, printing, degreasing, and dry cleaning.

b. Remedial programs: The plan identifies 27 sample regional initiatives for reducing ozone, which could be implemented in the three ozone problem areas: the Lower Fraser Valley, the Windsor–Quebec Corridor, the Southern Atlantic Region. Most initiatives involve the installation, retrofit, or enhancement of emission-control technologies for existing sources.

c. Study initiatives: The plan outlines 24 research initiatives aimed at determining the most effective control strategies for limiting the formation of ground-level ozone. Ambient air monitoring, modeling, and studies of industrial processes and emission sources will help to determine what controls on emissions of nitrogen oxides and VOCs will be necessary in Phases II and III of the plan.

d. Federal–provincial agreements: Federal–provincial agreements will establish the responsibilities of the respective governments for specific remedial actions required to reduce ground-level ozone concentrations. The agreements will also set out interim targets for emission reductions.

Phases II and III: Phase II of the management plan will establish emission caps for problem areas for the years 2000 and 2005. To meet these caps, it is likely that additional steps, over and above the initiatives laid out in Phase I, will be needed. Phase III will make final adjustments to emission caps and control programs.

Implementation of Phase I of the NO_x/VOC management plan should be a significant step toward solving the country's ground-level ozone problem by 2005. Maximum ground-level ozone concentrations should be reduced by 15–35 percent as a result of predicted Canadian and U.S. emission reductions. In addition, joint Canada–U.S. emission reductions will lead to a 40–60 percent reduction in the time during which the maximum acceptable ground-level ozone objective (82 ppb) is exceeded in the regions of greatest concern.

Some regional remedial measures already underway

- *The Montreal Urban Community* has passed regulations that require dry cleaning and printing facilities, surface-coating applications, and metal degreasing operations to control emissions. Substantial reductions have been achieved.
- *The B.C. Motor Vehicle Branch* is implementing mandatory vehicle emission testing starting in late 1992 under the *Air Care Program*. As a condition of annual license renewal, all light-duty vehicles in the province's Lower Mainland will be inspected for exhaust emissions and emission-control systems. Those not meeting the standards will undergo repairs.

Global efforts to address stratospheric ozone depletion have been underway since 1981. The Vienna Convention for the Protection of the Ozone Layer entered into force on September 22, 1988. As of March 1, 1989, thirty-seven countries, representing the vast majority of the industrialized nations of the world, had ratified this Convention. The Convention provides the framework for cooperative activities, including the exchange of data or information related to the ozone layer. This Convention provides for the subsequent creation of protocols (free-standing

treaties) for matters such as control of specific pollutants or families of pollutants. The first such protocol created was the Montreal Protocol on Substances that Deplete the Ozone Layer.

The Montreal Protocol was signed in Montreal, Canada on September 16, 1987. It is clearly a watershed in cooperative and collaborative international undertakings. It introduces many new features never before established in international law.

The Montreal Protocol had two requirements for entry into force; namely, 11 signatures of ratification by countries and these countries must represent at least two-thirds of global consumption of the controlled substances. The Montreal Protocol entered into force on January 1, 1989, and as of March 1989 had already attained 33 ratifications, again representing most of the industrialized states of the world.

On May 1988, Environment Canada published its first Control Options Report, titled: "Preserving the Ozone Layer: a first step." This report set out a series of options to implement regulations to meet Canada's obligations under the Montreal Protocol.

The Montreal Protocol calls for a 50 percent cutback in the 1986 levels of consumption of five chlorofluorocarbons (CFC 11, 12, 113, 114, and 115) and a freeze at 1986 consumption levels of three brominated fluorocarbons called Halons (Halon 1211, 1301, and 2402). At a series of United Nations Environment Programme (UNEP) meetings held in The Hague, Netherlands, in October 1988, the world's leading scientists expressed the consensus viewpoint that the Antarctic hole will remain unless the emissions of controlled CFCs are reduced by at least 85 percent from 1986 levels. The target reductions contained in the Montreal Protocol are currently undergoing international review. This review is expected to culminate in amendments that will tighten the Montreal Protocol. Canada contributed to both the organization and conduct the UNEP meeting in The Hague and fully supports the notion of reducing consumption further.

On February 20, 1989, the Federal Government of Canada announced that it had set as its objective the complete elimination of controlled CFCs within the next 10 years. It also called on the rest of the world to set as its common target a reduction of no less than 85 percent by no later than 1999. The Minister of the Environment further announced that the following actions would be taken to achieve the Canadian objective.

1. Implement the protocol

As a first step, regulations will be enacted under the Canadian Environmental Protection Act to implement the current control requirements set out in the protocol. These are a freeze in consumption at 1986 levels (CFCs on July 1, 1989, and Halons on January 1, 1992) and a two-step reduction in annual consumption of CFCs to 50 percent of 1986 levels by 1999.

2. Regulate a reduction in chlorofluorocarbon consumption of at least 85 percent

Draft regulations recently released call for at least an 85 percent reduction of the controlled CFCs by no later than 1999. Consultation on what is achievable is expected to increase the percentage reduction and tighten the time frame.

3. Prohibit specific CFC uses

As a third step, draft regulations have been released for discussion purposes that would prohibit the use of ozone-depleting substances for nonessential uses or where substitutes are available. For example, the import, manufacture, and sale of aerosol

products containing controlled CFCs (with the exception of certain medical and industrial applications for which alternatives are not yet available or for which fire safety is a particular concern); food packaging foam including food and beverage containers containing or manufactured with controlled CFCs; portable handheld fire extinguishers for home use containing Halons; and small pressurized canisters that contain CFCs, including refrigerants, air horns, and party streamers, will be prohibited by January 1, 1990. As safe alternatives become available, a similar prohibition will apply to the following aerosol products: release agents for molds used in the production of plastic and elastomeric materials, cleaners and solvents for commercial use on electrical or electronic equipment, and products used in mining applications where fire hazard is critical.

4. Further control measures

As a fourth step, this comprehensive control options report has been prepared to focus discussion on the earliest possible prohibition dates for remaining CFC uses. Some examples of prohibition dates proposed in this report are as follows:

1. Rigid foams
   a. Insulating foams
      - Polyurethane (1992–1994)
      - Polystyrene (1990–1991)
      - Phenolic (1991–1992)
   b. Packaging
      - Food (1990)
      - Other (1990–1991)
2. Flexible foams
   a. Car seats, furniture cushions, etc. (1990–1992)
3. Refrigeration
   a. New refrigeration and air-conditioning equipment (1994–1999)
   b. Existing equipment maintenance (as replacements are available)
4. Solvents
   a. Electronic (1991–1994)
   b. Metal cleaning (1990–1991)
   c. Dry cleaning (1991–1992)
5. Others
   a. Hospital sterilants, optical coatings (1990–1994)

As these remaining uses are prohibited, the total allowable quantities of controlled CFCs produced and consumed in Canada will be lowered.

This Control Options Report, “Preserving the Ozone Layer: A Step Beyond the Montreal Protocol,” describes the most promising options to control ozone-depleting substances for each process group and product group including rigid and flexible foams, refrigeration and air conditioning, solvents, sterilants, aerosols, and fire suppression systems in which controlled CFCs and Halons are used. The control options comprise three main categories:

- Emission controls
- Chemical and process substitutes
- Product substitutes

Depletion of Ozone in the Stratosphere

As excess ozone at lower atmospheric levels is a problem, reduced ozone in the stratosphere is also dangerous to life. Industry is therefore being pressured to take steps to alleviate this problem. Some background information follows.

Figure O-12* describes effects of ozone on vegetation and health.

Benefits and costs†

The benefits of the Montreal Protocol consist of two components. For benefits associated with reduced damage to materials, reduced damage to agricultural productivity, and reduced damage to fish stocks, estimates of the dollar amounts of benefits are shown. For health impacts, including reduced incidence of skin cancers, reduced fatal skin cancers, and reduced cataract incidence, estimates of averted health effects are shown. In both cases, the framework for producing the estimates is the same. Effects are estimated for the scenario in which no controls on the consumption of ozone-depleting substances are implemented relative to the alternative scenario represented by the introduction of the Montreal Protocol.

The overall benefits of the Montreal Protocol are shown in summary form in Table O-1. This table shows quantified dollar benefits of $459 billion plus a reduction in skin cancer cases of 3.4 million, 129 million fewer cataract cases, and more than 330,000 reduced fatalities.

Montreal Protocol costs are also substantial but small relative to benefits. The present discounted value of protocol costs over the time period from 1987 to 2060 total $235 billion. Although substantial, these costs are less than the quantifiable benefits from reduced damages to agriculture, fishing, and materials. The benefits from averted health effects are extremely large in relation to the costs. If reduced fatalities from skin cancers alone are considered, the number of cases averted from 1987 to 2060 is 333,500.

Costs of Replacing Ozone-Depleting Substances‡

Overview and methodology

The methodology for assessing the costs of implementing the Montreal Protocol focuses on the economic costs of the controls that have been introduced. In the economics literature, this is frequently referred to as assessing the real resource costs of the policy. See Table O-2.

As an example, consider refrigeration and air-conditioning services. Assume in this example that a constant quantity of these services will be produced—that is, there are no price impacts on this quantity. Prior to the regulation, CFCs are used as refrigerants throughout this sector. As a result of the introduction of the Montreal Protocol, a number of changes are made in this sector to reduce and then eliminate CFCs. The economic cost of the regulation is the difference in the cost of producing these outputs prior to the control and after it is introduced.

In this cost methodology, the real resource costs that are relevant consist of the additional quantity of resources needed to produce this constant level of output.

---

* Source: Environment Canada, "Preserving the Ozone Layer: A Step Beyond," April 1989. This report was written with specific reference to Canada, but provides applications information for readers anywhere.

† Source: Adapted from extracts from Environment Canada, "Montreal Protocol 1987 to 1997: Global Benefits and Costs of the Montreal Protocol on Substances That Deplete the Ozone Layer."

‡ Source: Environment Canada SOE Sheet 92-1, written with specific reference to the U.S.–Canada Air Quality Agreement and Ozone Levels in Canada.

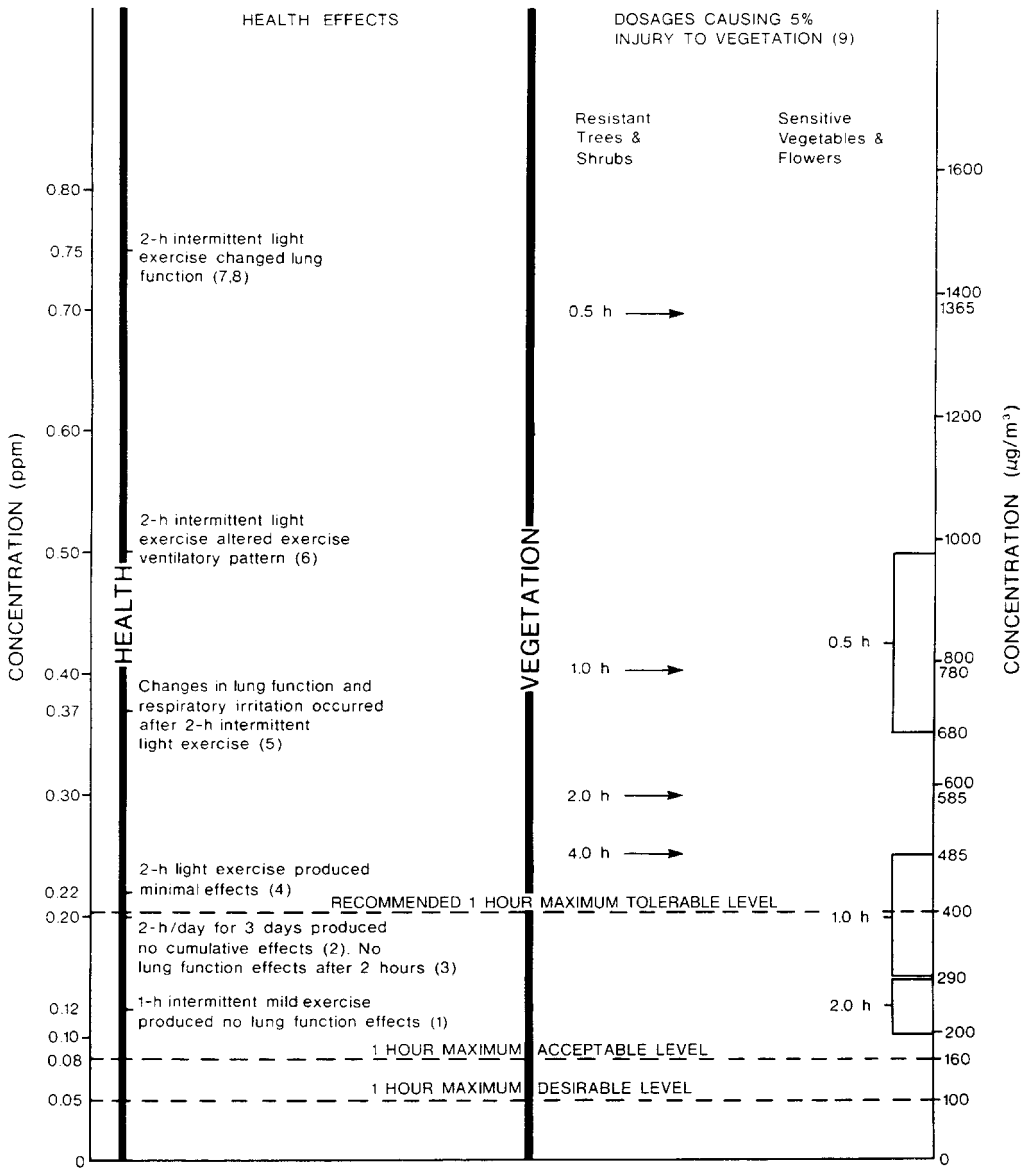


FIG. O-12 Effects of ozone on vegetation and health. (Source: Environment Canada.)

Other private sector costs may be incurred that are not reflected in the economic costs described above. If the price of CFCs increases, for example, due to regulatory restrictions on quantity, some private sector users will register this as an increased cost. This is not a economic cost, however, because no additional input resources are required to produce the output in question. Viewed somewhat differently, the price increase is not an economic cost because the additional costs of the purchasers are exactly offset by the additional revenues of the producers.

In any regulatory scenario, we would expect to observe both real resource cost impacts and price impacts as described above. Economists refer to the price effects as transfers in that they involve transfers from one group in society to another. In

TABLE O-1 Benefits and Costs of the Montreal Protocol, 1987–2060

BENEFITS	
Health benefits	
Reduced cases of nonmelanoma skin cancer	19,100,000
Reduced cases of melanoma skin cancer	1,500,000
Reduced cases of cataracts*	129,100,000
Reduced skin cancer fatalities*	333,500
Dollar benefits	
Reduced fisheries damage	$238 billion
Reduced agricultural damage	$191 billion
Reduced damage to materials	$30 billion
Total dollar benefits	$459 billion
COSTS	
Overall costs	$235 billion
NET BENEFITS	
Benefits minus costs	$224 billion plus health benefits

* Mean of upper- and lower-bound estimates. Base case discount rate is 5 percent.
Note that overall benefits are the sum of the health effects and the benefits that are expressed in dollar terms.
SOURCE: ARC estimates.

TABLE O-2 Global Consumption of Ozone-Depleting Substances (1986)

ODS		Mean ODP	Developed Countries	Article 5(1) Countries	Total Kilotonnes	Total ODP Units
CFCs		0.95	1,010.0	111.0	1,121.0	1,065.0
HCFCs		0.05	169.9	13.4	183.3	9.2
Halons	1,211	3.00	11.0	2.7	13.7	41.1
	1,301	10.00	9.9	1.2	11.1	111.0
	2,402	6.00	10.2	n.a.	10.2	61.2
Methyl chloroform		0.11	581.6	27.4	609.0	67.0
Carbon tetrachloride (dispersive, nonfeedstock uses)		1.10	92.2	4.4	96.6	106.3
Methyl bromide		0.06	42.6	10.8	53.4	32.0
Total units (kt)			1,927.4	170.9	2,098.3	1,492.8

SOURCE: Estimates by ARC, based on data from U.S. Environmental Protection Agency and UNEP. Note that small quantities of halon 2,402 were used in some Article 5(1) countries.

this case, the transfer is from users of CFCs to producers. Economic costs exclude transfers and in this way focus on the additional real resource costs associated with the regulation.

In this way of assessing costs, there are no incremental costs associated with firms that continue to use CFCs, with no change in their production process. They may pay higher CFC prices but this is not a real resource cost to society. The clearest example of costs is the case in which the firm switches to a new HCFC or HFC refrigerant. The refrigerant itself may be more expensive and it may also involve system or process changes. If there are capital costs, these must be amortized over the life of the equipment and in this way, these costs are divided among a range of current and future end users. This means that firms that convert away from CFCs will have incremental costs that must be measured. Other firms may try to reduce

the quantity of CFCs used per unit of output. Others may try to recycle or otherwise capture CFCs that are used. All of these adjustments involve economic costs.

Costs of CFC replacement

Aerosols. In this section we estimate the cost of controls applied on the use of CFCs in aerosols as a result of the Montreal Protocol for both developed countries and the Article 5(1) countries.

Prior to the protocol, significant reductions in CFC use in aerosols had taken place in Canada, the U.S., and a number of European countries following a ban on certain uses of CFCs in aerosols that became effective in 1979. Following the application of controls under the Montreal Protocol, consumption has declined steadily in most developed countries and is now accounted for only by essential uses, primarily in the medical field.

The consumption of ozone-depleting substances in aerosols for the period 1986 to 1995 is based on estimates provided in the 1994 Report of the Aerosols, Sterilants, Miscellaneous Uses and Carbon Tetrachloride Technical Options Committee. The forecast use without controls is based on the observed and expected growth of aerosol products in general. This market is essentially a mature market that would grow more slowly than GDP. So an overall growth rate of 1 percent per year was applied for the No Control scenario for the developed countries. In the Article 5(1) countries, consumption without controls is assumed to grow at 10 percent per year until 2005 and then at 5 percent. In the Montreal Protocol controls scenario, some minimal amount is still used to support certain vital applications, mostly in the medical field, for which substitutes are only slowly appearing on the market. That amount is assumed to decline to zero by the year 2010, when CFC use for these applications is mandated to cease.

From a material cost standpoint, the use of hydrocarbons as a substitute for CFCs in developed countries has cut the material cost of aerosol propellant material significantly, in some cases by as much as 80 percent. Overall, we have used an estimate of two thirds saving across all affected aerosol products. That saving is reflected as a negative cost differential for each year in our analysis, which extends to 2060. We assume that pumps and other ODS substitutes generate comparable cost savings.

The main sources of additional costs as a result of replacing CFCs by hydrocarbons in aerosols were:

- Upgrading of sprinkler systems and ventilation in production and storage facilities, and extra storage space for segregated storage of flammable materials, in aerosol manufacturing plants
- R&D for new formulations
- Changes in labels
- New permits

These were all one-time costs. Except for changes in material costs, it does not appear that operating costs were affected. Industry sources indicated that, once the appropriate fire protection steps were taken, insurance premiums did not rise appreciably. Aerosol cans themselves did not have to be changed.

Based on this information, we estimate that the total one-time cost in developed countries for R&D and the other items listed above will have been more than offset by operating cost savings from lower propellant costs. We have estimated an actual saving of $0.88 per kilogram of CFC displaced. That is, the present value of the

reduced cost of propellant exceeds the present value of the costs of converting to the use of hydrocarbons and other alternatives for developed countries.

The relative ease and low cost of the transition away from CFCs to hydrocarbons, particularly in the developed countries is closely related to the ban on CFCs as aerosol propellants in many countries in the late 1970s and early 1980s. This ban reduced the quantities of CFCs that had to be replaced as a result of the Montreal Protocol. More importantly, it also provided a firm base of technical knowledge on the use of hydrocarbon alternatives to CFC propellants.

The area within the aerosol sector in which the conversion process has been most difficult is in the phase-out of CFCs in metered dose inhalers (MDIs). Although the global quantities of CFC use are relatively small, the application is a critical one and has consumed substantial resources in the development of alternatives. In this study, we assume that development and testing costs for MDIs will total $2.5 billion in 1997 U.S. dollars. This offsets some of the cost reductions for other aerosol application areas in developed countries.

For Article 5(1) countries, the picture is somewhat different. In many of these countries, access to hydrocarbons may be a problem and the small scale of enterprises will make the up-front conversion costs more of a burden. The available evidence indicates substantial diversity in control costs in Article 5(1) countries. We estimate that 35 percent of consumption reduction in these countries will generate the same cost savings as described for developed countries. Another 35 percent will have no incremental costs while 30 percent will involve the average cost of $2.68 per kilogram described in the TEAP *Replenishment Report* (1996). Overall, this produces a cost per kilogram of $0.50.

Overall, our assessment is that for aerosols, the Montreal Protocol was responsible for reduced consumption of 29.9 million tonnes of CFCs in the developed countries and a reduction of 12.5 million tonnes in the Article 5(1) countries for the time period from 1987 to 2060. The controls and no-controls scenarios for CFCs in aerosols are shown graphically in Fig. O-13. Due to lower propellant costs in developed countries, the costs of meeting the protocol requirements translate into an overall cost saving of $6.5 billion. This figure nets out the R&D costs for MDIs. In the Article 5(1) countries, the adjustment cost is estimated to be $1.2 billion.

Flexible and rigid foams. In 1986, consumption of CFCs by the foam plastics manufacturing industry was extremely varied; however, the dominant volume uses were in polyurethane and polystyrene foam applications.

Polyurethane foam is produced as the result of a chemical reaction of a polyol, isocyanate, and blowing agents. In some foam plastics, the resulting cells are closed, trapping the blowing agent inside, while in others the cells are produced open and the blowing agent escapes. Used in a large variety of products, polyurethane foams can be classified into three major categories: rigid (closed cell), flexible (open cell), and integral skin. Product applications include insulating materials for buildings and appliances, cushioning products for furnishings and automobiles, packaging for protection of high-value products, and automobile bumpers and instrument panels.

Extruded polystyrene foam is produced by melting polystyrene resin in an extruder and injecting a blowing agent into the resin under high pressure. Dispersed into the molten polystyrene, the blowing agent volatizes as a mixture leaves the extruder, causing the plastic to foam. There are two distinct forms of extruded polystyrene: foam sheet, which is typically used in disposable food service packaging and packing products, and foam boardstock, which is typically used as an insulation product.

The physical and chemical properties of the blowing agent affect the processability

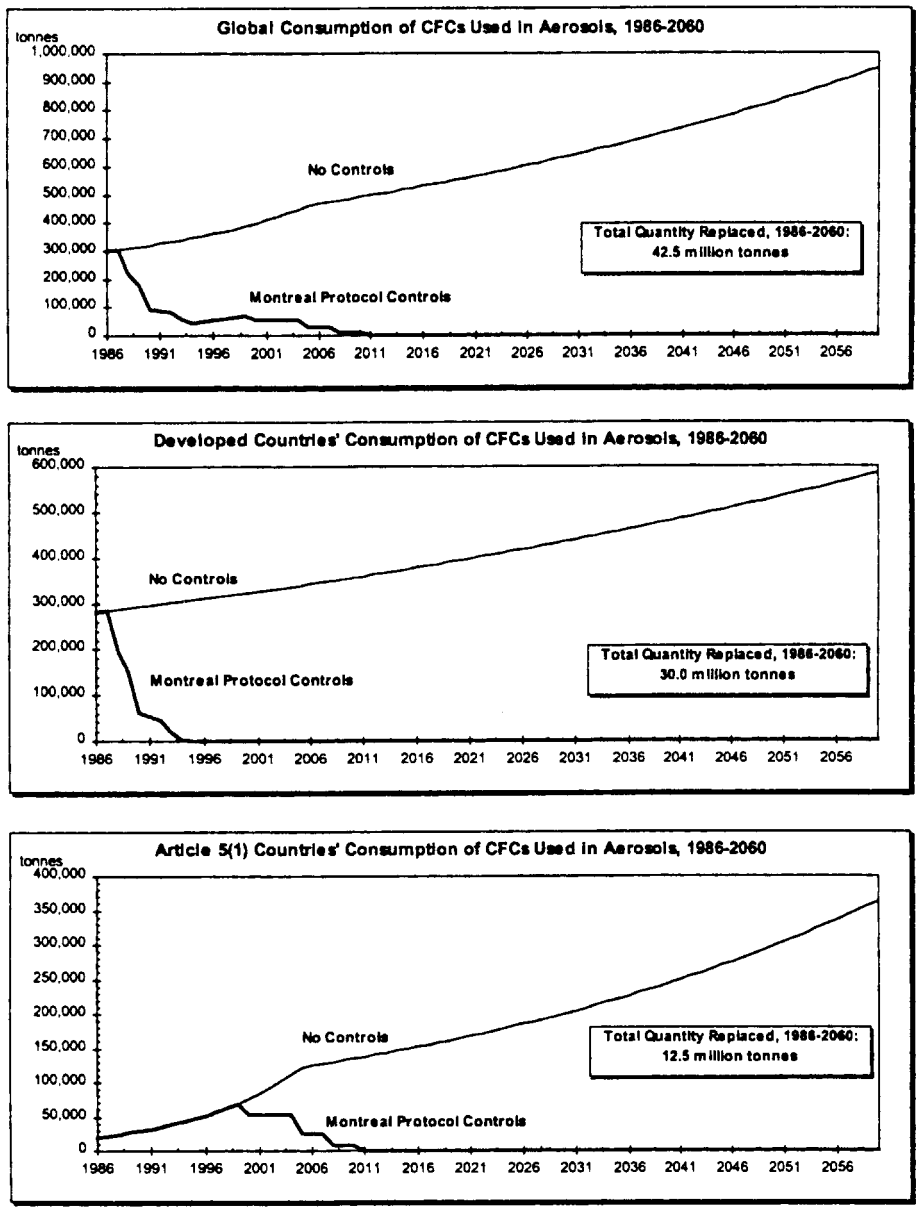


FIG. O-13 Consumption of CFCs in aerosols under Montreal Protocol and no-controls scenarios. (Source: Environment Canada.)

of the foam in manufacture and the characteristics of the foam product such as density and durability. For thermal insulating applications the blowing agent contributes to the overall insulating capacity of the product.

For flexible and rigid foams, global consumption of CFCs as blowing agents prior to the Montreal Protocol is estimated as follows (UNEP Foam TOC 1994):

- Rigid polyurethane: 149,000
- Flexible polyurethane: 60,400
- Phenolic: 1,400

- Extruded polystyrene: 37,600
- Polyolefin: 19,000
- Total: 267,400 (tonnes in 1986)

CFC-11 was the auxiliary blowing agent used in the production of soft and low-density polyurethane slabstock foam, sometimes augmented by methylene chloride. Prior to the Montreal Protocol and as late as 1990, this auxiliary blowing agent was used for approximately 25 percent of the flexible foam slabstock produced globally. The reduction of the use of CFC-11 in flexible foam was achieved through conversion to complete water-blowing formulations for molded products, replacement by methylene chloride in slabstock and a number of other measures including the use of new polyols and related systems and by the use of liquid carbon dioxide in slabstock foams.

For the other foam categories, the largest quantities are accounted for by rigid polyurethane foam, which used CFC-11 as a blowing agent, and extruded polystyrene, which used CFC-12 as a blowing agent. In developed countries, the transition out of CFCs was generally completed by 1996. Zero ODP alternatives are the substitutes of choice for certain rigid thermal insulation foams. However, in several markets and for certain applications HCFCs are necessary for rigid thermal insulating foams until zero ODP solutions are proven including high energy efficiency or properties required for safety.

The interim replacements for CFC-11 and CFC-12 as blowing agents have been primarily HCFC-141b and HCFC-142b, respectively. Including development and testing costs plus continuing operating cost differentials, costs are estimated to be $1.48 per kilogram for replacing CFC-12 in polystyrene and $2.01 per kilogram for replacing CFC-11 in polyurethane. In other foamed plastic applications, costs are estimated to be $2.00 per kilogram. For flexible foams, cost increases in some alternatives have been offset by decreases in others. As a result, no incremental costs are estimated for this sector.

For the entire foam sector, this report computes a weighted average cost per kilogram reflecting the costs per kilogram described above and the quantities used in each application. Overall, this provides an estimate of $1.48 per kilogram displaced for quantities used in developed countries. For Article 5(1) countries, we have used the cost per kilogram estimate of $8.31 from the TEAP *Replenishment Report* (1996) for 50 percent of foam production and the developed country estimate of $1.48 per kilogram for the remaining 50 percent. Overall, this produces an estimate of $4.90 per kilogram.

Global quantities per year are estimated on the basis of data shown in the 1994/1995 TEAP Foam Technical Options Report. The division of these quantities into developed country consumption and Article 5(1) consumption is on the basis of estimates by ARC (Applied Research Consultants) using baseline consumption data from the Multilateral Fund.

Global costs of replacing CFCs in the foam sector are estimated to be $17.9 billion for the conversion away from CFCs. Costs for converting away from HCFCs are estimated elsewhere in this report. The global cost estimate breaks down into costs of $14.8 billion in developed countries and $3.1 billion in Article 5(1) countries. All costs are present discounted values calculated from 1989 to 2060 and expressed in 1997 dollars. The controls and no controls quantity scenarios for the use of CFCs in foams are shown in Fig. O-14.

Refrigeration and air conditioning. This section provides estimates of the costs of implementing the Montreal Protocol in the Refrigeration and Air Conditioning

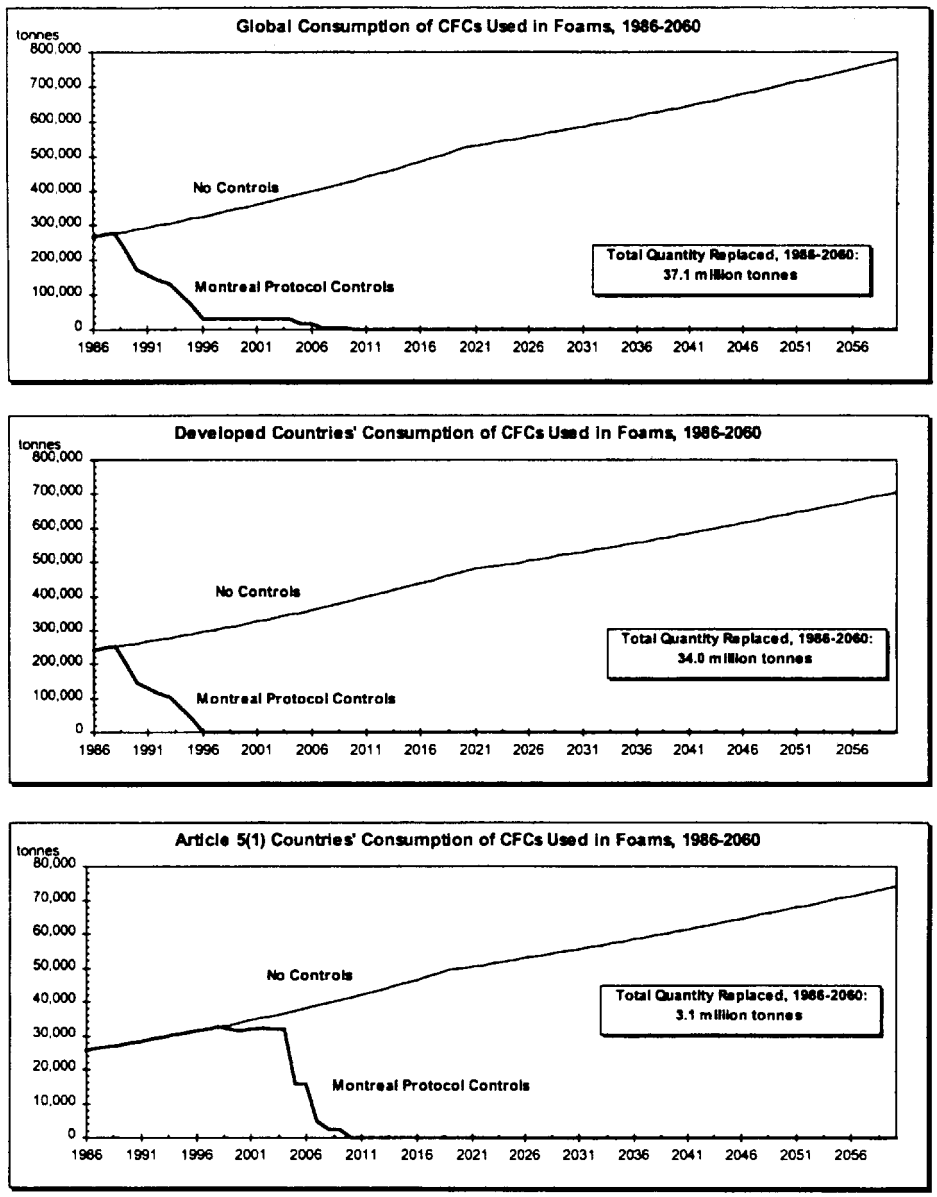


FIG. O-14 Consumption of CFCs in foams under Montreal Protocol and no-controls scenarios. (Source: Environment Canada.)

sector. The Refrigeration and Air Conditioning sector involves a wide variety of applications, the vast majority of which use refrigerants that have been targeted by the Montreal Protocol either in the short term (CFCs) or in the medium term (HCFCs). We estimate the costs of replacing HCFC refrigerants in a separate section of the report.

The 1994 Report of the Refrigeration Air Conditioning and Heat Pumps Technical Options Committee of UNEP provides a detailed review of the various subsectors that make up the Refrigeration and Air Conditioning sector, the quantities of various refrigerants used in these subsectors, as well as the options available for replacing CFCs and HCFCs, the two major refrigerant groups targeted by the Montreal Protocol.

For a number of refrigeration/air-conditioning applications the implementation

of the protocol will be a two-stage process. The first stage consists of replacing a CFC by an HCFC and the second stage involves replacing an HCFC by a zero-ODP refrigerant, usually an HFC. This section considers only the costs of the first stage. HCFC costs are estimated separately.

The dollar cost per kilogram of displaced ozone-depleting refrigerant is a constant cost which, when multiplied by the kilograms saved in each year, yields a stream of payments whose net present value is equal to the net present value of the total incremental costs of implementing the protocol for this sector. This cost is obtained by dividing the net present value of all incremental costs by the net present value of displaced ozone-depleting refrigerant.

For the developed countries, the cost estimation framework for converting away from CFCs in Refrigeration and Air Conditioning provides an estimate of $4.62 per kilogram for the developed countries.

For the Article 5(1) countries, the TEAP *Replenishment Report* (1996) provides an estimate of the upper bound incremental cost per kilogram of replacing CFCs in air-conditioning and refrigeration applications. Many lower cost reduction approaches such as improved service procedures are also available in Article 5(1) countries. Our overall cost estimate is based on a weighted average cost per kilogram. In this weighted average, 60 percent of refrigerant consumption is replaced at the same cost as in developed countries, 10 percent is at a cost premium of 1.5 relative to developed countries and the remaining 40 percent is at the upper bound incremental cost of $15.61 per kilogram from TEAP (1996). Using these data, the cost estimate in this report is that replacing CFCs in this sector in Article 5(1) countries costs $8.15 per kilogram.

Total costs for this sector are based on the cost per kilogram estimates described above and on estimates of the gap between consumption of CFC refrigerants with and without the Montreal Protocol controls. Figure O-15 shows the consumption patterns in the controls and no-controls scenarios. Based on these data, estimated costs for replacing CFC refrigerants are $95.4 billion for the global total. This is broken down into costs of $47.7 billion in the developed countries and $47.7 billion in the Article 5(1) countries. All costs are measured as a present value over the time period 1989 to 2060 and are expressed in constant 1997 dollars using a discount rate of 5 percent.

In the Article 5(1) countries, we use the cost per kilogram estimate for solvents that is provided in the TEAP Replenishment Report (1996) as a measure of upper bound incremental costs. The data shown in that report have been weighted by country type to produce an incremental upper bound cost estimate of $15.84 per kilogram in the Article 5(1) countries. Improved conservation practices suggest and the technology transfer practices of multinational firms suggest that half of the adjustments should occur at the same cost as in developed countries. A further 40 percent of adjustments are costed at 1.5 times the estimated cost in developed countries and 10 percent of adjustments are costed at the upper bound incremental cost. These calculations generate an overall cost per kilogram in Article 5(1) countries of $2.82.

Total costs are based on the costs per kilogram and on estimates of the difference between consumption with and without controls for CFC-113. The two consumption scenarios are shown graphically in Fig. O-16. Using these data, this report estimates global costs for this sector of $18.7 billion. This is the present discounted value of costs at a discount rate of 5 percent measured over the time period 1989 to 2060 and expressed in 1997 dollars. These costs include $16.5 billion for the developed countries and $2.2 billion for the Article 5(1) countries.

Sterilants. Prior to the development of the Montreal Protocol controls, CFC-12 was used extensively as a diluent with ethylene oxide (EO) to produce the sterilant gas

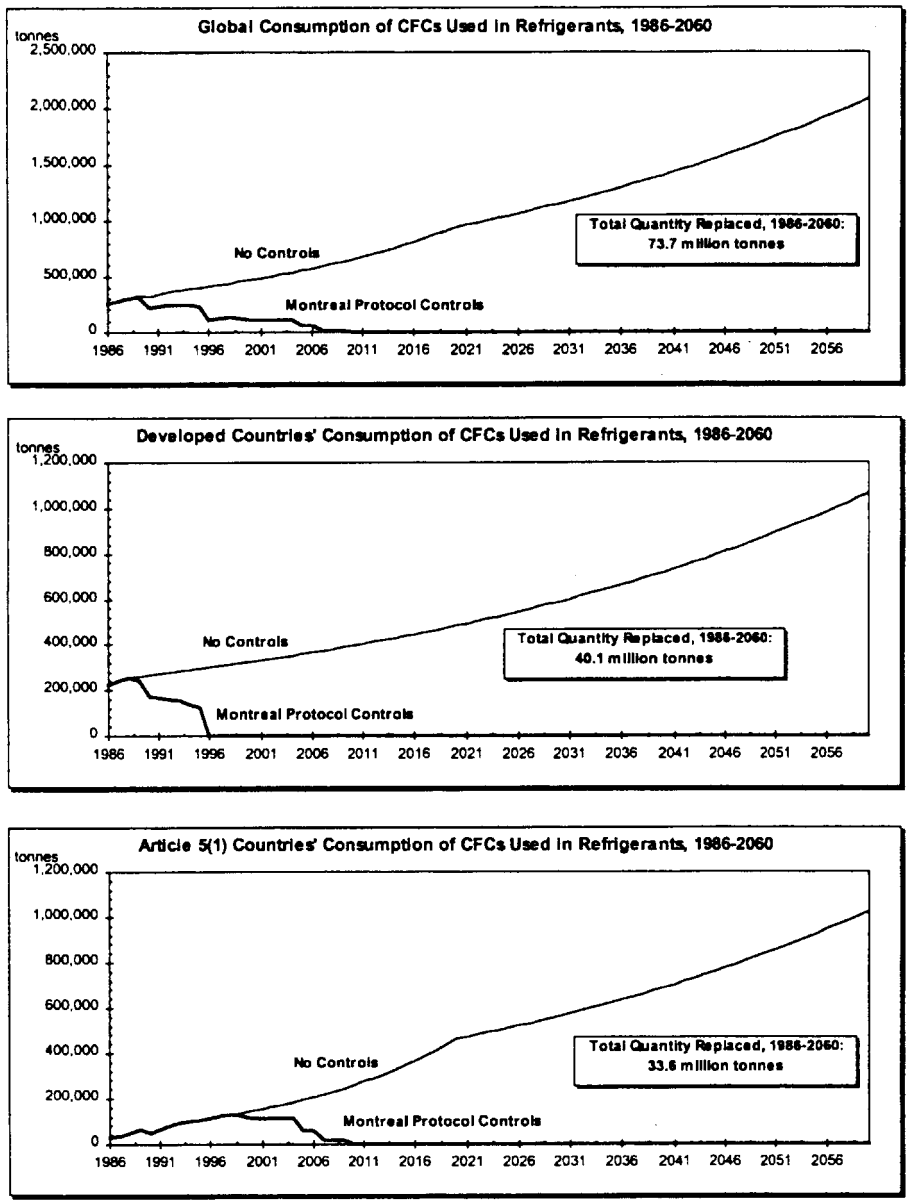


FIG. O-15 Consumption of CFCs in refrigerants under Montreal Protocol and no-controls scenarios. (Source: Environment Canada.)

used by hospitals and commercial sterilizers. We estimate that 1989 global consumption of CFC-12 in this application was approximately 23,000 tonnes, with only very small quantities used in Article 5(1) countries.

In this application, CFC-12 was used to reduce the flammability of the EO and accounted for 88 percent of the mixture with EO accounting for 12 percent. The resulting 12/88 mixture was released to the environment at the end of the sterilization cycle.

Many options to reduce and then eliminate the consumption of CFC-12 were pursued in this sector with different costs associated with each. Conservation, for example, led to significant consumption reductions. Sterilizers that previously had

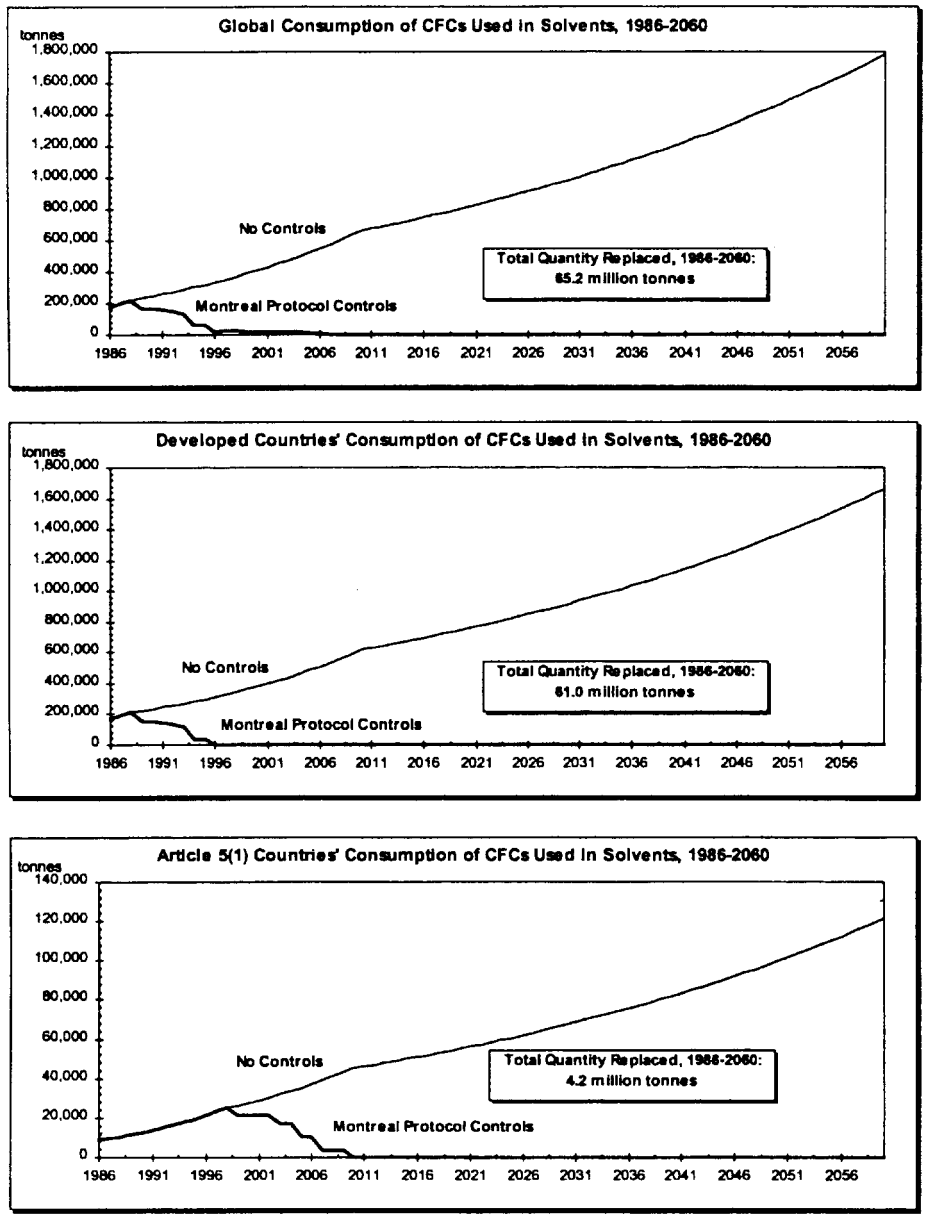


FIG. O-16 Consumption of CFCs in solvents under Montreal Protocol and no-controls scenarios. (Source: Environment Canada.)

been run partly full were run only when full. The major advantage of 12/88 units related to sterilization of medical devices made of polyvinylchloride or polyethylene compounds that melt or deform at the high temperatures of steam sterilization. Restricting the use of 12/88 units to only these items accounted for a further reduction in the use of CFC-12. Large commercial sterilizers were able to convert to a product that was an EO-nitrogen blend to eliminate the use of CFC-12 entirely and some hospitals were able to use small units with only EO. In addition, a drop-in substitute for 12/88 was developed in which the CFC-12 was replaced with HCFC-124.

The preceding paragraph suggests that many options with different control costs

contributed to the phase-out of the use of CFC-12 in sterilant gas. In our assessment of the costs of replacing CFC-12, we have developed a weighted average cost per kilogram reflecting the different costs of each option. The conservation and steam sterilization options involve cost reductions in that expensive CFCs and their replacements did not have to be purchased in the same quantities. The EO conversion of contract sterilizers and the switch to HCFC-124 involved substantial costs. In 1986, for example, the cost of CFC-12 in 12/88 units was approximately $1.50 per kilogram whereas our estimate of the cost of HCFC-124 is approximately $12.00 per kilogram. However, we estimate that HCFC-124 replaced CFC-12 in only 15 percent of sterilant applications.

Overall, we estimate an average cost per kilogram of $1.43 for replacing the use of CFC-12 in sterilization. We estimate that in the absence of the Montreal Protocol, 1989 global consumption of approximately 23,000 tonnes of CFC-12 would have grown at 2 percent per year until 2060. See Fig. O-16. Calculated from the beginning of the impacts of the Montreal Protocol until 2060, we estimate control costs of $1.3 billion in discounted 1997 dollars. Since almost all of the 12/88 systems were found in developed countries, we estimate that $1.2 billion of these costs should be allocated to developed countries with $0.1 billion to Article 5(1) countries.

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.

P

Packaging

The most common materials used in packaging are paper, plastics, and paper coated with plastics. *See* Environmental Accountability; Plastics; Pulp and Paper.

Paper (*see* Pulp and Paper)

Pipe (*see* Some Commonly Used Specifications, Codes, Standards, and Texts)

Plastics

This is an industry that has a vast network of products and producers, as well its own detailed handbooks. For the process engineer, two revolutionary developments on the horizon are worth a look, although neither has yet resulted in gross-scale production.

Today's* growing environmental awareness has forced many businesses to modify their operations and their marketing strategies in response to consumer demands. For instance, a wide range of products available in the market now claim to be "environmentally friendly."

Most kinds of plastic, however, can make no such claim. Widely used to make an almost unlimited number of everyday consumer goods, plastic is generally not biodegradable. As a result, plastic waste has built up, creating a growing disposal problem and posing a threat to the ecology. To make the problem even worse, some types of plastic release potentially harmful chemicals into the atmosphere when incinerated.

In the United States and Western Europe, recycling is becoming the preferred way of "getting rid of" plastic waste. Even in less powerful economies, such as Thailand, some companies have been recycling a limited quantity of plastic for the past few years. But each time plastic is recycled, its quality declines. This means that recycled plastic can mostly be used to manufacture relatively inexpensive, low-quality goods. Another problem with recycling is that it is often difficult to separate plastic waste from other types of refuse. For recycling to work, there must be an efficient garbage sorting system in place. Also, some types of plastic, most notably thermosetting plastics, cannot be recycled at all. This is why researchers have been at work developing another, more environmentally friendly type of plastic, one that is biodegradable.

The processes by which plastic degrades naturally can be divided into these four types:

1. *Environmental degradation.* Environmental degradation is the process by which plastic gradually degrades as a result of surrounding environmental factors, such as sunlight, heat, water, atmospheric conditions, and microorganisms.

---

* Source: Adapted from extracts from Chantravekin, "Biodegradable Plastic: An Alternative for Better Environment," *NPC Focus*, Vol. 5, No. 24, February 1997.

2. *Photodegradation.* This is the process by which plastic slowly degrades in the presence of sunlight or ultraviolet rays.
3. *Oxidative degradation.* This is the degradation that results when plastic reacts chemically with oxygen or ozone in the air.
4. *Biodegradation.* Biodegradation occurs when microorganisms such as bacteria and fungi break down the carbon atoms in the plastic molecules.

The key factors that determine the speed at which plastic degrades include the molecular weight and structure of the plastic; its melting point and crystallinity; the volume of microorganisms, temperature; moisture; pH factor; and the quantity of nutrients in the surrounding environment. Generally, it has been found that plastics with low molecular weight, a low melting point, and a relatively low level of crystallinity or a straight molecular structure degrade more quickly than other types of plastic. Degradation occurs because microorganisms release an enzyme that is used to break down the carbon molecules contained in plastic into carbon dioxide and water in the case of aerobic respiration or into other organic substances by the temperature at which the enzyme can function most cost-effectively.

Generally speaking, naturally occurring biopolymers, such as polypeptides, polynucleotides, and polysaccharides are most subject to natural degradation, whereas synthetic polymers like polyethylene, polystyrene, and polyvinyl chloride are most resistant. However, there are two types of synthetic polymers that are biodegradable. These are aliphatic polyesters, including polycarpolactone, which is used in the manufacture of medical equipment, artificial organs, and contraceptive devices, and polyethylene glycol, which is used to produce lubricants and coating materials.

Polyethylene plastic bags that contain a starch filler are another attempt to produce goods with a minimal impact on the environment. But because polyethylene is highly resistant to the process of natural degradation, only the starch filler degrades when the bags are subjected to soil burial. This is why this particular type of plastic is not classified as biodegradable but rather as biodistegrable. Once the filler has degraded, small pieces of plastic remain, and unlike most polyethylene plastic, these pieces have greatly reduced mechanical properties, such as tensile strength and elongation. Just how reduced these properties are depends on the amount of filler used in the manufacture of the bags.

Biodegradable plastics are attracting considerable attention at present, but because they are still relatively expensive, their use is currently restricted almost exclusively to the medical field. Attempts to extend the use of biodegradable plastics to the production of everyday consumer items are being led by a U.S. company called Bioplastics Inc. The company is a joint project between Michigan State University and the Michigan Biotechnology Institute. The company has experimented with a reactive blending process combining cornstarch and polycarpolactone to produce a polymer alloy resin, which goes by the trade name ENVAR.

ENVAR's most outstanding feature is that it has mechanical properties comparable to those of low-density polyethylene (LDPE), thereby making it suitable for use in the manufacture of plastic film and sheeting. ENVAR can be used to produce consumer items such as plastic garbage bags and shopping bags. What distinguishes ENVAR from polyethylene, however, is that it is biodegradable. In soil burial tests, ENVAR was found to be 100 percent biodegradable, its carbon atoms having been converted into carbon dioxide after only three weeks. ENVAR's other principal advantage is that because cornstarch is an inexpensive commodity, the biodegradable plastic is cheaper than synthetic polycarpolactone. This should help to make the dream of using biodegradation plastics in everyday lives a reality.

Two chief advantages could be gained by adapting biodegradable plastics technology to, for instance, countries like Thailand. First, the problem of safety disposing of plastic waste would be greatly reduced. The country's landfills would be less severely taxed by the thousands of tons of plastic waste that are sent for burial every day. Second, the country would find an answer to the trade barriers erected by the European Community against Thai exports of tapioca starch. By using the native-grown starch in the production of biodegradable plastic, the country would add greatly to the value of this crop. To do this, it would be necessary first to submit the starch to a process of plasticization to obtain thermoplastic tapioca starch. This then would be combined with polycarpolactone in a reactive blending process to produce biodegradable plastic resin with similar properties to ENVAR resin. The major drawback to this proposal, however, is that because polycarpolactone is a specialty chemical, it is relatively expensive. This would, in turn, make the production of biodegradable plastic more expensive than other types of plastic.

Nevertheless, it will probably not be long before we begin to see biodegradable plastic goods in markets around the world. It will take cooperation between the private and government sectors and a joint commitment to improving environmental conditions.

Electric Plastics*

Engineers at AGFA in Köln, Germany, were facing a critical problem with the production of photofilm in the late 1980s. Static discharges were ruining the huge, costly rolls of the company's film; induced by friction, the little electric sparks generated big losses. The engineers' investigation showed that the inorganic salts AGFA traditionally used as an antistatic coating failed to work when the humidity dropped below 50 percent. These water-soluble ionic compounds also washed away after developing, again leaving the photofilm vulnerable to stray sparks.

AGFA turned to parent company Bayer AG in Krefeld, Germany, to see whether its central research arm could develop a new low-cost antistatic agent. The antistatic coating had to operate independent of air humidity, with a surface resistance greater than 10^8 ohms square; it had to be transparent and free of heavy metals; and it had to be produced from a waterborne solution.

The most promising candidate to fill these criteria was, surprisingly, an electrically conductive polymer material known as polythiophene. Such polymers have always had great commercial potential because of their unusual ability (for a plastic) to provide a path for electrons, but they had not found any wide commercial applications to that point.

Following a thorough development effort involving the selection of the ideal polythiophene derivative, its subsequent synthesis, and its polymerization, the Bayer research team succeeded in inventing an aqueous processing route for the plastic coating. As of early 1998, more than 10,000 m^2 of AGFA photographic film had been coated with the conductive polymer.

Now the chemical company is marketing the polythiophene under the trade name Baytron. The material could also be used to make plastics paintable by adding the conductive agent first, or in the electrodes of small, high-performance tantalum capacitors found in telecommunications, computer, and automotive products.

---

* Source: Adapted from extracts from Ashley, "Electric Plastics," *Mechanical Engineering*, ASME, April 1998.

Another significant potential application is in the through-hole plating of circuit boards. The chemical process of depositing the initial layers of copper into these holes requires formaldehyde, a known carcinogen. Blasberg Oberflae-chentechnik in Soligen, Germany, has patented a method using polythiophene as the first coat instead of the electroless copper. The new plating technology has been licensed to several Japanese circuit-board makers and to Enthone Inc., a subsidiary of ASARCO Inc. in West Haven, Conn.

Long-time researchers on conductive polymers point to Bayer's Baytron polythiophene as the most notable success story in the field. As with most new materials, finding sufficient demand is the key to convincing manufacturers to go into full-scale production. Antistatic applications have a huge potential, but conductive polymers have yet to make many inroads. The once highly acclaimed technology has been reduced to the point that the only successful large application—antistatic coatings for AGFA photofilm—is for internal company use.

Companies from Alstom and AlliedSignal to Westinghouse and W.R. Grace have tried to make conductive polymers into a success, but they have reportedly curtailed or aborted their research. Even though one application for the material—flat-panel displays for televisions and computers—is starting to involve researchers again, much of the payoff for this technology lies in the future.

That future looked a lot brighter for conductive polymers in the 1980s. Probably the most significant commercialization of conductive polymers was for flexible, long-lived batteries that were produced in quantity by Bridgestone Corp. and Seiko Co. in Japan and by BASF/Varta in Germany. Fifteen years ago, when they first came to the market, interest in conductive-polymer batteries was high. In the end, though the batteries worked, they were difficult to sell because their costs were not significantly lower than those of the competition. So the battery products were withdrawn due to insufficient demand. (Researchers at the Johns Hopkins Applied Physics Laboratory in Baltimore recently developed a nontoxic, flexible, all-plastic battery made from another class of conductive plastics called fluorophenylthiophenes, but little is expected of the technology.)

Another once-promising product incorporating conductive polymers is Contex, a fiber that has been manufactured by Milliken & Co. in Spartanburg, S.C., since 1990. The fiber is coated with a conductive-polymer material called polypyrrole and can be woven to create an antistatic fabric. Milliken had been interested in using this type of antistatic technology for its carpet products.

The material's best chance for success was in military applications. Polypyrrole was approved for use in the U.S. Navy's A-12 stealth attack carrier aircraft. The polymer was to be used in edge cards—components that dissipate incoming radar energy by conducting electric charge across a gradient of increasing resistance that the plastic material produces. The A-12 program has been canceled, however.

Milliken also tried to market ultralight camouflage netting based on Contex to help conceal military equipment and personnel from near-infrared and radar detection, but the company lost a U.S. Army contract for conductive camouflage material in 1997. Despite a recent modest contract with NASA to produce conductive-polymer electromagnetic shielding for the space shuttle, Milliken's research program was in financial jeopardy by early 1998.

Despite ups and downs, electrically conductive polymers have attracted a substantial amount of attention since they were accidentally discovered two decades ago, when a Tokyo Institute of Technology student added too much catalyst to a batch of polyacetylene. When the resulting silvery film was later doped with various oxidizing agents at the University of Pennsylvania in Philadelphia, it became conductive, and the race was on to invent new conductive polymers.

Conductive polymers are long, carbon-based chains composed of simple repeating units called monomers. When the Japanese student made his fortuitous error, he

TABLE P-1 Electrical Conductivity of Conductive Polymers and Other Conducting Materials

Material (Year Conductivity Discovered)	Amperes Conducted per Volt Centimeter
Conducting Polymers	
Polyacetylene (1977)	1.7×10^5
Polypyrrole (1979)	7.5×10^3
Poly-*p*-phenylene (1979)	1×10^3
Poly-*p*-phenylenevinylene (1979)	5×10^3
Polyaniline (1980)	2×10^2
Polythiophene (1981)	1×10^3
Other Conducting Materials	
Copper—good conductor	5.8×10^7
Silicon—semiconductor	4×10^{-4}
Quartz—insulator	2×10^{-17}

converted the standard single-bond carbon chains to polymer backbones with alternating single and double bonds, a change that provided a pathway for free-electron-charge carriers. To make the altered polymer materials conductive, they are doped with atoms that donate negative or positive charges (oxidizing or reducing agents) to each unit, enabling current to travel down the chain. Depending on the dopant, conductive polymers exhibit either *p*- or *n*-type conductivity.

The most extensively studied conductive-polymer systems are based on polyaniline, polythiophene, polypyrrole, and polyacetylene. The principal attractions of these polymers over conventional conducting materials are their potential ease of processing, relative robustness, and light weight. Successful commercial applications require a fine balance of conductivity, processability, and stability, but until recently, materials researchers could not obtain all three properties simultaneously.

Conductive polymers are much more electrically conductive than standard polymers but much less than metals such as copper. (See Table P-1.) In practice, the conductivity of these materials is characterized by low-charge carrier mobility—a measure of how easily electric charge moves. This characteristic limits response speed in the case of a transistor, for example, making such a device rather inefficient.

Still, efforts to produce semiconductor devices from conductive polymers are proceeding. In 1994, a team at the Laboratory of Molecular Materials in Thais, France, made a field-effect transistor from polythiophene using printing techniques. Rolling up, bending, and twisting did not affect the transistor's electrical characteristics.

The opportunity to produce relatively low-cost semiconductor devices that are insensitive to mechanical deformation is an attractive one. Probably the most exciting development in this area is the intensifying effort to use conductive polymers to produce flat, flexible plastic screens for TVs and computers. See Fig. P-1. This screen technology emerged from the discovery that certain conductive polymers, such as poly-*p*-phenylenevinylene, emit light when sandwiched between oppositely charged electrodes, a configuration that fits in well with current flat-panel display designs.

The current leader in this work is Cambridge Display Technology (CDT) in Cambridge, England. CDT recently entered into a collaboration with Japanese electronics maker Seiko-Epson to develop light-emitting polymer screens. Philips Electronics NV in the Netherlands is also working on a portable telephone using such a display. Other licensees include Hoechst AG in Germany and Uniax Corp. in Santa Barbara, Calif. While it is likely to be some time before this technology

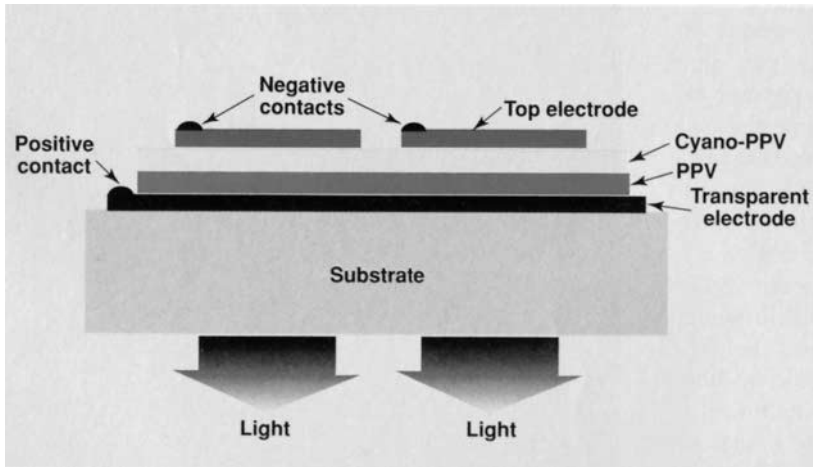


FIG. P-1 Flat-panel display technology for televisions and computers using poly-*p*-phenylenevinylene (PPV) has emerged as one of the most promising applications for conductive polymers. Cambridge Display Technologies in Cambridge, England, is the current leader in this area. (Source: Ashley.)

makes it to the market in flexible flat-panel screens, the development work has created new interest in conductive polymers.

Another promising application is in capacitor technology, where there has been good progress, due mainly to federal funding of ultracapacitors for future electric vehicles. Kemet Electronics Corp. in Greenville, S.C., is working on using polythiophene or polypyrrole to replace manganese dioxide counterelectrodes in tantalum surface-mount capacitors, which are widely used in the electronics industry. Conductive polymers can provide lower equivalent-series resistance (ESR). With the designers of mobile electronics constantly being pushed for space, the new capacitors can simultaneously be smaller and have a lower ESR.

Kemet is operating a small pilot line to produce the electrodes. Kemet predicted that a capacitor product using conductive polymer would be available shortly. High-volume production could follow.

Yet another emerging application for electrically conductive polymer materials is biosensors and chemical sensors, which can convert chemical information into a measurable electrical response. Abtech Scientific Inc. in Yardley, Pa., is making chemical transducers from mostly polyaniline as well as polythiophene and polypropylene for analytical applications in which one measures conductivity and as a result infers what the chemical composition is. In other words, a very small change in the redox composition brought about by small quantities of a range of chemicals can induce a large, rapid change in electrical conductivity.

The challenge for Abtech is how to confer specificity to these materials. One way is to build biopolymer/conductive-polymer complexes. Using this technique, Abtech has developed a range of enzyme biosensors. For example, immobilized glucose oxidase can be incorporated into this polymer transducer system, which acts like a glucose-sensitive biosensor, as the enzyme-catalyzed oxidation of the glucose produces an oxidant by-product that is measured indirectly. Levels of therapeutic drugs in patients can also be monitored in a similar way.

Abtech is developing the technology for point-of-care testing by physicians, a market that is of great interest to several major medical-product companies. The disposable point-of-care-testing product will be used to make many medical tests much cheaper.

An area with some further-off potential—smart membranes of conductive polymers—is being pursued by a team at Los Alamos National Laboratory in Los Alamos, N.M. The lab's Chemical Sciences and Technical Development Division has developed engineered porous-fiber materials with electrically controlled porosity using polyaniline. The technology could find use in gas separation, pharmaceutical separation, environmental cleanup, batteries, or capacitors. A spin-off company to develop the technology already has been established.

The list of potential applications for conductive polymers remains a long one, and includes antiradiation coatings, batteries, catalysts, deicer panels, electrochromic windows, electromechanical actuators, embedded-array antennas, fuel cells, lithographic resists, nonlinear optics, radar dishes, and wave guides. Just how big an impact the materials will make in these markets remains unclear, however. Most observers are putting their money on antistatic coatings and flat-panel displays. Abtech predicts that neither application is going to be a "big winner" initially because they are displacing other established approaches, but they do have promise, especially if they are not oversold.

Pollutants, Chemical; Pollutants, (from) Chemical Processes; Pollutant Indicators; Pollutants, Toxic; Pollutants, Toxic Chemicals (*see also* Emissions; Environmental Accountability)

Toxic chemical pollutants have two main damaging effects to life and the environment:

1. These pollutants generally have a long half-life and can remain unchanged for decades and, in some cases, centuries. They can then move from air to land to water and find their way into the systems of humans or other living organisms. During the winter, some toxins may condense on the ground, and then reenter the atmosphere when the ground warms up.
2. Toxic chemicals can accumulate in fatty tissue of humans and other animals. These chemical concentrations increase as you go higher up the food chain. So, for instance, if we consider polychlorinated biphenyls (PCBs) that migrate, airborne (and via the air they then contaminate water and land), to arctic latitudes, the level of PCBs is higher in polar bears and people (the Inuit) than it is in seals and fish. The bears and the humans are at the top of the food chain.

Toxins travel well, unfortunately. Toxaphene, a pesticide used in the cotton fields of the southern United States, is found in the arctic. So are lead, dioxin, DDT, chlordane, and lindane (industrial chemicals and pesticides). The bad news is that the presence of these compounds in the arctic means they are already well distributed in areas where more people live.

Pollutant indicators are measurements of concentrations of these pollutants in air, water, or soil. The governing environmental bodies in developed countries track these concentrations per indicator that is "tagged" for observation.

Pollutants know no national boundaries, as is illustrated well in the case of the Great Lakes, which are both in Canada and the United States.

The following material is provided to give the reader an indication of how vast the issue of pollutants is. It is relevant that this material—although not comprehensive in terms of the overall subject—be provided. This is because improvements in pollutant measurement technology and constant changes in legislation can make a process plant the potential recipient of an order to shut down or curtail operations. Massive changes in process systems, requiring retrofits and

reengineering, may result. Increasingly, the process engineer must also be, in part, an environmental—or at least an environmentally well intentioned—engineer. If this material raises more questions than it answers, the process engineer may eventually be better off, as this could prompt further reading (see additional reading recommended for this and other related entries).

Note also that the information source Environment Canada is an advisory entity and cannot itself prosecute plants not in compliance with legislated limits. Its information is good, easily available on request from the source, and an excellent benchmark in most cases for end users wanting to set up their own monitoring standards, as well as check on their own. Note also that the U.S. Environmental Protection Agency (EPA) is a regulatory body and can penalize and fine offending companies. The information provided in this book, therefore, should be used as a condensed basis for asking questions in the country the process engineer is working in—not as "complete" in itself.

Country Case Study (Canada)*

The extracts from the "NPRI Summary Report, 1994" that follow include some of the process industry data released by the National Pollutant Release Inventory (NPRI) by Environment Canada.

The NPRI was established to provide a national, publicly accessible database of pollutants released to the Canadian environment from industrial and transportation sources. This second Summary Report reflects the data reported to the NPRI for calendar year 1994 and estimated for various sectors.

Anyone in Canada who owns or operates a facility with 10 or more full-time employees in the reporting year and that manufactures, processes, or otherwise uses any of the NPRI-listed substances, in concentration equal to or greater than 1 percent and in quantities equal to or greater than 10 tonnes (10,000 kg), must file a report with Environment Canada and identify any releases or transfers in waste of those substances to air, water, or land.

What are releases?

A release is an on-site discharge of a substance to the environment. This includes emissions to air, discharges to surface waters, releases to land within the boundaries of the facility, and deep-well injection.

Releases are further subdivided as follows:

- Air
  - Stack/point
  - Storage/handling
  - Fugitive
  - Spills
  - Other nonpoint
- Surface water
  - Direct
  - Spills
  - Leaks

* Source: Environment Canada.

- Land
  - Landfill
  - Land treatment
  - Spills
  - Leaks
  - Other
- Underground injection

Land treatment, also called application farming, is a disposal method in which a waste containing a listed substance is applied or incorporated into soil for biological degradation. Landfills are sites in which wastes are buried. These two disposal methods are generally conducted under permit.

A leak differs from a spill in terms of the time required for an event. Spills normally occur over a period of hours to days, whereas leaks occur over periods of days to months.

Underground injection is another method of waste disposal. Wastes are injected into known geological formations, generally at great depths. This disposal method is subject to provincial regulation.

What are transfers?

A transfer is a shipment of a listed substance in waste to an off-site location. Facilities must provide the name and location of the off-site facility receiving the shipment. Waste is defined as material that is sent for final disposal or for treatment prior to final disposal. There are seven off-site disposal or treatment methods:

- Physical treatment such as encapsulation and vitrification
- Chemical treatment such as stabilization and neutralization
- Biological treatment such as bio-oxidation
- Incineration or thermal treatment
- Containment in a landfill and other storage
- Municipal sewage treatment plant
- Underground injection

Off-site transfers in waste are reported separately from on-site releases because:

- Off-site transfers represent a movement of the substance to a different geographic location than that of the facility
- Transfers off site may not necessarily represent entry of the substance into the environment
- Management of the substance becomes the responsibility of another owner or operator
- Reporting on off-site transfers completes information on the fate of the substance
- Wastes could be transferred a number of times leading to some double counting

The NPRI requires that only the quantity of the listed substance in the waste be reported. Waste materials, such as sludges, are often a mixture of many compounds associated with water and other inert material with a small quantity of potentially hazardous substances. As a result, the total reported to the NPRI may be smaller than the quantity reported in other inventories since only the net weight of a listed substance is reported.

What is recovery, re-use, and recycling?

As with transfers, recovery, re-use, and recycling (3Rs) represent a movement of the substance to an off-site facility, generally under the jurisdiction of another owner or operator. Facilities were required to report the name and the address of the receiving off-site facility.

The definition of waste for the 1994 reporting year has been modified from that of 1993. For 1994 reporting, waste excluded substances sent for the 3Rs. These 3Rs could be reported voluntarily under another section of the reporting form.

Generally, materials sent for 3Rs are those transferred to recyclers, such as metal shavings or turnings, those materials transferred off-site for processing, cleaning, or reclamation and returned to the facility, and those materials sent back to the suppliers for credit or payment.

Energy recovery is applicable only when recuperated energy from combustion is used as an alternative to fossil fuels or other forms of energy.

Extracts from Appendices 4 through 11 of NPRI Summary Report, 1994

Appendix 4: List of NPRI substances and releases (tonnes)

CAS #	Substance Name	Total Releases
75-07-0	Acetaldehyde	133.287
67-64-1	Acetone	3,841.890
75-05-8	Acetonitrile	79.130
79-06-1	Acrylamide	5.878
79-10-7	Acrylic acid	0.889
107-13-1	Acrylonitrile	19.556
7429-90-5	Aluminum (fume or dust)	60.326
1344-28-1	Aluminum oxide (fibrous forms)	34.609
7664-41-7	Ammonia	26,465.316
6484-52-2	Ammonium nitrate (solution)	1,900.121
7783-20-2	Ammonium sulfate (solution)	1,193.082
62-53-3	Aniline	0.005
120-12-7	Anthracene	2.517
NA	Antimony (and its compounds)	19.398
NA	Arsenic (and its compounds)	3,980.656
1332-21-4	Asbestos	352.184
71-43-2	Benzene	2,675.468
94-36-0	Benzoyl peroxide	0.000
100-44-7	Benzyl chloride	0.142
92-52-4	Biphenyl	15.435
103-23-1	*Bis*(2-ethylhexyl) adipate	7.276
117-81-7	*Bis*(2-ethylhexyl) phthalate	92.804
106-99-0	1,3-Butadiene	310.180
141-32-2	Butyl acrylate	1.141
78-83-1	*i*-Butyl alcohol	103.616
71-36-3	*n*-Butyl alcohol	1,248.821
78-92-2	*sec*-Butyl alcohol	0.000
75-65-0	*tert*-Butyl alcohol	91.512
85-68-7	Butyl benzyl phthalate	3.699
4680-78-8	C.I. Acid Green 3	0.000
NA	Cadmium (and its compounds)	96.041
156-62-7	Calcium cyanamide	0.000
75-15-0	Carbon disulfide	26.168
56-23-5	Carbon tetrachloride	18.135
7782-50-5	Chlorine	2,086.098
10049-04-4	Chlorine dioxide	1,735.537

CAS #	Substance Name	Total Releases
79-11-8	Chloroacetic acid	0.000
75-00-3	Chloroethane	336.931
67-66-3	Chloroform	0.172
74-87-3	Chloromethane	1,177.083
NA	Chromium (and its compounds)	800.859
NA	Cobalt (and its compounds)	29.182
NA	Copper (and its compounds)	16,069.629
1319-77-3	Cresol (mixed isomers)	2.224
95-48-7	*o*-Cresol	0.001
98-82-8	Cumene	28.859
80-15-9	Cumene hydroperoxide	0.020
NA	Cyanides (ionic)	17.008
110-82-7	Cyclohexane	2,730.773
1163-19-5	Decabromodiphenyl oxide	0.000
84-74-2	Dibutyl phthalate	12.258
95-50-1	*o*-Dichlorobenzene	0.480
106-46-7	*p*-Dichlorobenzene	10.400
107-06-2	1,2-Dichloroethane	7.579
75-09-2	Dichloromethane	2,222.089
111-42-2	Diethanolamine	790.216
84-66-2	Diethyl phthalate	0.000
77-78-1	Dimethyl sulfate	0.011
534-52-1	4,6-Dinitro-*o*-cresol	0.000
121-14-2	2,4-Dinitrotoluene	4.000
117-84-0	Di-*n*-octyl phthalate	0.480
123-91-1	1,4-Dioxane	9.144
106-89-8	Epichlorohydrin	0.133
110-80-5	2-Ethoxyethanol	2.354
111-15-9	2-Ethoxyethyl acetate	92.003
140-88-5	Ethyl acrylate	1.120
100-41-4	Ethylbenzene	537.561
74-85-1	Ethylene	2,558.541
107-21-1	Ethylene glycol	3,333.492
75-21-8	Ethylene oxide	51.067
50-00-0	Formaldehyde	1,116.417
302-01-2	Hydrazine	1.242
7647-01-0	Hydrochloric acid	2,293.897
74-90-8	Hydrogen cyanide	0.130
7664-39-3	Hydrogen fluoride	28.597
123-31-9	Hydroquinone	0.021
67-63-0	Isopropyl alcohol	12,132.317
80-05-7	*p*,*p*′-Isopropylidenediphenol	0.169
NA	Lead (and its compounds)	2,142.220
108-31-6	Maleic anhydride	27.609
NA	Manganese (and its compounds)	3,087.366
NA	Mercury (and its compounds)	3.806
67-56-1	Methanol	30,403.335
109-86-4	2-Methoxyethanol	17.000
96-33-3	Methyl acrylate	2.112
1634-04-4	Methyl *tert*-butyl ether	28.157
101-14-4	*p*,*p*′-Methylene*bis*(2-chloroaniline)	0.005
101-68-8	Methylene*bis*(phenylisocyanate)	1.914
78-93-3	Methyl ethyl ketone	4,448.431
108-10-1	Methyl isobutyl ketone	894.763
80-62-6	Methyl methacrylate	32.442
1313-27-5	Molybdenum trioxide	44.094
91-20-3	Naphthalene	84.592
NA	Nickel (and its compounds)	704.496
7697-37-2	Nitric acid	64.920
139-13-9	Nitrilotriacetic acid	1.001
55-63-0	Nitroglycerin	9.000
79-46-9	2-Nitropropane	0.125
79-21-0	Peracetic acid	0.000
108-95-2	Phenol	205.405

CAS #	Substance Name	Total Releases
90-43-7	*o*-Phenylphenol	0.000
7664-38-2	Phosphoric acid	820.110
7723-14-0	Phosphorus (yellow or white)	14.611
85-44-9	Phthalic anhydride	13.250
115-07-1	Propylene	1,188.704
75-56-9	Propylene oxide	11.071
NA	Selenium (and its compounds)	8.208
NA	Silver (and its compounds)	4.674
100-42-5	Styrene	1,792.518
96-09-3	Styrene oxide	0.130
7664-93-9	Sulfuric acid	22,294.515
127-18-4	Tetrachloroethylene	163.335
62-56-6	Thiourea	0.000
7550-45-0	Titanium tetrachloride	0.023
108-88-3	Toluene	7,723.317
584-84-9	Toluene-2,4-diisocyanate	0.148
26471-62-5	Toluenediisocyanate (mixed isomers)	1.033
79-00-5	1,1,2-Trichloroethane	0.000
79-01-6	Trichloroethylene	859.483
95-63-6	1,2,4-Trimethylbenzene	398.468
7440-62-2	Vanadium (fume or dust)	30.552
108-05-4	Vinyl acetate	126.078
75-01-4	Vinyl chloride	23.725
75-35-4	Vinylidene chloride	0.000
1330-20-7	Xylene (mixed isomers)	8,878.386
108-38-3	*m*-Xylene	18.764
95-47-6	*o*-Xylene	11.775
106-42-3	*p*-Xylene	16.274
NA	Zinc (and its compounds)	10,006.590
7440-66-6	Zinc (fume or dust)	656.034
	Total released	**190,263.945**

Appendix 5: NPRI substances released by two-digit SIC code (tonnes)

02—Service industries incidental to agriculture

Substance Name	Air	Water	Underground	Land	Total Releases*
Cobalt (and its compounds)	0.000	0.000	0.000	0.000	0.000
Copper (and its compounds)	0.000	0.000	0.000	0.000	0.000
Manganese (and its compounds)	0.000	0.000	0.000	0.000	0.000
Selenium (and its compounds)	0.000	0.000	0.000	0.000	0.000
Zinc (and its compounds)	0.000	0.000	0.000	0.000	0.000
Total	**0.000**	**0.000**	**0.000**	**0.000**	**0.000**

NOTE: See p. P-31 for footnote.

03—Fishing and trapping industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Chlorine	0.000	0.000	0.000	0.000	0.000
Total	**0.000**	**0.000**	**0.000**	**0.000**	**0.000**

04—Logging industry

Substance Name	Air	Water	Underground	Land	Total Releases*
Methanol	77.900	0.000	0.000	2.800	80.700
Sulfuric acid	10.240	0.000	0.000	0.000	10.240
Chlorine dioxide	7.169	0.000	0.000	0.000	7.169
Ammonia	1.650	0.830	0.000	0.000	2.480
Chlorine	0.680	0.000	0.000	0.000	0.680
Phosphoric acid	0.000	0.000	0.000	0.000	0.000
Total	**97.639**	**0.830**	**0.000**	**2.800**	**101.269**

06—Mining industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Copper (and its compounds)	79.183	14,431.987	0.000	0.473	14,512.520
Zinc (and its compounds)	141.500	4,489.490	0.000	0.492	4,632.582
Arsenic (and its compounds)	38.296	45.489	3,800.000	0.000	3,883.785
Ammonia	570.560	436.503	0.000	0.000	1,007.064
Sulfuric acid	82.457	719.000	0.000	25.000	826.717
Phosphoric acid	0.000	774.000	0.000	0.000	774.000
Ammonium sulphate (solution)	95.000	577.000	0.000	0.000	672.720
Acetone	377.250	0.000	0.000	0.000	377.250
Lead (and its compounds)	174.654	154.108	0.000	0.025	329.454
Methanol	0.000	0.000	0.000	37.400	37.400
Ethylene glycol	27.070	0.000	0.000	6.400	33.470
Hydrochloric acid	0.000	0.000	7.000	13.050	20.240
Cyanides (ionic)	8.467	6.705	0.000	0.000	15.776
Cadmium (and its compounds)	6.665	6.581	0.000	0.000	13.246
Antimony (and its compounds)	9.600	3.300	0.000	0.000	12.900
Manganese (and its compounds)	0.202	11.812	0.000	0.000	12.014
Nitric acid	10.560	0.000	0.000	0.000	10.690
Selenium (and its compounds)	0.185	3.896	0.000	0.000	4.081
Mercury (and its compounds)	3.580	0.148	0.000	0.000	3.728
Silver (and its compounds)	0.130	3.320	0.000	0.000	3.670
Nickel (and its compounds)	0.601	2.911	0.000	0.000	3.512
Zinc (fume or dust)	3.350	0.032	0.000	0.000	3.493
Hydrogen fluoride	2.969	0.132	0.000	0.000	3.102
Ammonium nitrate (solution)	1.048	0.000	0.000	0.000	1.071
Asbestos	0.000	0.000	0.000	0.000	0.587
Chlorine	0.160	0.000	0.000	0.000	0.160
Acrylamide	0.000	0.000	0.000	0.000	0.130
Isopropyl alcohol	0.000	0.000	0.000	0.000	0.000
Molybdenum trioxide	0.000	0.000	0.000	0.000	0.000
Methyl isobutyl ketone	0.000	0.000	0.000	0.000	0.000
Chromium (and its compounds)	0.000	0.000	0.000	0.000	0.000
Cobalt (and its compounds)	0.000	0.000	0.000	0.000	0.000
Carbon disulfide	0.000	0.000	0.000	0.000	0.000
Total	**1,633.487**	**21,666.414**	**3,807.000**	**82.840**	**27,195.362**

07—Crude petroleum and natural gas industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Hydrochloric acid	0.020	0.000	924.00	0.000	924.020
Diethanolamine	2.940	3.800	694.713	1.028	702.511
Methanol	18.269	0.000	496.854	4.222	522.120
Ethylene glycol	70.721	8.690	390.132	38.200	508.623

Substance Name	Air	Water	Underground	Land	Total Releases*
Toluene	260.407	0.008	15.860	0.365	283.157
Xylene (mixed isomers)	234.722	0.009	1.043	0.757	242.664
Benzene	175.608	0.008	45.900	0.459	228.494
Ammonia	44.145	1.600	48.600	0.000	94.345
Propylene	79.161	0.000	0.000	0.000	79.291
Cyclohexane	46.891	0.000	0.000	0.084	53.171
Asbestos	0.000	0.000	0.000	51.000	51.000
Ethylene	50.660	0.000	0.000	0.000	50.790
1,2,4-Trimethylbenzene	45.543	0.000	0.000	0.552	49.921
Molybdenum trioxide	0.000	0.000	0.000	41.800	41.800
Nickel (and its compounds)	5.234	0.449	0.000	27.190	32.873
Ethylbenzene	30.870	0.000	0.000	0.158	32.638
Vanadium (fume or dust)	19.040	0.000	0.000	0.000	19.040
Chlorine	5.320	0.394	11.670	0.000	17.384
Isopropyl alcohol	0.130	0.000	2.702	0.000	2.832
Naphthalene	2.580	0.000	0.000	0.170	2.750
Cumene	2.240	0.000	0.000	0.044	2.284
Zinc (and its compounds)	0.000	0.000	0.000	0.000	0.880
Carbon disulphide	0.448	0.000	0.000	0.000	0.448
Sulfuric acid	0.130	0.000	0.000	0.000	0.220
o-Xylene	0.000	0.000	0.000	0.000	0.219
Methyl ethyl ketone	0.000	0.000	0.190	0.000	0.190
Manganese (and its compounds)	0.170	0.000	0.000	0.000	0.170
Copper (and its compounds)	0.000	0.000	0.000	0.000	0.130
Aluminum oxide (fibrous forms)	0.000	0.000	0.000	0.000	0.000
Phosphoric acid	0.000	0.000	0.000	0.000	0.000
Cobalt (and its compounds)	0.000	0.000	0.000	0.000	0.000
Total	**1,095.249**	**14.958**	**2,631.664**	**166.029**	**3,943.965**

09—Service industries incidental to mineral extraction

Substance Name	Air	Water	Underground	Land	Total Releases*
Xylene (mixed isomers)	3.553	0.000	0.000	0.000	3.553
Methanol	1.247	0.000	0.000	0.000	1.247
Isopropyl alcohol	1.083	0.000	0.000	0.000	1.083
Toluene	0.783	0.000	0.000	0.000	0.783
Styrene	0.728	0.000	0.000	0.000	0.728
1,2,4-Trimethylbenzene	0.293	0.000	0.000	0.000	0.293
Ethylene glycol	0.197	0.000	0.000	0.000	0.197
Ethylbenzene	0.046	0.000	0.000	0.000	0.046
Total	**7.930**	**0.000**	**0.000**	**0.000**	**7.930**

10—Food industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Hydrochloric acid	0.000	16.900	0.000	0.000	17.835
Ammonia	10.000	0.000	0.000	0.000	10.130
Nitric acid	10.000	0.000	0.000	0.000	10.000
Phosphoric acid	0.000	0.000	0.000	0.000	0.130
Copper (and its compounds)	0.000	0.000	0.000	0.000	0.000
Manganese (and its compounds)	0.000	0.000	0.000	0.000	0.000
Selenium (and its compounds)	0.000	0.000	0.000	0.000	0.000
Zinc (and its compounds)	0.000	0.000	0.000	0.000	0.000
Nickel (and its compounds)	0.000	0.000	0.000	0.000	0.000

Substance Name	Air	Water	Underground	Land	Total Releases*
Peracetic acid	0.000	0.000	0.000	0.000	0.000
Formaldehyde	0.000	0.000	0.000	0.000	0.000
Chlorine dioxide	0.000	0.000	0.000	0.000	0.000
Sulfuric acid	0.000	0.000	0.000	0.000	0.000
Trichloroethylene	0.000	0.000	0.000	0.000	0.000
Chlorine	0.000	0.000	0.000	0.000	0.000
Total	**20.000**	**16.900**	**0.000**	**0.000**	**38.095**

11—Beverage industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Ammonia	5.200	0.000	0.000	0.000	5.200
Methanol	2.220	0.000	0.000	0.000	2.220
Isopropyl alcohol	1.480	0.000	0.000	0.000	1.480
Toluene	1.265	0.000	0.000	0.000	1.265
Acetone	0.910	0.000	0.000	0.000	0.910
Tetrachloroethylene	0.000	0.000	0.000	0.000	0.630
Dichloromethane	0.000	0.000	0.000	0.000	0.630
Methyl ethyl ketone	0.000	0.000	0.000	0.000	0.380
n-Butyl alcohol	0.000	0.000	0.000	0.000	0.380
Xylene (mixed isomers)	0.000	0.000	0.000	0.000	0.380
Chloroethane	0.000	0.000	0.000	0.000	0.130
Ethylene glycol	0.000	0.000	0.000	0.000	0.130
Trichloroethylene	0.000	0.000	0.000	0.000	0.130
Nitric acid	0.000	0.000	0.000	0.000	0.000
Formaldehyde	0.000	0.000	0.000	0.000	0.000
Phosphoric acid	0.000	0.000	0.000	0.000	0.000
Sulfuric acid	0.000	0.000	0.000	0.000	0.000
Total	**11.075**	**0.000**	**0.000**	**0.000**	**13.865**

15—Rubber products industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Xylene (mixed isomers)	1,273.236	0.000	0.000	0.000	1,273.236
Toluene	670.091	0.000	0.000	0.000	670.091
Methyl ethyl ketone	125.664	0.000	0.000	0.000	125.664
Trichloroethylene	97.863	0.000	0.000	0.000	97.863
Bis(2-ethylhexyl) phthalate	11.957	0.000	0.000	12.624	24.581
Zinc (and its compounds)	1.857	0.391	0.000	15.300	19.175
Methyl isobutyl ketone	16.941	0.000	0.000	0.000	16.941
Asbestos	0.000	0.000	0.000	10.800	10.800
Ethylbenzene	10.267	0.000	0.000	0.000	10.267
Sulfuric acid	2.200	0.000	0.000	0.000	2.200
Aluminum oxide (fibrous forms)	0.000	0.000	0.000	1.854	1.854
Styrene	0.000	0.000	0.000	0.000	0.500
Ammonia	0.000	0.421	0.000	0.000	0.421
Lead (and its compounds)	0.298	0.000	0.000	0.000	0.298
Methylene*bis*(phenylisocyanate)	0.000	0.000	0.000	0.000	0.026
Copper (and its compounds)	0.000	0.012	0.000	0.000	0.012
Cobalt (and its compounds)	0.000	0.000	0.000	0.000	0.000
Bis(2-ethylhexyl) adipate	0.000	0.000	0.000	0.000	0.000
Di-*n*-octyl phthalate	0.000	0.000	0.000	0.000	0.000
1,3-Butadiene	0.000	0.000	0.000	0.000	0.000
Acrylonitrile	0.000	0.000	0.000	0.000	0.000

Substance Name	Air	Water	Underground	Land	Total Releases*
Hydrochloric acid	0.000	0.000	0.000	0.000	0.000
Zinc (fume or dust)	0.000	0.000	0.000	0.000	0.000
Epichlorohydrin	0.000	0.000	0.000	0.000	0.000
Total	**2,210.374**	**0.824**	**0.000**	**40.578**	**2,253.929**

16—Plastic products industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Dichloromethane	1,356.821	0.000	0.000	0.000	1,356.821
Methyl ethyl ketone	1,195.662	0.000	0.000	0.000	1,195.917
Toluene	385.113	0.000	0.000	0.000	385.748
Isopropyl alcohol	303.485	0.000	0.000	0.000	303.610
Styrene	198.445	0.380	0.000	0.000	200.720
Acetone	177.062	0.000	0.000	0.000	177.317
Chloroethane	162.331	0.000	0.000	0.000	162.331
Methanol	109.702	1.970	0.000	0.000	111.802
n-Butyl alcohol	89.117	18.680	0.000	0.000	108.052
2-Ethoxyethyl acetate	92.000	0.000	0.000	0.000	92.000
Formaldehyde	0.173	85.450	0.000	0.000	85.753
Bis(2-ethylhexyl) phthalate	8.730	0.000	0.000	55.847	64.608
Trichloroethylene	33.995	0.000	0.000	0.000	33.995
Ammonia	0.090	17.880	0.000	0.000	18.126
1,3-Butadiene	17.529	0.000	0.000	0.000	17.529
Bis(2-ethylhexyl) adipate	0.000	5.600	0.000	1.660	7.260
i-Butyl alcohol	0.012	6.440	0.000	0.000	6.582
Methyl methacrylate	4.545	0.000	0.000	0.000	4.545
Xylene (mixed isomers)	0.749	0.752	0.000	0.000	2.136
Acrylonitrile	1.669	0.122	0.000	0.000	2.046
Antimony (and its compounds)	0.000	0.000	0.000	0.000	1.083
Methylene*bis*(phenylisocyanate)	0.001	0.000	0.000	0.300	1.043
Toluenediisocyanate (mixed isomers)	0.230	0.000	0.000	0.000	0.631
Phosphoric acid	0.000	0.000	0.000	0.000	0.510
Methyl isobutyl ketone	0.000	0.000	0.000	0.000	0.385
1,2,4-Trimethylbenzene	0.260	0.000	0.000	0.000	0.260
Vinyl chloride	0.000	0.000	0.000	0.000	0.255
Ethylene	0.000	0.000	0.000	0.000	0.255
Propylene	0.000	0.000	0.000	0.000	0.255
Phenol	0.023	0.000	0.000	0.000	0.153
Lead (and its compounds)	0.000	0.000	0.000	0.000	0.142
Nitric acid	0.000	0.000	0.000	0.000	0.130
Copper (and its compounds)	0.000	0.000	0.000	0.000	0.130
Phthalic anhydride	0.000	0.000	0.000	0.000	0.130
Styrene oxide	0.000	0.000	0.000	0.000	0.130
Chromium (and its compounds)	0.000	0.000	0.000	0.000	0.130
Dibutyl phthalate	0.000	0.000	0.000	0.000	0.130
Sulfuric acid	0.000	0.000	0.000	0.000	0.130
Ethyl acrylate	0.000	0.000	0.000	0.000	0.130
Hydrogen cyanide	0.000	0.000	0.000	0.000	0.130
Nickel (and its compounds)	0.000	0.000	0.000	0.000	0.130
Propylene oxide	0.000	0.000	0.000	0.000	0.130
Hydrochloric acid	0.000	0.000	0.000	0.000	0.130
Ethylene glycol	0.000	0.000	0.000	0.000	0.125
Ammonium nitrate (solution)	0.000	0.000	0.000	0.000	0.125
Cyclohexane	0.000	0.000	0.000	0.000	0.125
2-Nitropropane	0.000	0.000	0.000	0.000	0.125
Epichlorohydrin	0.000	0.000	0.000	0.000	0.125
Di-*n*-octyl phthalate	0.000	0.000	0.000	0.000	0.104
Acrylic acid	0.000	0.000	0.000	0.000	0.010

Substance Name	Air	Water	Underground	Land	Total Releases*
Zinc (and its compounds)	0.000	0.000	0.000	0.000	0.007
Diethanolamine	0.000	0.000	0.000	0.000	0.002
Maleic anhydride	0.001	0.000	0.000	0.000	0.001
Toluene-2,4-diisocyanate	0.000	0.000	0.000	0.000	0.000
Butyl benzyl phthalate	0.000	0.000	0.000	0.000	0.000
Manganese (and its compounds)	0.000	0.000	0.000	0.000	0.000
Cadmium (and its compounds)	0.000	0.000	0.000	0.000	0.000
C.I. Acid Green 3	0.000	0.000	0.000	0.000	0.000
Total	**4,137.745**	**137.274**	**0.000**	**57.807**	**4,344.179**

17—Leather and allied products industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Toluene	22.865	0.000	0.000	0.000	22.865
2-Methoxyethanol	17.000	0.000	0.000	0.000	17.000
Acetone	11.470	0.000	0.000	0.000	11.470
Xylene (mixed isomers)	10.200	0.000	0.000	0.000	10.200
Ammonium sulphate (solution)	0.000	0.000	0.000	0.000	0.000
Chromium (and its compounds)	0.000	0.000	0.000	0.000	0.000
Sulfuric acid	0.000	0.000	0.000	0.000	0.000
Bis(2-ethylhexyl) phthalate	0.000	0.000	0.000	0.000	0.000
Methylene*bis* (phenylisocyanate)	0.000	0.000	0.000	0.000	0.000
Total	**61.535**	**0.000**	**0.000**	**0.000**	**61.535**

18—Primary textile industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Sulfuric acid	0.000	85.000	0.000	0.000	85.000
Acetaldehyde	43.551	0.000	0.000	0.000	43.551
Xylene (mixed isomers)	30.324	0.000	0.000	0.000	30.324
Biphenyl	8.137	0.060	0.000	0.059	8.256
1,4-Dioxane	1.161	4.643	0.000	0.000	5.884
Chlorine	0.300	0.700	0.000	0.000	1.000
Methanol	0.000	1.000	0.000	0.000	1.000
Ethylene glycol	0.000	0.000	0.000	0.000	0.840
Tetrachloroethylene	0.000	0.000	0.000	0.000	0.700
Antimony (and its compounds)	0.000	0.000	0.000	0.000	0.355
Phosphoric acid	0.002	0.000	0.000	0.000	0.002
Phenol	0.000	0.000	0.000	0.000	0.000
Total	**83.475**	**91.403**	**0.000**	**0.059**	**176.912**

19—Textile products industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Methyl ethyl ketone	186.098	0.000	0.000	0.000	186.098
Acetone	108.936	0.000	0.000	0.000	108.936
Sulfuric acid	0.000	0.000	0.000	0.000	0.493
Phenol	0.000	0.000	0.000	0.000	0.130
Formaldehyde	0.000	0.000	0.000	0.000	0.130
Lead (and its compounds)	0.009	0.000	0.000	0.000	0.009

Substance Name	Air	Water	Underground	Land	Total Releases*
Toluenediisocyanate (mixed isomers)	0.001	0.000	0.000	0.000	0.001
Hydrochloric acid	0.000	0.000	0.000	0.000	0.000
Ammonium sulphate (solution)	0.000	0.000	0.000	0.000	0.000
Total	**295.044**	**0.000**	**0.000**	**0.000**	**295.797**

25—Wood industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Formaldehyde	455.173	0.000	0.000	0.000	455.173
Methanol	149.151	0.000	0.000	0.000	149.151
Benzene	29.206	0.000	0.000	0.000	29.206
Phenol	16.354	0.000	0.000	0.000	16.354
Xylene (mixed isomers)	16.000	0.000	0.000	0.000	16.000
Toluene	14.398	0.000	0.000	0.000	14.398
Acetone	12.633	0.000	0.000	0.000	12.633
Isopropyl alcohol	5.000	0.000	0.000	0.000	5.000
Ammonia	3.221	0.000	0.000	0.000	3.221
Ethylene glycol	0.000	0.000	0.000	1.600	1.600
Methylene*bis*(phenylisocyanate)	0.173	0.000	0.000	0.000	0.303
Chromium (and its compounds)	0.000	0.020	0.000	0.000	0.170
Arsenic (and its compounds)	0.000	0.009	0.000	0.000	0.109
Copper (and its compounds)	0.000	0.007	0.000	0.000	0.078
Sulfuric acid	0.023	0.000	0.000	0.000	0.023
Anthracene	0.000	0.000	0.000	0.000	0.000
Phosphoric acid	0.000	0.000	0.000	0.000	0.000
Biphenyl	0.000	0.000	0.000	0.000	0.000
Naphthalene	0.000	0.000	0.000	0.000	0.000
Total	**701.332**	**0.036**	**0.000**	**1.600**	**703.419**

26—Furniture and fixture industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Toluene	292.430	0.000	0.000	0.000	292.430
Xylene (mixed isomers)	119.212	0.000	0.000	0.000	119.212
Methyl ethyl ketone	62.476	0.000	0.000	0.000	62.476
n-Butyl alcohol	18.681	0.000	0.000	0.000	18.681
Dichloromethane	18.207	0.000	0.000	0.000	18.207
1,2,4-Trimethylbenzene	13.143	0.000	0.000	0.000	13.143
Trichloroethylene	13.000	0.000	0.000	0.000	13.000
Methyl isobutyl ketone	11.160	0.000	0.000	0.000	11.160
Styrene	10.614	0.000	0.000	0.000	10.614
Toluenediisocyanate (mixed isomers)	0.003	0.000	0.000	0.000	0.003
Diethanolamine	0.000	0.000	0.000	0.000	0.000
Total	**558.926**	**0.000**	**0.000**	**0.000**	**558.926**

27—Paper and allied products industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Methanol	8,229.524	11,973.449	0.000	125.928	20,329.031
Sulfuric acid	352.996	2,719.703	0.000	0.000	3,072.959
Chlorine	1,930.280	0.000	0.000	0.000	1,931.812

Substance Name	Air	Water	Underground	Land	Total Releases*
Chlorine dioxide	1,721.135	5.273	0.000	0.000	1,726.408
Toluene	1,242.286	0.000	0.000	0.000	1,242.286
Xylene (mixed isomers)	1,132.400	0.000	0.000	0.000	1,132.400
Methyl ethyl ketone	937.295	0.010	0.000	0.000	937.305
Hydrochloric acid	507.649	9.128	0.000	0.000	517.337
Isopropyl alcohol	398.601	0.440	0.000	0.000	399.041
Ammonia	101.345	188.077	0.000	0.000	290.369
Formaldehyde	19.912	221.560	0.000	0.130	241.602
Acetone	212.710	0.315	0.000	0.000	213.155
Phenol	92.400	2.430	0.000	0.000	94.830
Ethylene glycol	76.165	17.390	0.000	0.346	93.901
Ammonium nitrate (solution)	0.000	26.700	0.000	0.000	26.700
Phosphoric acid	0.000	25.000	0.000	0.000	25.130
Methyl isobutyl ketone	22.800	0.000	0.000	0.000	22.800
n-Butyl alcohol	14.000	0.000	0.000	0.000	14.000
Acrylamide	0.000	0.316	0.000	2.845	3.161
Vinyl acetate	3.000	0.000	0.000	0.000	3.000
Asbestos	0.000	0.000	0.000	3.000	3.000
Diethanolamine	1.200	0.000	0.000	0.000	1.200
Zinc (and its compounds)	0.090	0.000	0.000	0.000	0.090
Copper (and its compounds)	0.039	0.000	0.000	0.000	0.039
Manganese (and its compounds)	0.000	0.000	0.000	0.000	0.000
Tetrachloroethylene	0.000	0.000	0.000	0.000	0.000
Nitric acid	0.000	0.000	0.000	0.000	0.000
Aluminum (fume or dust)	0.000	0.000	0.000	0.000	0.000
Total	**16,995.844**	**15,189.791**	**0.000**	**132.249**	**32,321.573**

28—Printing, publishing and allied industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Toluene	706.528	5.181	0.000	0.000	711.709
Isopropyl alcohol	330.919	0.544	0.000	0.000	331.463
Methanol	144.961	0.000	0.000	0.000	144.961
Methyl ethyl ketone	100.487	0.000	0.000	0.000	100.487
Acetone	36.768	0.000	0.000	0.000	36.768
Cyclohexane	21.365	0.120	0.000	0.000	21.485
Xylene (mixed isomers)	15.936	0.077	0.000	0.000	16.013
Dibutyl phthalate	11.427	0.000	0.000	0.000	11.427
Naphthalene	10.000	0.000	0.000	0.000	10.000
Tetrachloroethylene	8.808	0.000	0.000	0.000	8.808
n-Butyl alcohol	2.620	0.000	0.000	0.000	2.620
Sulfuric acid	0.438	0.000	0.000	0.000	0.438
Nitric acid	0.386	0.000	0.000	0.000	0.386
Ammonium nitrate (solution)	0.000	0.000	0.000	0.000	0.000
Ethylene glycol	0.000	0.000	0.000	0.000	0.000
Phosphoric acid	0.000	0.000	0.000	0.000	0.000
Total	**1,390.643**	**5.922**	**0.000**	**0.000**	**1,396.565**

29—Primary metal industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Zinc (and its compounds)	1,026.974	33.898	0.000	3,835.094	4,898.773
Manganese (and its compounds)	132.340	40.585	0.000	2,744.702	2,918.725
Sulfuric acid	2,804.985	7.500	0.000	4.000	2,817.179

Substance Name	Air	Water	Underground	Land	Total Releases*
Lead (and its compounds)	889.192	2.388	0.000	859.628	1,753.169
Copper (and its compounds)	478.457	11.529	0.000	961.388	1,453.894
Benzene	1,110.960	0.007	0.000	0.130	1,111.097
Styrene	1,101.453	0.004	0.000	0.000	1,102.870
Ammonia	451.258	500.167	0.000	0.000	951.425
Chromium (and its compounds)	7.382	28.319	0.000	739.668	777.575
Nickel (and its compounds)	541.693	65.379	0.000	46.839	655.459
Zinc (fume or dust)	38.264	57.000	0.000	554.000	649.644
Trichloroethylene	419.483	0.000	0.000	0.000	419.483
Toluene	135.092	0.005	0.000	0.000	135.097
Tetrachloroethylene	122.180	0.000	0.000	0.000	122.180
Isopropyl alcohol	103.300	0.000	0.000	0.000	104.118
Hydrochloric acid	89.224	0.000	0.000	11.400	100.624
Arsenic (and its compounds)	94.634	1.706	0.000	0.000	96.472
Cadmium (and its compounds)	66.105	1.717	0.000	14.000	82.795
Chlorine	78.274	1.095	0.000	0.000	80.399
Ethylene glycol	8.365	54.056	0.000	13.564	76.945
Xylene (mixed isomers)	60.560	0.006	0.000	0.000	60.566
Ethylene	37.180	0.000	0.000	0.000	37.180
Aluminum (fume or dust)	20.443	4.790	0.000	11.900	37.133
Methanol	24.560	11.000	0.000	0.000	35.560
Cobalt (and its compounds)	11.172	0.990	0.000	8.800	21.092
Acetone	20.800	0.000	0.000	0.000	20.800
Naphthalene	19.054	0.011	0.000	0.010	19.075
Propylene	15.300	0.000	0.000	0.000	15.300
Phenol	6.260	7.585	0.000	0.000	13.845
Hydrogen fluoride	12.900	0.290	0.000	0.000	13.190
Formaldehyde	5.075	0.000	0.000	0.000	5.075
Nitric acid	1.295	3.500	0.000	0.000	5.055
Selenium (and its compounds)	0.220	3.560	0.000	0.000	4.127
Vanadium (fume or dust)	0.061	0.030	0.000	3.600	3.702
Antimony (and its compounds)	1.150	0.000	0.000	2.400	3.668
Ethylbenzene	1.460	0.000	0.000	0.000	1.460
Cyanides (ionic)	0.000	1.162	0.000	0.000	1.162
Anthracene	0.871	0.037	0.000	0.003	0.911
Silver (and its compounds)	0.206	0.016	0.000	0.008	0.874
Molybdenum trioxide	0.000	0.000	0.000	0.000	0.323
Aluminum oxide (fibrous forms)	0.000	0.000	0.000	0.000	0.260
Phosphoric acid	0.000	0.000	0.000	0.000	0.130
Bis(2-ethylhexyl) adipate	0.000	0.000	0.000	0.000	0.010
Ammonium sulfate (solution)	0.000	0.000	0.000	0.004	0.004
Mercury (and its compounds)	0.000	0.000	0.000	0.000	0.002
Hydrazine	0.000	0.000	0.000	0.000	0.000
Decabromodiphenyl oxide	0.000	0.000	0.000	0.000	0.000
Methylene*bis*(phenylisocyanate)	0.000	0.000	0.000	0.000	0.000
Cumene hydroperoxide	0.000	0.000	0.000	0.000	0.000
Dichloromethane	0.000	0.000	0.000	0.000	0.000
Thiourea	0.000	0.000	0.000	0.000	0.000
Total	**9,938.182**	**838.332**	**0.000**	**9,811.138**	**20,608.427**

30—Fabricated metal products industries (except machinery and trans. equipment industries)

Substance Name	Air	Water	Underground	Land	Total Releases*
Xylene (mixed isomers)	594.745	0.000	0.000	0.000	594.954
n-Butyl alcohol	409.842	0.000	0.000	0.000	410.875
Methyl ethyl ketone	162.496	0.000	0.000	0.000	163.197

Substance Name	Air	Water	Underground	Land	Total Releases*
Trichloroethylene	148.430	0.000	0.000	0.000	148.430
Zinc (and its compounds)	92.328	0.263	0.000	20.995	116.578
Toluene	110.015	0.000	0.000	0.000	110.143
Methyl isobutyl ketone	93.574	0.000	0.000	0.000	94.347
Copper (and its compounds)	86.047	0.000	0.000	0.314	87.006
Hydrochloric acid	12.785	0.050	0.000	51.000	64.074
Dichloromethane	42.474	0.000	0.000	0.000	42.474
Acetone	33.984	0.000	0.000	0.000	33.984
Methanol	12.792	0.000	0.000	0.000	12.792
Sulfuric acid	6.163	0.400	0.000	0.150	8.198
Lead (and its compounds)	5.032	1.531	0.000	0.000	7.083
Isopropyl alcohol	6.370	0.000	0.000	0.000	6.370
Chromium (and its compounds)	2.935	0.021	0.000	0.176	3.865
Ethylene glycol	2.500	0.000	0.000	0.000	2.500
Zinc (fume or dust)	0.000	0.000	0.000	0.000	1.482
Manganese (and its compounds)	0.288	0.002	0.000	0.262	1.406
Phosphoric acid	0.000	0.000	0.000	0.000	1.260
Bis(2-ethylhexyl) phthalate	0.870	0.000	0.000	0.000	0.870
Nickel (and its compounds)	0.651	0.000	0.000	0.194	0.845
Aluminum (fume or dust)	0.130	0.000	0.000	0.630	0.760
Nitric acid	0.055	0.000	0.000	0.000	0.410
Ethylbenzene	0.000	0.000	0.000	0.000	0.369
Ammonia	0.000	0.000	0.000	0.000	0.131
Methylene*bis*(phenylisocyanate)	0.001	0.000	0.000	0.000	0.131
Formaldehyde	0.070	0.000	0.000	0.000	0.070
Methyl methacrylate	0.061	0.000	0.000	0.000	0.061
Hydrogen fluoride	0.010	0.000	0.000	0.000	0.010
Di-*n*-octyl phthalate	0.000	0.000	0.000	0.000	0.006
Aluminum oxide (fibrous forms)	0.000	0.000	0.000	0.000	0.004
Chlorine	0.000	0.000	0.000	0.000	0.000
Antimony (and its compounds)	0.000	0.000	0.000	0.000	0.000
Cadmium (and its compounds)	0.000	0.000	0.000	0.000	0.000
Total	**1,824.666**	**2.267**	**0.000**	**73.721**	**1,914.685**

31—Machinery industries (except electrical machinery)

Substance Name	Air	Water	Underground	Land	Total Releases*
Xylene (mixed isomers)	35.946	0.000	0.000	0.000	35.946
1,2,4-Trimethylbenzene	34.785	0.000	0.000	0.000	34.785
Aluminum oxide (fibrous forms)	0.000	0.000	0.000	31.600	31.600
Sulfuric acid	0.000	21.700	0.000	0.000	21.700
Vanadium (fume or dust)	3.600	0.000	0.000	1.600	5.200
Manganese (and its compounds)	0.134	0.000	0.000	0.630	1.050
Copper (and its compounds)	0.000	0.000	0.000	0.130	0.130
Chromium (and its compounds)	0.000	0.000	0.000	0.130	0.130
Dibutyl phthalate	0.000	0.000	0.000	0.000	0.130
Nickel (and its compounds)	0.000	0.000	0.000	0.130	0.130
Zinc (and its compounds)	0.000	0.000	0.000	0.000	0.000
Hydrochloric acid	0.000	0.000	0.000	0.000	0.000
Toluene	0.000	0.000	0.000	0.000	0.000
Methylene*bis*(phenylisocyanate)	0.000	0.000	0.000	0.000	0.000
Ethylene glycol	0.000	0.000	0.000	0.000	0.000
Total	**74.465**	**21.700**	**0.000**	**34.220**	**130.801**

32—Transportation equipment industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Xylene (mixed isomers)	3,187.581	0.000	0.000	0.000	3,187.581
Toluene	1,482.724	0.000	0.000	0.000	1,482.894
Acetone	629.204	0.000	0.000	0.000	629.204
Methyl ethyl ketone	566.682	0.000	0.000	0.000	566.682
n-Butyl alcohol	553.828	0.000	0.000	0.000	554.348
Methyl isobutyl ketone	571.276	0.000	0.000	0.000	517.276
Methanol	253.186	0.000	0.000	0.000	253.697
Isopropyl alcohol	245.095	0.000	0.000	0.000	245.095
Styrene	132.352	0.000	0.000	0.000	132.352
Asbestos	0.447	0.000	0.000	105.840	106.287
Ethylbenzene	106.270	0.000	0.000	0.001	106.271
i-Butyl alcohol	89.955	0.000	0.000	0.000	90.765
Trichloroethylene	89.334	0.000	0.000	0.000	89.334
1,2,4-Trimethylbenzene	41.413	0.000	0.000	0.000	41.413
Ammonia	39.488	0.000	0.000	0.000	39.488
Phenol	16.763	0.000	0.000	0.000	16.763
Aluminum (fume or dust)	6.615	0.000	0.000	5.752	12.367
Dichloromethane	11.627	0.000	0.000	0.000	11.627
Ethylene glycol	9.487	0.110	0.000	0.000	10.477
Zinc (and its compounds)	2.829	2.075	0.000	0.260	5.277
Copper (and its compounds)	3.530	0.006	0.000	0.000	3.698
Chromium (and its compounds)	0.049	0.000	0.000	0.000	2.878
Hydrochloric acid	1.288	0.000	0.000	0.000	1.662
Diethanolamine	1.107	0.000	0.000	0.000	1.607
Lead (and its compounds)	0.895	0.015	0.000	0.000	1.054
Manganese (and its compounds)	0.509	0.012	0.000	0.000	1.020
Phosphoric acid	0.555	0.000	0.000	0.000	0.555
o-Dichlorobenzene	0.380	0.000	0.000	0.000	0.380
Sulfuric acid	0.329	0.000	0.000	0.000	0.373
Nickel (and its compounds)	0.199	0.000	0.000	0.000	0.232
Cresol (mixed isomers)	0.130	0.000	0.000	0.000	0.130
Methylene*bis*(phenylisocyanate)	0.003	0.000	0.000	0.110	0.123
Nitric acid	0.018	0.000	0.000	0.000	0.018
Toluenediisocyanate (mixed isomers)	0.000	0.000	0.000	0.000	0.016
Zinc (fume or dust)	0.010	0.000	0.000	0.000	0.010
Butyl benzyl phthalate	0.002	0.000	0.000	0.000	0.002
Vinylidene chloride	0.000	0.000	0.000	0.000	0.000
Calcim cyanamide	0.000	0.000	0.000	0.000	0.000
Toluene-2,4-diisocyanate	0.000	0.000	0.000	0.000	0.000
Antimony (and its compounds)	0.000	0.000	0.000	0.000	0.000
Decabromodiphenyl oxide	0.000	0.000	0.000	0.000	0.000
Bis(2-ethylhexyl) phthalate	0.000	0.000	0.000	0.000	0.000
Total	**7,991.16**	**2.218**	**0.000**	**111.963**	**8,112.956**

33—Electrical and electronic products industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Xylene (mixed isomers)	237.340	0.000	0.000	0.000	237.940
Isopropyl alcohol	58.915	0.000	0.000	0.000	58.915
Toluene	50.297	0.000	0.000	0.000	50.297
Trichloroethylene	33.309	0.022	0.000	0.000	33.331
Tetrachloroethylene	22.070	0.000	0.000	0.000	22.070
1,2,4-Trimethylbenzene	18.430	0.000	0.000	0.000	18.430
Ethylene glycol	16.260	0.000	0.000	0.000	16.260
n-Butyl alcohol	13.904	0.000	0.000	0.000	13.904
Ammonia	0.327	10.120	0.000	0.000	10.447

Substance Name	Air	Water	Underground	Land	Total Releases*
Copper (and its compounds)	0.013	0.036	0.000	5.044	6.517
Lead (and its compounds)	2.597	1.079	0.000	0.669	5.366
Phenol	2.840	0.000	0.000	0.000	2.840
Cresol (mixed isomers)	1.960	0.000	0.000	0.000	1.960
Hydrochloric acid	1.100	0.000	0.000	0.000	1.100
Zinc (and its compounds)	0.171	0.089	0.000	0.535	0.795
Nitric acid	0.000	0.000	0.000	0.000	0.690
Cumene	0.000	0.000	0.000	0.000	0.630
Sulfuric acid	0.000	0.000	0.000	0.000	0.610
Antimony (and its compounds)	0.000	0.000	0.000	0.000	0.490
Manganese (and its compounds)	0.130	0.000	0.000	0.000	0.130
Formaldehyde	0.130	0.000	0.000	0.000	0.130
Toluenediisocyanate (mixed isomers)	0.000	0.000	0.000	0.000	0.112
Cobalt (and its compounds)	0.000	0.000	0.000	0.000	0.000
Nickel (and its compounds)	0.000	0.000	0.000	0.000	0.000
Zinc (fume or dust)	0.000	0.000	0.000	0.000	0.000
Aluminum (fume or dust)	0.000	0.000	0.000	0.000	0.000
Methylene*bis*(phenylisocyanate)	0.000	0.000	0.000	0.000	0.000
Chromium (and its compounds)	0.000	0.000	0.000	0.000	0.000
Hydrogen fluoride	0.000	0.000	0.000	0.000	0.000
Phosphoric acid	0.000	0.000	0.000	0.000	0.000
Total	**459.793**	**11.346**	**0.000**	**6.248**	**482.964**

35—Nonmetallic mineral products industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Toluene	661.297	0.000	0.000	0.000	661.427
Xylene (mixed isomers)	601.563	0.000	0.000	0.000	601.693
Ammonia	239.798	0.036	0.000	0.000	239.834
Isopropyl alcohol	175.479	45.415	0.000	0.000	220.894
Methanol	146.324	0.000	0.000	0.000	146.324
Zinc (and its compounds)	127.685	0.040	0.130	9.200	137.185
Formaldehyde	134.520	1.276	0.000	0.630	136.426
n-Butyl alcohol	73.070	0.000	0.000	0.000	73.070
Manganese (and its compounds)	70.000	0.000	0.000	0.630	71.760
Dichloromethane	66.000	0.000	0.000	0.000	66.130
Methyl ethyl ketone	50.335	0.000	0.000	0.000	50.335
Lead (and its compounds)	30.000	0.000	0.000	5.700	35.708
Phenol	21.484	0.000	0.000	0.000	21.484
1,2,4-Trimethylbenzene	17.970	0.000	0.000	0.000	17.970
Acetone	5.134	0.000	0.000	0.000	5.134
Cyclohexane	4.107	0.000	0.000	0.000	4.107
Chromium (and its compounds)	1.723	0.010	0.000	2.300	4.037
Ammonium sulphate (solution)	3.898	0.000	0.000	0.000	3.898
Ethylene glycol	0.000	0.000	0.000	0.000	0.140
Tetrachloroethylene	0.000	0.000	0.000	0.000	0.130
Chloroform	0.000	0.000	0.000	0.000	0.130
1,2-Dichloroethane	0.000	0.000	0.000	0.000	0.130
Trichloroethylene	0.000	0.000	0.000	0.000	0.130
Phosphoric acid	0.000	0.000	0.000	0.000	0.021
Sulfuric acid	0.000	0.000	0.000	0.000	0.000
Cobalt (and its compounds)	0.000	0.000	0.000	0.000	0.000
Hydrochloric acid	0.000	0.000	0.000	0.000	0.000
Total	**2,430.387**	**46.777**	**0.130**	**18.460**	**2,498.097**

36—Refined petroleum and coal products industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Ammonia	335.477	138.887	5,021.400	0.020	5,495.784
Toluene	947.616	0.793	19.160	1.525	969.333
Methanol	820.260	0.000	0.000	0.000	820.468
Xylene (mixed isomers)	753.650	0.976	13.590	1.100	769.866
Benzene	498.697	0.582	20.390	0.745	520.634
Propylene	516.889	0.000	0.000	0.000	516.889
Ethylene	422.360	0.000	0.000	0.000	423.370
Methyl ethyl ketone	247.280	0.000	0.000	0.000	247.280
Cyclohexane	205.556	0.067	0.050	1.367	207.670
1,2,4-Trimethylbenzene	159.210	0.087	0.610	0.017	160.339
Ethylbenzene	154.430	0.213	1.870	0.442	157.259
Methyl isobutyl ketone	116.830	0.000	0.000	0.000	116.830
Sulfuric acid	81.469	0.000	0.000	0.000	81.469
Diethanolamine	44.435	21.400	0.000	0.000	66.685
Isopropyl alcohol	41.060	0.000	0.000	0.000	41.710
Naphthalene	30.663	0.395	0.045	0.084	31.187
Cumene	24.627	0.020	0.029	0.000	25.645
Acetone	24.710	0.081	0.000	0.000	24.791
Hydrochloric acid	0.000	0.000	22.680	0.000	22.680
1,3-Butadiene	22.200	0.000	0.000	0.000	22.289
Manganese (and its compounds)	0.198	0.000	0.000	22.000	22.198
Methyl *tert*-butyl ether	19.231	0.400	0.000	0.000	19.631
Ethylene glycol	18.175	1.200	0.000	0.000	19.375
Phenol	9.223	5.221	1.980	0.000	16.554
m-Xylene	13.150	0.000	0.000	0.000	13.150
p-Xylene	8.080	0.000	0.000	0.000	8.080
o-Xylene	7.791	0.000	0.000	0.000	7.791
Zinc (and its compounds)	0.059	0.607	0.077	5.910	6.653
Styrene	6.190	0.000	0.000	0.000	6.190
Biphenyl	1.729	0.000	0.000	0.000	1.729
Anthracene	1.030	0.000	0.000	0.000	1.476
Chlorine	1.260	0.063	0.000	0.000	1.323
Nickel (and its compounds)	0.123	0.000	0.000	0.850	1.003
Copper (and its compounds)	0.100	0.000	0.000	0.730	0.830
2-Ethoxyethanol	0.490	0.000	0.000	0.000	0.490
Chromium (and its compounds)	0.000	0.000	0.000	0.450	0.450
Molybdenum trioxide	0.319	0.000	0.000	0.000	0.441
Nitric acid	0.300	0.000	0.000	0.000	0.396
Arsenic (and its compounds)	0.000	0.000	0.000	0.290	0.290
Lead (and its compounds)	0.000	0.000	0.000	0.140	0.259
Phosphoric acid	0.150	0.000	0.000	0.000	0.226
Tetrachloroethylene	0.130	0.064	0.000	0.000	0.194
Dichloromethane	0.000	0.000	0.000	0.000	0.100
Hydrogen fluoride	0.000	0.000	0.000	0.000	0.039
o-Dichlorobenzene	0.020	0.000	0.000	0.000	0.020
Mercury (and its compounds)	0.000	0.000	0.000	0.011	0.011
Vanadium (fume or dust)	0.010	0.000	0.000	0.000	0.010
Cresol (mixed isomers)	0.000	0.000	0.000	0.000	0.004
Carbon tetrachloride	0.000	0.000	0.000	0.000	0.000
Asbestos	0.000	0.000	0.000	0.000	0.000
Cobalt (and tis compounds)	0.000	0.000	0.000	0.000	0.000
Butyl benzyl phthalate	0.000	0.000	0.000	0.000	0.000
Vinyl chloride	0.000	0.000	0.000	0.000	0.000
Trichloroethylene	0.000	0.000	0.000	0.000	0.000
Total	**5,535.177**	**171.056**	**5,101.881**	**35.681**	**10,851.091**

37—Chemical and chemical products industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Ammonia	16,062.078	335.649	998.000	3.475	17,403.510
Sulfuric acid	108.269	15,002.700	0.000	2.196	15,116.683
Methanol	7,099.540	43.716	400.000	0.025	7,548.896
Cyclohexane	2,443.454	0.500	0.000	0.100	2,444.054
Acetone	1,984.247	38.550	84.000	49.400	2,159.373
Ethylene	2,046.103	0.000	0.000	0.000	2,046.946
Ammonium nitrate (solution)	334.100	663.100	828.000	47.023	1,872.225
Chloromethane	1,176.020	0.880	0.000	0.000	1,176.900
Benzene	776.256	0.455	7.600	1.577	786.018
Dichloromethane	668.759	0.000	0.000	0.039	670.490
Hydrochloric acid	566.804	54.000	0.000	0.662	623.788
Toluene	579.833	0.348	0.310	3.691	590.656
Propylene	576.964	0.000	0.000	0.000	576.969
Ammonium sulphate (solution)	43.900	454.300	0.000	18.260	516.460
Methyl ethyl ketone	155.973	0.000	280.000	0.129	440.391
Xylene (mixed isomers)	355.526	0.131	0.000	0.423	365.759
Isopropyl alcohol	322.428	0.130	0.000	0.000	328.446
Styrene	309.701	0.020	0.185	0.196	314.172
1,3-Butadiene	270.302	0.058	0.000	0.002	270.362
Ethylbenzene	196.770	0.101	0.000	0.175	197.046
Formaldehyde	102.345	2.504	69.920	0.000	178.702
Chloroethane	174.100	0.370	0.000	0.000	174.470
Ethylene glycol	145.933	9.487	0.000	0.084	159.636
Vinyl acetate	111.339	0.000	7.700	0.695	121.578
Asbestos	0.130	0.000	0.000	98.380	98.510
tert-Butyl alcohol	48.410	1.100	42.000	0.002	91.512
Acetaldehyde	70.506	13.200	6.000	0.030	89.736
Methyl isobutyl ketone	79.696	0.000	0.000	0.155	80.631
Acetonitrile	79.000	0.130	0.000	0.000	79.130
1,2,4-Trimethylbenzene	60.712	0.146	0.000	0.000	61.034
Manganese (and its compounds)	0.133	58.000	0.000	0.000	58.893
n-Butyl alcohol	48.593	0.000	0.000	0.100	52.459
Ethylene oxide	50.842	0.000	0.000	0.000	51.067
Nitric acid	0.179	32.700	0.000	1.600	35.038
Methyl methacrylate	27.289	0.000	0.000	0.000	27.829
Maleic anhydride	26.662	0.000	0.000	0.000	27.478
Carbon disulphide	13.435	10.405	0.000	0.000	25.720
Vinyl chloride	23.146	0.164	0.000	0.000	23.340
Phenol	19.595	0.599	0.000	0.000	22.452
Naphthalene	20.442	0.000	0.000	0.237	21.580
Diethanolamine	17.378	0.000	0.100	0.000	17.821
Acrylonitrile	16.600	0.040	0.000	0.000	17.510
Carbon tetrachloride	12.773	4.732	0.000	0.000	17.505
Chlorine	8.430	3.160	0.000	0.000	15.832
Phosphorus (yellow or white)	0.025	4.586	0.000	10.000	14.611
Phthalic anhydride	11.115	0.000	0.000	0.000	13.120
Hydrogen fluoride	12.200	0.000	0.000	0.000	12.256
Propylene oxide	10.941	0.000	0.000	0.000	10.941
p-Dichlorobenzene	9.900	0.000	0.000	0.500	10.400
Nickel (and its compounds)	5.194	3.610	0.000	0.000	10.311
Phosphoric acid	2.841	2.212	0.000	3.000	10.037
Nitroglycerin	0.000	9.000	0.000	0.000	9.000
Methyl *tert*-butyl ether	8.396	0.000	0.000	0.000	8.396
p-Xylene	8.194	0.000	0.000	0.000	8.194
Cobalt (and its compounds)	6.840	1.120	0.000	0.000	8.090
Lead (and its compounds)	5.370	0.360	0.000	0.000	7.545
1,2-Dichloroethane	6.886	0.523	0.000	0.040	7.449
Tetrachloroethylene	6.440	0.010	0.000	0.000	7.448
Zinc (and its compounds)	1.300	2.369	0.000	0.000	6.504
Aluminum (fume or dust)	0.001	0.000	0.380	4.810	6.156
i-Butyl alcohol	4.457	0.000	0.000	0.000	6.137

Substance Name	Air	Water	Underground	Land	Total Releases*
Chromium (and its compounds)	1.700	0.708	0.000	0.000	5.694
m-Xylene	5.604	0.000	0.000	0.000	5.614
Biphenyl	5.000	0.000	0.000	0.320	5.450
Copper (and its compounds)	1.340	2.509	0.000	0.530	4.515
2,4-Dinitrotoluene	0.000	4.000	0.000	0.000	4.000
o-Xylene	3.765	0.000	0.000	0.000	3.765
Vanadium (fume or dust)	0.000	2.600	0.000	0.000	2.600
Methyl acrylate	2.112	0.000	0.000	0.000	2.112
Acrylamide	0.897	0.000	0.000	0.000	1.977
Chloride dioxide	1.960	0.000	0.000	0.000	1.960
2-Ethoxyethanol	1.415	0.000	0.000	0.000	1.864
Molybdenum trioxide	1.400	0.000	0.000	0.000	1.530
Zinc (fume or dust)	1.405	0.000	0.000	0.000	1.405
Butyl acrylate	0.435	0.000	0.000	0.000	1.141
Nitrilotriacetic acid	0.000	0.000	0.000	0.000	1.001
Ethyl acrylate	0.130	0.000	0.000	0.000	0.990
Aluminum oxide (fibrous forms)	0.000	0.000	0.000	0.000	0.890
Acrylic acid	0.139	0.000	0.000	0.000	0.879
1,4-Dioxane	0.860	0.000	0.000	0.000	0.860
Antimony (and its compounds)	0.009	0.003	0.000	0.000	0.772
Trichloroethylene	0.000	0.000	0.000	0.000	0.702
Dibutyl phthalate	0.309	0.000	0.000	0.000	0.569
Bis(2-ethylhexyl) phthalate	0.244	0.000	0.000	0.161	0.415
Di-*n*-octyl phthalate	0.180	0.000	0.000	0.000	0.310
Cumene	0.300	0.000	0.000	0.000	0.300
p,*p*′-Isopropylidenediphenol	0.000	0.000	0.000	0.000	0.169
Toluene-2,4-diisocyanate	0.000	0.000	0.000	0.000	0.148
Toluenediisocyanate (mixed isomers)	0.011	0.000	0.000	0.000	0.147
Benzyl chloride	0.012	0.000	0.000	0.000	0.142
Anthracene	0.000	0.000	0.000	0.000	0.130
Cresol (mixed isomers)	0.000	0.000	0.000	0.000	0.130
Butyl benzyl phthalate	0.130	0.000	0.000	0.000	0.130
o-Dichlorobenzene	0.000	0.000	0.000	0.000	0.080
Cyanides (ionic)	0.000	0.000	0.000	0.000	0.070
Mercury (and its compounds)	0.057	0.008	0.000	0.000	0.065
Chloroform	0.039	0.002	0.000	0.001	0.042
Methylene*bis*(phenylisocyanate)	0.026	0.000	0.000	0.000	0.028
Titanium tetrachloride	0.023	0.000	0.000	0.000	0.023
Hydroquinone	0.000	0.000	0.000	0.000	0.021
Cumene hydroperoxide	0.000	0.000	0.000	0.000	0.020
Dimethyl sulphate	0.011	0.000	0.000	0.000	0.011
Epichlorohydrin	0.000	0.000	0.000	0.000	0.008
Bis(2-ethylhexyl) adipate	0.000	0.000	0.000	0.000	0.006
p,*p*′-Methylene*bis*(2-chloroaniline)	0.000	0.000	0.000	0.000	0.005
Aniline	0.005	0.000	0.000	0.000	0.005
2-Ethoxyethyl acetate	0.003	0.000	0.000	0.000	0.003
o-Cresol	0.000	0.000	0.000	0.000	0.001
2-Methoxyethanol	0.000	0.000	0.000	0.000	0.000
Cadmium (and its compounds)	0.000	0.000	0.000	0.000	0.000
4,6-Dinitro-*o*-cresol	0.000	0.000	0.000	0.000	0.000
Chloroacetic acid	0.000	0.000	0.000	0.000	0.000
sec-Butyl alcohol	0.000	0.000	0.000	0.000	0.000
1,1,2-Trichloroethane	0.000	0.000	0.000	0.000	0.000
Peracetic acid	0.000	0.000	0.000	0.000	0.000
Hydrazine	0.000	0.000	0.000	0.000	0.000
Benzoyl peroxide	0.000	0.000	0.000	0.000	0.000
Decabromodiphenyl oxide	0.000	0.000	0.000	0.000	0.000
o-Phenylphenol	0.000	0.000	0.000	0.000	0.000
Total	**37,364.316**	**16,764.995**	**2,724.195**	**248.018**	**57,207.901**

39—Other manufacturing industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Methyl ethyl ketone	321.720	0.000	0.000	0.000	321.720
Methanol	95.900	0.000	0.000	0.000	96.195
Trichloroethylene	22.748	0.000	0.000	0.000	22.748
Styrene	12.030	0.000	0.000	0.000	12.030
Xylene (mixed isomers)	10.858	0.000	0.000	0.000	10.858
Methyl isobutyl ketone	10.042	0.000	0.000	0.000	10.042
Phosphoric acid	7.420	0.000	0.000	0.130	7.550
Chromium (and its compounds)	0.000	0.000	0.000	5.930	5.930
Butyl benzyl phthalate	3.550	0.000	0.000	0.000	3.550
Acetone	2.850	0.000	0.000	0.000	2.852
Bis(2-ethylhexyl) phthalate	2.330	0.000	0.000	0.000	2.330
Lead (and its compounds)	1.603	0.000	0.000	0.000	1.993
Nitric acid	1.800	0.000	0.000	0.000	1.800
1,2,4-Trimethylbenzene	0.880	0.000	0.000	0.000	0.880
Zinc (and its compounds)	0.000	0.000	0.000	0.000	0.270
Silver (and its compounds)	0.000	0.000	0.000	0.000	0.130
Methylene*bis*(phenylisocyanate)	0.000	0.000	0.000	0.000	0.130
Ethylene glycol	0.000	0.000	0.000	0.000	0.130
Antimony (and its compounds)	0.000	0.000	0.000	0.000	0.130
Copper (and its compounds)	0.000	0.000	0.000	0.000	0.130
Vinyl chloride	0.000	0.000	0.000	0.000	0.130
Toluenediisocyanate (mixed isomers)	0.000	0.000	0.000	0.000	0.123
Nickel (and its compounds)	0.000	0.000	0.000	0.000	0.001
Isopropyl alcohol	0.000	0.000	0.000	0.000	0.001
Aluminum oxide (fibrous forms)	0.000	0.000	0.000	0.001	0.001
Naphthalene	0.000	0.000	0.000	0.000	0.000
1,1,2-Trichloroethane	0.000	0.000	0.000	0.000	0.000
Benzene	0.000	0.000	0.000	0.000	0.000
Ethylene	0.000	0.000	0.000	0.000	0.000
Aluminum (fume or dust)	0.000	0.000	0.000	0.000	0.000
p-Dichlorobenzene	0.000	0.000	0.000	0.000	0.000
Toluene	0.000	0.000	0.000	0.000	0.000
Methyl *tert*-butyl ether	0.000	0.000	0.000	0.000	0.000
Ammonia	0.000	0.000	0.000	0.000	0.000
Sulfuric acid	0.000	0.000	0.000	0.000	0.000
Chlorine	0.000	0.000	0.000	0.000	0.000
Dichloromethane	0.000	0.000	0.000	0.000	0.000
Ethylene oxide	0.000	0.000	0.000	0.000	0.000
Hydrochloric acid	0.000	0.000	0.000	0.000	0.000
Total	**493.731**	**0.000**	**0.000**	**6.061**	**501.654**

41—Industrial and heavy (engineering) construction industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Ammonia	111.798	13.170	0.000	585.200	710.168
Sulfuric acid	0.000	114.382	0.000	0.000	114.382
Asbestos	0.000	0.000	0.000	82.000	82.000
Chlorine	1.558	0.390	0.000	0.000	1.948
Acrylamide	0.000	0.610	0.000	0.000	0.610
Hydrazine	0.000	0.000	0.000	0.000	0.345
Xylene (mixed isomers)	0.000	0.000	0.000	0.000	0.300
Lead (and its compounds)	0.000	0.000	0.000	0.000	0.000
Nickel (and its compounds)	0.000	0.000	0.000	0.000	0.000
Zinc (and its compounds)	0.000	0.000	0.000	0.000	0.000
Copper (and its compounds)	0.000	0.000	0.000	0.000	0.000
Chromium (and its compounds)	0.000	0.000	0.000	0.000	0.000
Antimony (and its compounds)	0.000	0.000	0.000	0.000	0.000

Substance Name	Air	Water	Underground	Land	Total Releases*
Bis(2-ethylhexyl) phthalate	0.000	0.000	0.000	0.000	0.000
Aluminum (fume or dust)	0.000	0.000	0.000	0.000	0.000
Ethylene glycol	0.000	0.000	0.000	0.000	0.000
Vinyl acetate	0.000	0.000	0.000	0.000	0.000
Chlorine dioxide	0.000	0.000	0.000	0.000	0.000
Nitric acid	0.000	0.000	0.000	0.000	0.000
Vanadium (fume or dust)	0.000	0.000	0.000	0.000	0.000
Hydrochloric acid	0.000	0.000	0.000	0.000	0.000
Total	**113.356**	**128.552**	**0.000**	**667.200**	**909.753**

42—Trade contracting industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Xylene (mixed isomers)	72.092	0.000	0.000	0.000	72.092
Methylene*bis*(phenylisocyanate)	0.000	0.000	0.000	0.000	0.130
Sulfuric acid	0.000	0.000	0.000	0.000	0.000
Total	**72.092**	**0.000**	**0.000**	**0.000**	**72.222**

44—Service industries incidental to construction

Substance Name	Air	Water	Underground	Land	Total Releases*
Ethylene glycol	0.000	0.000	0.000	0.000	0.000
Methanol	0.000	0.000	0.000	0.000	0.000
Total	**0.000**	**0.000**	**0.000**	**0.000**	**0.000**

45—Transportation industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Ethylene glycol	2.180	0.000	0.000	2,224.068	2,226.378
Styrene	0.000	10.000	0.000	0.000	10.000
Propylene	0.000	0.000	0.000	0.000	0.000
Total	**2.180**	**10.000**	**0.000**	**2,224.068**	**2,236.378**

46—Pipeline transport industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Xylene (mixed isomers)	75.200	0.000	0.000	0.000	75.200
Phosphoric acid	0.000	0.000	0.000	0.000	0.000
Methylene*bis*(phenylisocyanate)	0.000	0.000	0.000	0.000	0.000
Total	**75.200**	**0.000**	**0.000**	**0.000**	**75.200**

47—Storage and warehousing industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Methanol	1.600	0.000	0.000	0.000	1.610
Vinyl acetate	1.500	0.000	0.000	0.000	1.500
n-Butyl alcohol	0.000	0.000	0.000	0.000	0.000
Sulfuric acid	0.000	0.000	0.000	0.000	0.000
Methyl ethyl ketone	0.000	0.000	0.000	0.000	0.000
Isopropyl alcohol	0.000	0.000	0.000	0.000	0.000
Toluene	0.000	0.000	0.000	0.000	0.000
Xylene (mixed isomers)	0.000	0.000	0.000	0.000	0.000
Methyl isobutyl ketone	0.000	0.000	0.000	0.000	0.000
Total	**3.100**	**0.000**	**0.000**	**0.000**	**3.110**

48—Communication industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Isopropyl alcohol	10,032.000	0.000	0.000	0.000	10,032.000
Total	**10,032.000**	**0.000**	**0.000**	**0.000**	**10,032.000**

49—Other utility industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Ammonia	0.000	183.013	0.000	0.000	183.013
Sulfuric acid	0.000	129.641	0.000	0.000	129.641
Chlorine	0.000	35.300	0.000	0.000	35.300
Hydrazine	0.000	0.000	0.000	0.000	0.897
Manganese (and its compounds)	0.000	0.000	0.000	0.000	0.000
Hydrochloric acid	0.000	0.000	0.000	0.000	0.000
Ethylene glycol	0.000	0.000	0.000	0.000	0.000
Chlorine dioxide	0.000	0.000	0.000	0.000	0.000
Vanadium (fume or dust)	0.000	0.000	0.000	0.000	0.000
Asbestos	0.000	0.000	0.000	0.000	0.000
Total	**0.000**	**347.954**	**0.000**	**0.000**	**348.851**

52—Food, beverage, drug, and tobacco industries, wholesale

Substance Name	Air	Water	Underground	Land	Total Releases*
Isopropyl alcohol	0.000	0.000	0.000	0.000	0.000
Total	**0.000**	**0.000**	**0.000**	**0.000**	**0.000**

55—Motor vehicle, parts, and accessories industries, wholesale

Substance Name	Air	Water	Underground	Land	Total Releases*
Toluene	92.200	0.000	0.000	0.000	92.200
Methyl ethyl ketone	41.620	0.000	0.000	0.000	41.620
Isopropyl alcohol	40.830	0.000	0.000	0.000	40.830
Ethylbenzene	32.200	0.000	0.000	0.000	32.200
Methyl isobutyl ketone	23.400	0.000	0.000	0.000	23.400
Xylene (mixed isomers)	15.800	0.000	0.000	0.000	15.800

Substance Name	Air	Water	Underground	Land	Total Releases*
Acetone	7.000	0.000	0.000	0.000	7.000
Aluminum (fume or dust)	0.000	0.000	0.000	3.030	3.030
1,4-Dioxane	2.400	0.000	0.000	0.000	2.400
Formaldehyde	0.000	0.000	0.000	0.000	0.640
Carbon tetrachloride	0.000	0.000	0.000	0.000	0.630
Methanol	0.000	0.000	0.000	0.000	0.350
i-Butyl alcohol	0.000	0.000	0.000	0.000	0.110
Zinc (and its compounds)	0.000	0.000	0.000	0.000	0.000
Phenol	0.000	0.000	0.000	0.000	0.000
Total	**255.450**	**0.000**	**0.000**	**3.030**	**260.210**

56—Metals, hardware, plumbing, heating, and building materials industries, wholesale

Substance Name	Air	Water	Underground	Land	Total Releases*
Sulfuric acid	4.730	0.000	0.000	0.000	4.730
Lead (and its compounds)	0.000	0.140	0.000	0.000	0.140
Zinc (and its compounds)	0.000	0.031	0.000	0.000	0.031
Copper (and its compounds)	0.000	0.000	0.000	0.000	0.000
Hydrochloric acid	0.000	0.000	0.000	0.000	0.000
Arsenic (and its compounds)	0.000	0.000	0.000	0.000	0.000
Chromium (and its compounds)	0.000	0.000	0.000	0.000	0.000
Total	**4.730**	**0.171**	**0.000**	**0.000**	**4.901**

59—Other products and industries, wholesale

Substance Name	Air	Water	Underground	Land	Total Releases*
Zinc (and its compounds)	0.000	0.000	0.000	181.790	181.790
Methanol	104.890	0.000	0.000	0.000	107.811
Dichloromethane	55.480	0.000	0.000	0.000	55.610
Acetone	18.064	0.000	0.000	0.000	20.281
Formaldehyde	12.456	0.000	0.000	0.000	12.716
Isopropyl alcohol	6.624	0.000	0.000	0.130	9.397
Methyl ethyl ketone	6.541	0.000	0.000	0.000	8.657
Toluene	4.115	0.000	0.000	0.000	6.520
Xylene (mixed isomers)	2.294	0.000	0.000	0.000	3.701
Tetrachloroethylene	1.014	0.000	0.000	0.000	1.175
Sulfuric acid	0.130	0.000	0.000	0.000	1.130
Methyl isobutyl ketone	0.456	0.000	0.000	0.000	0.951
Aluminum (fume or dust)	0.000	0.000	0.000	0.000	0.880
Ethylene glycol	0.141	0.000	0.000	0.000	0.795
Phosphoric acid	0.014	0.000	0.000	0.000	0.559
n-Butyl alcohol	0.133	0.000	0.000	0.000	0.432
Hydrochloric acid	0.000	0.000	0.000	0.000	0.407
Diethanolamine	0.000	0.000	0.000	0.000	0.390
Trichloroethylene	0.001	0.000	0.000	0.000	0.337
Nitric acid	0.168	0.000	0.000	0.000	0.307
Chlorine	0.000	0.000	0.000	0.000	0.260
Chloromethane	0.000	0.000	0.000	0.000	0.183
Cyclohexane	0.026	0.000	0.000	0.000	0.161
Ammonia	0.030	0.000	0.000	0.000	0.160
Maleic anhydride	0.130	0.000	0.000	0.000	0.130
Methyl *tert*-butyl ether	0.000	0.000	0.000	0.000	0.130
Di-*n*-octyl phthalate	0.000	0.000	0.000	0.000	0.060
Styrene	0.000	0.000	0.000	0.000	0.048
i-Butyl alcohol	0.001	0.000	0.000	0.000	0.022
Methyl methacrylate	0.000	0.000	0.000	0.000	0.007

Substance Name	Air	Water	Underground	Land	Total Releases*
Dibutyl phthalate	0.000	0.000	0.000	0.000	0.002
Butyl benzyl phthalate	0.000	0.000	0.000	0.000	0.000
Diethyl phthalate	0.000	0.000	0.000	0.000	0.000
Ethylbenzene	0.000	0.000	0.000	0.000	0.000
Lead (and its compounds)	0.000	0.000	0.000	0.000	0.000
Copper (and its compounds)	0.000	0.000	0.000	0.000	0.000
sec-Butyl alcohol	0.000	0.000	0.000	0.000	0.000
Bis(2-ethylhexyl) phthalate	0.000	0.000	0.000	0.000	0.000
2-Ethoxyethanol	0.000	0.000	0.000	0.000	0.000
2-Ethoxyethyl acetate	0.000	0.000	0.000	0.000	0.000
Bis(2-ethylhexyl) adipate	0.000	0.000	0.000	0.000	0.000
2-Methoxyethanol	0.000	0.000	0.000	0.000	0.000
Total	**212.708**	**0.000**	**0.000**	**181.920**	**415.009**

77—Business service industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Chlorine	0.000	0.000	0.000	0.000	0.000
Total	**0.000**	**0.000**	**0.000**	**0.000**	**0.000**

81—Federal government service industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Ethylene glycol	0.000	0.000	0.000	181.970	181.970
Isopropyl alcohol	0.000	0.000	0.000	0.000	0.032
Acetone	0.000	0.000	0.000	0.000	0.032
Methyl ethyl ketone	0.000	0.000	0.000	0.000	0.032
Benzene	0.000	0.000	0.000	0.000	0.019
Toluene	0.000	0.000	0.000	0.000	0.018
Xylene (mixed isomers)	0.000	0.000	0.000	0.000	0.012
Ethylbenzene	0.000	0.000	0.000	0.000	0.005
Styrene	0.000	0.000	0.000	0.000	0.005
Total	**0.000**	**0.000**	**0.000**	**181.970**	**182.125**

97—Personal and household service industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Sulfuric acid	0.000	0.000	0.000	0.000	0.000
Total	**0.000**	**0.000**	**0.000**	**0.000**	**0.000**

99—Other service industries

Substance Name	Air	Water	Underground	Land	Total Releases*
Styrene	2.289	0.000	0.000	0.000	2.289
Sulfuric acid	0.000	0.000	0.000	0.000	0.000
Total	**2.289**	**0.000**	**0.000**	**0.000**	**2.289**

* Total releases may be greater than the sum of the releases by environmental medium since releases of less than one tonne could be reported as an undifferentiated total release.

Appendix 6: Sectorial releases of toxic,* carcinogenic,† or potentially carcinogenic substances in alphabetical order (tonnes)

Acetaldehyde

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
37	Chemical and chemical products industries	70.506	13.200	6.000	0.030	89.736
18	Primary textile industries	43.551	0.000	0.000	0.000	43.551
	Total	**114.057**	**13.200**	**6.000**	**0.030**	**133.287**

NOTE: See p. P-40 for footnotes.

Acrylamide

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
27	Paper and allied products industries	0.000	0.316	0.000	2.845	3.161
37	Chemical and chemical products industries	0.897	0.000	0.000	0.000	1.977
41	Industrial and heavy (engineering) construction industries	0.000	0.610	0.000	0.000	0.610
06	Mining industries	0.000	0.000	0.000	0.000	0.130
	Total	**0.897**	**0.926**	**0.000**	**2.845**	**5.878**

Acrylonitrile

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
37	Chemical and chemical products industries	16.600	0.040	0.000	0.000	17.510
16	Plastic products industries	1.669	0.122	0.000	0.000	2.046
15	Rubber products industries	0.000	0.000	0.000	0.000	0.000
	Total	**18.269**	**0.162**	**0.000**	**0.000**	**19.556**

Arsenic (and its compounds)

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
06	Mining industries	38.296	45.489	3,800.000	0.000	3,883.785
29	Primary metal industries	94.634	1.706	0.000	0.000	96.472
36	Refined petroleum and coal products industries	0.000	0.000	0.000	0.290	0.290
25	Wood industries	0.000	0.009	0.000	0.000	0.109
56	Metals, hardware, plumbing, heating, and building materials industries, wholesale	0.000	0.000	0.000	0.000	0.000
	Total	**132.930**	**47.204**	**3,800.000**	**0.290**	**3,980.656**

Asbestos

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
32	Transportation equipment industries	0.447	0.000	0.000	105.840	106.287
37	Chemical and chemical products industries	0.130	0.000	0.000	98.380	98.510
41	Industrial and heavy (engineering) construction industries	0.000	0.000	0.000	82.000	82.000

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
07	Crude petroleum and natural gas industries	0.000	0.000	0.000	51.000	51.000
15	Rubber products industries	0.000	0.000	0.000	10.800	10.800
27	Paper and allied products industries	0.000	0.000	0.000	3.000	3.000
06	Mining industries	0.000	0.000	0.000	0.000	0.587
49	Other utility industries	0.000	0.000	0.000	0.000	0.000
36	Refined petroleum and coal products industries	0.000	0.000	0.000	0.000	0.000
	Total	**0.557**	**0.000**	**0.000**	**351.020**	**352.184**

Benzene

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
29	Primary metal industries	1,110.960	0.007	0.000	0.130	1,111.097
37	Chemical and chemical products industries	776.256	0.455	7.600	1.577	786.018
36	Refined petroleum and coal products industries	498.697	0.582	20.390	0.745	520.634
07	Crude petroleum and natural gas industries	175.608	0.008	45.900	0.459	228.494
25	Wood industries	29.206	0.000	0.000	0.000	29.206
81	Federal government service industries	0.000	0.000	0.000	0.000	0.019
39	Other manufacturing industries	0.000	0.000	0.000	0.000	0.000
	Total	**2,590.727**	**1.052**	**73.890**	**2.911**	**2,675.468**

***Bis*(2-ethylhexyl) phthalate**

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
16	Plastic products industries	8.730	0.000	0.000	55.874	64.608
15	Rubber products industries	11.957	0.000	0.000	12.624	24.581
39	Other manufacturing industries	2.330	0.000	0.000	0.000	2.330
30	Fabricated metal products industries (except machinery and trans. equipment industries)	0.870	0.000	0.000	0.000	0.870
37	Chemical and chemical products industries	0.244	0.000	0.000	0.161	0.415
32	Transportation equipment industries	0.000	0.000	0.000	0.000	0.000
17	Leather and allied products industries	0.000	0.000	0.000	0.000	0.000
59	Other products and industries, wholesale	0.000	0.000	0.000	0.000	0.000
41	Industrial and heavy (engineering) construction industries	0.000	0.000	0.000	0.000	0.000
	Total	**24.131**	**0.000**	**0.000**	**68.632**	**92.804**

1,3-Butadiene

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
37	Chemical and chemical products industries	270.302	0.058	0.000	0.002	270.362
36	Refined petroleum and coal products industries	22.200	0.000	0.000	0.000	22.289
16	Plastic products industries	17.529	0.000	0.000	0.000	17.529
15	Rubber products industries	0.000	0.000	0.000	0.000	0.000
	Total	**310.031**	**0.058**	**0.000**	**0.002**	**310.180**

Cadmium (and its compounds)

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
29	Primary metal industries	66.105	1.717	0.000	14.000	82.795
06	Mining industries	6.665	6.581	0.000	0.000	13.246
16	Plastic products industries	0.000	0.000	0.000	0.000	0.000
37	Chemical and chemical products industries	0.000	0.000	0.000	0.000	0.000
30	Fabricated metal products industries (except machinery and trans. equipment industries)	0.000	0.000	0.000	0.000	0.000
	Total	**72.770**	**8.298**	**0.000**	**14.000**	**96.041**

Carbon tetrachloride

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
37	Chemical and chemical products industries	12.773	4.732	0.000	0.000	17.505
55	Motor vehicle, parts, and accessories industries, wholesale	0.000	0.000	0.000	0.000	0.630
36	Refined petroleum and coal products industries	0.000	0.000	0.000	0.000	0.000
	Total	**12.773**	**4.732**	**0.000**	**0.000**	**18.135**

Chloroform

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
35	Nonmetallic mineral products industries	0.000	0.000	0.000	0.000	0.130
37	Chemical and chemical products industries	0.039	0.002	0.000	0.001	0.042
	Total	**0.039**	**0.002**	**0.000**	**0.001**	**0.172**

Chromium (and its compounds)

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
29	Primary metal industries	7.382	28.319	0.000	739.668	777.575
39	Other manufacturing industries	0.000	0.000	0.000	5.930	5.930
37	Chemical and chemical products industries	1.700	0.708	0.000	0.000	5.694
35	Nonmetallic mineral products industries	1.723	0.010	0.000	2.300	4.037
30	Fabricated metal products industries (except machinery and trans. equipment industries)	2.593	0.021	0.000	0.176	3.865
32	Transportation equipment industries	0.049	0.000	0.000	0.000	2.878
36	Refined petroleum and coal products industries	0.000	0.000	0.000	0.450	0.450
25	Wood industries	0.000	0.020	0.000	0.000	0.170
16	Plastic products industries	0.000	0.000	0.000	0.000	0.130
31	Machinery industries (except electrical machinery)	0.000	0.000	0.000	0.130	0.130
17	Leather and allied products industries	0.000	0.000	0.000	0.000	0.000
33	Electrical and electronic products industries	0.000	0.000	0.000	0.000	0.000
41	Industrial and heavy (engineering) construction industries	0.000	0.000	0.000	0.000	0.000
06	Mining industries	0.000	0.000	0.000	0.000	0.000
56	Metals, hardware, plumbing, heating, and building materials industries, wholesale	0.000	0.000	0.000	0.000	0.000
	Total	**13.807**	**29.078**	**0.000**	**748.654**	**800.859**

_p_-Dichlorobenzene

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
37	Chemical and chemical products industries	9.900	0.000	0.000	0.500	10.400
39	Other manufacturing industries	0.000	0.000	0.000	0.000	0.000
	Total	**9.900**	**0.000**	**0.000**	**0.500**	**10.400**

1,2-Dichloroethane

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
37	Chemical and chemical products industries	6.886	0.523	0.000	0.040	7.449
35	Nonmetallic mineral products industries	0.000	0.000	0.000	0.000	0.130
	Total	**6.886**	**0.523**	**0.000**	**0.040**	**7.579**

Dichloromethane

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
16	Plastic products industries	1,356.821	0.000	0.000	0.000	1,356.821
37	Chemical and chemical products industries	668.759	0.000	0.000	0.039	670.490
35	Nonmetallic mineral products industries	66.000	0.000	0.000	0.000	66.130
59	Other products and industries, wholesale	55.480	0.000	0.000	0.000	55.610
30	Fabricated metal products industries (except machinery and trans. equipment industries)	42.474	0.000	0.000	0.000	42.474
26	Furniture and fixture industries	18.207	0.000	0.000	0.000	18.207
32	Transportation equipment industries	11.627	0.000	0.000	0.000	11.627
11	Beverage industries	0.000	0.000	0.000	0.000	0.630
36	Refined petroleum and coal products industries	0.000	0.000	0.000	0.000	0.100
29	Primary metal industries	0.000	0.000	0.000	0.000	0.000
39	Other manufacturing industries	0.000	0.000	0.000	0.000	0.000
	Total	**2,219.368**	**0.000**	**0.000**	**0.039**	**2,222.089**

Dimethyl sulfate

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
37	Chemical and chemical products industries	0.011	0.000	0.000	0.000	0.011
	Total	**0.011**	**0.000**	**0.000**	**0.000**	**0.011**

1,4-Dioxane

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
18	Primary textile industries	1.161	4.643	0.000	0.000	5.884
55	Motor vehicle, parts, and accessories industries, wholesale	2.400	0.000	0.000	0.000	2.400
37	Chemical and chemical products industries	0.860	0.000	0.000	0.000	0.860
	Total	**4.421**	**4.643**	**0.000**	**0.000**	**9.144**

Epichlorohydrin

SIC #	Sector Name	Air	Water	Underground	Land	Total Releases‡
16	Plastic products industries	0.000	0.000	0.000	0.000	0.125
37	Chemical and chemical products industries	0.000	0.000	0.000	0.000	0.008
15	Rubber products industries	0.000	0.000	0.000	0.000	0.000
	Total	**0.000**	**0.000**	**0.000**	**0.000**	**0.133**

Ethyl acrylate

SIC #	Sector Name	Air	Water	Underground	Land	Total Releases‡
37	Chemical and chemical products industries	0.130	0.000	0.000	0.000	0.990
16	Plastic products industries	0.000	0.000	0.000	0.000	0.130
	Total	**0.130**	**0.000**	**0.000**	**0.000**	**1.120**

Ethylene oxide

SIC #	Sector Name	Air	Water	Underground	Land	Total Releases‡
37	Chemical and chemical products industries	50.842	0.000	0.000	0.000	51.067
39	Other manufacturing industries	0.000	0.000	0.000	0.000	0.000
	Total	**50.842**	**0.000**	**0.000**	**0.000**	**51.067**

Formaldehyde

SIC #	Sector Name	Air	Water	Underground	Land	Total Releases‡
25	Wood industries	455.173	0.000	0.000	0.000	455.173
27	Paper and allied products industries	19.912	221.560	0.000	0.130	241.602
37	Chemical and chemical products industries	102.345	2.504	69.920	0.000	178.702
35	Nonmetallic mineral products industries	134.520	1.276	0.000	0.630	136.426
16	Plastic products industries	0.173	85.450	0.000	0.000	85.753
59	Other products and industries, wholesale	12.456	0.000	0.000	0.000	12.716
29	Primary metal industries	5.075	0.000	0.000	0.000	5.075
55	Motor vehicle, parts, and accessories industries, wholesale	0.000	0.000	0.000	0.000	0.640
19	Textile products industries	0.000	0.000	0.000	0.000	0.130
33	Electrical and electronic products industries	0.130	0.000	0.000	0.000	0.130
30	Fabricated metal products industries (except machinery and trans. equipment industries)	0.070	0.000	0.000	0.000	0.070
11	Beverage industries	0.000	0.000	0.000	0.000	0.000
10	Food industries	0.000	0.000	0.000	0.000	0.000
	Total	**729.854**	**310.790**	**69.920**	**0.760**	**1,116.417**

Hydrazine

SIC #	Sector Name	Air	Water	Underground	Land	Total Releases‡
49	Other utility industries	0.000	0.000	0.000	0.000	0.897
41	Industrial and heavy (engineering) construction industries	0.000	0.000	0.000	0.000	0.345
29	Primary metal industries	0.000	0.000	0.000	0.000	0.000
37	Chemical and chemical products industries	0.000	0.000	0.000	0.000	0.000
	Total	**0.000**	**0.000**	**0.000**	**0.000**	**1.242**

Lead (and its compounds)

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
29	Primary metal industries	889.192	2.388	0.000	859.628	1,753.169
06	Mining industries	174.654	154.108	0.000	0.025	329.454
35	Nonmetallic mineral products industries	30.000	0.000	0.000	5.700	35.708
37	Chemical and chemical products industries	5.370	0.360	0.000	0.000	7.545
30	Fabricated metal products industries (except machinery and trans. equipment industries)	5.032	1.531	0.000	0.000	7.083
33	Electrical and electronic products industries	2.597	1.079	0.000	0.669	5.366
39	Other manufacturing industries	1.603	0.000	0.000	0.000	1.993
32	Transportation equipment industries	0.895	0.015	0.000	0.000	1.054
15	Rubber products industries	0.298	0.000	0.000	0.000	0.298
36	Refined petroleum and coal products industries	0.000	0.000	0.000	0.140	0.259
16	Plastic products industries	0.000	0.000	0.000	0.000	0.142
56	Metals, hardware, plumbing, heating, and building materials industries, wholesale	0.000	0.140	0.000	0.000	0.140
19	Textile products industries	0.009	0.000	0.000	0.000	0.009
41	Industrial and heavy (engineering) construction industries	0.000	0.000	0.000	0.000	0.000
59	Other products and industries, wholesale	0.000	0.000	0.000	0.000	0.000
	Total	**1,109.650**	**159.621**	**0.000**	**866.162**	**2,142.220**

Mercury (and its compounds)

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
06	Mining industries	3.580	0.148	0.000	0.000	3.728
37	Chemical and chemical products industries	0.057	0.008	0.000	0.000	0.065
36	Refined petroleum and coal products industries	0.000	0.000	0.000	0.011	0.011
29	Primary metal industries	0.000	0.000	0.000	0.000	0.002
	Total	**3.637**	**0.156**	**0.000**	**0.011**	**3.806**

***p,p,*-Methylene*bis*(2-chloroaniline)**

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
37	Chemical and chemical products industries	0.000	0.000	0.000	0.000	0.005
	Total	**0.000**	**0.000**	**0.000**	**0.000**	**0.005**

Nickel (and its compounds)

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
29	Primary metal industries	541.693	65.379	0.000	46.839	655.459
07	Crude petroleum and natural gas industries	5.234	0.449	0.000	27.190	32.873
37	Chemical and chemical products industries	5.194	3.610	0.000	0.000	10.311
06	Mining industries	0.601	2.911	0.000	0.000	3.512
36	Refined petroleum and coal products industries	0.123	0.000	0.000	0.850	1.003
30	Fabricated metal products industries (except machinery and trans. equipment industries)	0.651	0.000	0.000	0.194	0.845
32	Transportation equipment industries	0.199	0.000	0.000	0.000	0.232
31	Machinery industries (except electrical machinery)	0.000	0.000	0.000	0.130	0.130
16	Plastic products industries	0.000	0.000	0.000	0.000	0.130
39	Other manufacturing industries	0.000	0.000	0.000	0.000	0.001
33	Electrical and electronic products industries	0.000	0.000	0.000	0.000	0.000
41	Industrial and heavy (engineering) construction industries	0.000	0.000	0.000	0.000	0.000
10	Food industries	0.000	0.000	0.000	0.000	0.000
	Total	**553.695**	**72.349**	**0.000**	**75.203**	**704.496**

Nitrilotriacetic acid

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
37	Chemical and chemical products industries	0.000	0.000	0.000	0.000	1.001
	Total	**0.000**	**0.000**	**0.000**	**0.000**	**1.001**

2-Nitropropane

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
16	Plastic products industries	0.000	0.000	0.000	0.000	0.125
	Total	**0.000**	**0.000**	**0.000**	**0.000**	**0.125**

Propylene oxide

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
37	Chemical and chemical products industries	10.941	0.000	0.000	0.000	10.941
16	Plastic products industries	0.000	0.000	0.000	0.000	0.130
	Total	**10.941**	**0.000**	**0.000**	**0.000**	**11.071**

Styrene

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
29	Primary metal industries	1,101.453	0.004	0.000	0.000	1,102.870
37	Chemical and chemical products industries	309.701	0.020	0.185	0.196	314.172
16	Plastic products industries	198.445	0.380	0.000	0.000	200.720
32	Transportation equipment industries	132.352	0.000	0.000	0.000	132.352
39	Other manufacturing industries	12.030	0.000	0.000	0.000	12.030
26	Furniture and fixture industries	10.614	0.000	0.000	0.000	10.614
45	Transportation industries	0.000	10.000	0.000	0.000	10.000
36	Refined petroleum and coal products industries	6.190	0.000	0.000	0.000	6.190
99	Other service industries	2.289	0.000	0.000	0.000	2.289
09	Service industries incidental to mineral extraction	0.728	0.000	0.000	0.000	0.728
15	Rubber products industries	0.000	0.000	0.000	0.000	0.500
59	Other products and industries, wholesale	0.000	0.000	0.000	0.000	0.048
81	Federal government service industries	0.000	0.000	0.000	0.000	0.005
	Total	**1,773.802**	**10.404**	**0.185**	**0.196**	**1,792.518**

Styrene oxide

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
16	Plastic products industries	0.000	0.000	0.000	0.000	0.130
	Total	**0.000**	**0.000**	**0.000**	**0.000**	**0.130**

Tetrachloroethylene

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
29	Primary metal industries	122.180	0.000	0.000	0.000	122.180
33	Electrical and electronic products industries	22.070	0.000	0.000	0.000	22.070
28	Printing, publishing, and allied industries	8.808	0.000	0.000	0.000	8.808
37	Chemical and chemical products industries	6.440	0.010	0.000	0.000	7.448
59	Other products and industries, wholesale	1.014	0.000	0.000	0.000	1.175
18	Primary textile industries	0.000	0.000	0.000	0.000	0.700
11	Beverage industries	0.000	0.000	0.000	0.000	0.630
36	Refined petroleum and coal products industries	0.130	0.064	0.000	0.000	0.194
35	Nonmetallic mineral products industries	0.000	0.000	0.000	0.000	0.130
27	Paper and allied products industries	0.000	0.000	0.000	0.000	0.000
	Total	**160.642**	**0.074**	**0.000**	**0.000**	**163.335**

Thiourea

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
29	Primary metal industries	0.000	0.000	0.000	0.000	0.000
	Total	**0.000**	**0.000**	**0.000**	**0.000**	**0.000**

Toluene-2,4-diisocyanate

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
37	Chemical and chemical products industries	0.000	0.000	0.000	0.000	0.148
32	Transportation equipment industries	0.000	0.000	0.000	0.000	0.000
16	Plastic products industries	0.000	0.000	0.000	0.000	0.000
	Total	**0.000**	**0.000**	**0.000**	**0.000**	**0.148**

Toluenediisocyanate (mixed isomers)

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
16	Plastic products industries	0.230	0.000	0.000	0.000	0.631
37	Chemical and chemical products industries	0.011	0.000	0.000	0.000	0.147
39	Other manufacturing industries	0.000	0.000	0.000	0.000	0.123
33	Electrical and electronic products industries	0.000	0.000	0.000	0.000	0.112
32	Transportation equipment industries	0.000	0.000	0.000	0.000	0.016
26	Furniture and fixture industries	0.003	0.000	0.000	0.000	0.003
19	Textile products industries	0.001	0.000	0.000	0.000	0.001
	Total	**0.245**	**0.000**	**0.000**	**0.000**	**1.033**

Trichloroethylene

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
29	Primary metal industries	419.483	0.000	0.000	0.000	419.483
30	Fabricated metal products industries (except machinery and trans. equipment industries)	148.430	0.000	0.000	0.000	148.430
15	Rubber products industries	97.863	0.000	0.000	0.000	97.863
32	Transportation equipment industries	89.334	0.000	0.000	0.000	89.334
16	Plastic products industries	33.995	0.000	0.000	0.000	33.995

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
33	Electrical and electronic products industries	33.309	0.022	0.000	0.000	33.331
39	Other manufacturing industries	22.748	0.000	0.000	0.000	22.748
26	Furniture and fixture industries	13.000	0.000	0.000	0.000	13.000
37	Chemical and chemical products industries	0.000	0.000	0.000	0.000	0.702
59	Other products and industries, wholesale	0.001	0.000	0.000	0.000	0.337
35	Nonmetallic mineral products industries	0.000	0.000	0.000	0.000	0.130
11	Beverage industries	0.000	0.000	0.000	0.000	0.130
36	Refined petroleum and coal products industries	0.000	0.000	0.000	0.000	0.000
10	Food industries	0.000	0.000	0.000	0.000	0.000
	Total	**858.163**	**0.022**	**0.000**	**0.000**	**859.483**

Vinyl chloride

SIC #	Sector Name	Air	Water	Under-ground	Land	Total Releases‡
37	Chemical and chemical products industries	23.146	0.164	0.000	0.000	23.340
16	Plastic products industries	0.000	0.000	0.000	0.000	0.255
39	Other manufacturing industries	0.000	0.000	0.000	0.000	0.130
36	Refined petroleum and coal products industries	0.000	0.000	0.000	0.000	0.000
	Total	**23.146**	**0.164**	**0.000**	**0.000**	**23.725**

* Toxic includes Schedule 1 and CEPA-Toxic.
† Carcinogenic and potentially carcinogenic include IARC 1, 2a, 2b and NTP classification K and P substances.
‡ Total releases may be greater than the sum of the releases by environmental medium since releases of less than one tonne could be reported as an undifferentiated total release.

Appendix 7: Surface waters receiving NPRI substances

Surface Water Name and Province	CAS #	Substance Name	Releases (Tonnes)	# of Reports
Alberni Inlet, BC	NA	Zinc (and its compounds)	1.500	1
	NA	Copper (and its compounds)	0.130	1
	7664-41-7	Ammonia	0.130	1
		Total	1.760	3
Amikougami Creek, PQ	NA	Cyanides (ionic)	0.058	1
		Total	0.058	1
Assiniboine River, MB	6484-52-2	Ammonium nitrate (solution)	6.900	1
	7783-20-2	Ammonium sulfate (solution)	2.300	1
	7664-41-7	Ammonia	0.100	1
		Total	9.300	3
Athabasca River, AB	7664-41-7	Ammonia	3.847	3
	111-42-2	Diethanolamine	3.800	1
	107-21-1	Ethylene glycol	3.700	2
	NA	Nickel (and its compounds)	0.449	1
		Total	11.796	7
Baker Creek, NT	NA	Arsenic (and its compounds)	0.630	1
		Total	0.630	1
Balmer Lake, ON	NA	Cyanides (ionic)	0.660	1
	7440-66-6	Zinc (fume or dust)	0.032	1
		Total	0.692	2
Balmer River, ON	NA	Arsenic (and its compounds)	2.126	1
	NA	Cyanides (ionic)	1.311	1
		Total	3.437	2
Battle River, AB	7664-41-7	Ammonia	58.683	1
		Total	58.683	1

Surface Water Name and Province	CAS #	Substance Name	Releases (Tonnes)	# of Reports
Baie de Chaleur, NB	7440-62-2	Vanadium (fume or dust)	2.600	1
	NA	Cadmium (and its compounds)	1.716	1
	NA	Lead (and its compounds)	1.034	1
	NA	Arsenic (and its compounds)	0.630	1
	NA	Silver (and its compounds)	0.001	1
		Total	5.981	5
Bay of Grand-Pabos, PQ	7664-38-2	Phosphoric acid	25.000	1
		Total	25.000	1
Bearskin Lake, BC	NA	Cyanides (ionic)	0.003	1
		Total	0.003	1
Bourlamaque River, PQ	NA	Cyanides (ionic)	0.130	1
	NA	Lead (and its compounds)	0.130	1
	NA	Zinc (and its compounds)	0.130	1
		Total	0.390	3
Bousquet River, PQ	NA	Copper (and its compounds)	85.940	1
		Total	85.940	1
Bow River, AB	6484-52-2	Ammonium nitrate (solution)	0.700	1
	7664-41-7	Ammonia	0.100	1
		Total	0.800	2
Burntwood River, MB	NA	Nickel (and its compounds)	16.319	1
	NA	Arsenic (and its compounds)	0.725	1
	NA	Cobalt (and its compounds)	0.555	1
	NA	Copper (and its compounds)	0.425	1
		Total	18.024	4
Burrard Inlet, BC	7664-41-7	Ammonia	15.880	1
	100-42-5	Styrene	10.000	1
	7782-50-5	Chlorine	6.600	1
	111-42-2	Diethanolamine	1.660	1
	108-95-2	Phenol	1.010	1
	107-21-1	Ethylene glycol	0.437	1
	107-06-2	1,2-Dichloroethane	0.432	1
	NA	Zinc (and its compounds)	0.160	1
		Total	36.179	8
Canagagique Creek, ON	108-88-3	Toluene	0.130	1
		Total	0.130	1
Columbia River, BC	NA	Zinc (and its compounds)	4,444.000	1
	NA	Copper (and its compounds)	898.000	1
	7664-38-2	Phosphoric acid	774.000	1
	7664-93-9	Sulfuric acid	719.000	1
	7783-20-2	Ammonium sulfate (solution)	577.000	1
	7664-41-7	Ammonia	403.450	2
	NA	Lead (and its compounds)	149.000	1
	NA	Arsenic (and its compounds)	41.500	1
	NA	Cadmium (and its compounds)	6.000	1
	NA	Silver (and its compounds)	3.320	1
	NA	Antimony (and its compounds)	3.300	1
	NA	Manganese (and its compounds)	1.400	1
	NA	Mercury (and its compounds)	0.140	1
		Total	8,020.110	14
Coniston Creek, ON	NA	Nickel (and its compounds)	1.589	1
	NA	Copper (and its compounds)	0.077	1
	7440-62-2	Vanadium (fume or dust)	0.030	1
	NA	Zinc (and its compounds)	0.008	1
	NA	Cobalt (and its compounds)	0.008	1
	NA	Manganese (and its compounds)	0.004	1
	NA	Arsenic (and its compounds)	0.003	1
	NA	Chromium (and its compounds)	0.003	1
	NA	Lead (and its compounds)	0.003	1
	NA	Cadmium (and its compounds)	0.001	1
		Total	1.726	10
Contwoyto Lake, NT	NA	Zinc (and its compounds)	0.209	1
	NA	Arsenic (and its compounds)	0.039	1
	NA	Cyanides (ionic)	0.010	1
	NA	Lead (and its compounds)	0.004	1
		Total	0.262	4

Surface Water Name and Province	CAS #	Substance Name	Releases (Tonnes)	# of Reports
Demontigny Lake, PQ	NA	Copper (and its compounds)	0.102	1
	NA	Cyanides (ionic)	0.017	1
	NA	Lead (and its compounds)	0.019	1
		Total	0.138	3
Detroit River, ON	7664-41-7	Ammonia	158.500	1
	7440-66-6	Zinc (fume or dust)	57.000	1
	NA	Manganese (and its compounds)	6.200	1
	108-95-2	Phenol	5.500	1
	7429-90-5	Aluminum (fume or dust)	1.400	1
	NA	Zinc (and its compounds)	0.200	1
	107-21-1	Ethylene glycol	0.110	1
	NA	Lead (and its compounds)	0.003	1
		Total	228.913	8
Discovery Passage, BC	7647-01-0	Hydrochloric acid	0.350	1
	7664-41-7	Ammonia	0.160	1
		Total	0.510	2
Dodds Creek, ON	NA	Zinc (and its compounds)	0.110	1
	7664-93-9	Sulfuric acid	0.000	1
	107-21-1	Ethylene glycol	0.000	1
	7664-38-2	Phosphoric acid	0.000	1
		Total	0.110	4
Dormenan Creek, PQ	NA	Cyanides (ionic)	0.880	1
	NA	Zinc (and its compounds)	0.380	1
	NA	Copper (and its compounds)	0.380	1
	NA	Lead (and its compounds)	0.130	1
		Total	1.770	4
Estuary Pond, NB	7664-41-7	Ammonia	16.133	1
		Total	16.133	1
Fox Creek, ON	NA	Zinc (and its compounds)	7.435	1
	NA	Copper (and its compounds)	0.522	1
		Total	7.957	2
Frank Lake, ON	NA	Cyanides (ionic)	0.380	1
		Total	0.380	1
Fraser River, BC	7664-41-7	Ammonia	18.000	1
	67-56-1	Methanol	1.438	1
	7664-93-9	Sulfuric acid	0.200	1
	50-00-0	Formaldehyde	0.130	1
	7647-01-0	Hydrochloric acid	0.050	1
		Total	19.818	5
Gold Bar Creek, AB	107-21-1	Ethylene glycol	2.780	1
	NA	Manganese (and its compounds)	0.026	1
	NA	Zinc (and its compounds)	0.007	1
	NA	Lead (and its compounds)	0.004	1
	NA	Nickel (and its compounds)	0.003	1
	NA	Chromium (and its compounds)	0.002	1
		Total	2.822	6
Grande Rivière du Loup, PQ	67-63-0	Isopropyl alcohol	0.130	1
		Total	0.130	1
Great Slave Lake, NT	NA	Cyanides (ionic)	0.263	1
	NA	Copper (and its compounds)	0.262	1
		Total	0.525	2
Halifax Harbour, NS	7664-41-7	Ammonia	11.830	5
	108-88-3	Toluene	0.502	4
	NA	Zinc (and its compounds)	0.447	2
	1330-20-7	Xylene (mixed isomers)	0.404	4
	91-20-3	Naphthalene	0.369	3
	71-43-2	Benzene	0.290	4
	95-63-6	1,2,4-Trimethylbenzene	0.087	1
	100-41-4	Ethylbenzene	0.075	3
	110-82-7	Cyclohexane	0.067	2
	98-82-8	Cumene	0.020	1
	108-95-2	Phenol	0.010	1
		Total	14.101	30

Surface Water Name and Province	CAS #	Substance Name	Releases (Tonnes)	# of Reports
Hamilton Harbour, ON	7664-41-7	Ammonia	35.500	1
	67-56-1	Methanol	11.000	1
	107-21-1	Ethylene glycol	7.400	1
	NA	Zinc (and its compounds)	2.430	1
	NA	Lead (and its compounds)	0.280	1
		Total	56.610	5
Harricana River, PQ	NA	Cyanides (ionic)	0.130	1
	NA	Lead (and its compounds)	0.130	1
		Total	0.260	2
Hesse Lake, PQ	6484-52-2	Ammonium nitrate (solution)	5.000	1
		Total	5.000	1
Hillsborough River, PE	7664-93-9	Sulfuric acid	21.700	1
	67-64-1	Acetone	8.050	1
	67-56-1	Methanol	7.090	1
	108-88-3	Toluene	0.077	1
		Total	36.917	4
Hour Lake, ON	NA	Cyanides (ionic)	0.030	1
		Total	0.030	1
Humber River, ON	108-88-3	Toluene	5.181	1
	110-82-7	Cyclohexane	0.120	1
	1330-20-7	Xylene (mixed isomers)	0.077	1
		Total	5.378	3
Junction Creek, ON	NA	Nickel (and its compounds)	25.144	3
	NA	Copper (and its compounds)	7.274	3
	NA	Cobalt (and its compounds)	0.427	2
	NA	Lead (and its compounds)	0.202	2
	NA	Arsenic (and its compounds)	0.108	1
	NA	Silver (and its compounds)	0.015	1
		Total	33.170	12
Kaministikwia River, ON	107-21-1	Ethylene glycol	2.900	1
	7664-41-7	Ammonia	0.922	1
		Total	3.822	2
Kitimat Arm of the Douglas Channel, BC	67-56-1	Methanol	11.000	1
	7664-41-7	Ammonia	7.300	1
		Total	18.300	2
Kitimat River, BC	108-95-2	Phenol	2.430	1
		Total	2.430	1
Kootenay River, BC	6484-52-2	Ammonium nitrate (solution)	20.100	1
	67-56-1	Methanol	9.000	1
		Total	29.100	2
Lachine Canal, PQ	NA	Zinc (and its compounds)	0.080	1
	NA	Lead (and its compounds)	0.020	1
		Total	0.100	2
LaHave River, NS	7664-41-7	Ammonia	0.421	1
	NA	Zinc (and its compounds)	0.133	1
	NA	Copper (and its compounds)	0.012	1
		Total	0.566	3
Lake Erie, ON	7647-01-0	Hydrochloric acid	16.900	1
	107-21-1	Ethylene glycol	10.950	1
	7664-41-7	Ammonia	1.715	1
	NA	Manganese (and its compounds)	1.675	1
	NA	Zinc (and its compounds)	1.257	1
	7782-50-5	Chlorine	1.095	1
	NA	Copper (and its compounds)	0.233	1
	NA	Chromium (and its compounds)	0.070	1
	71-43-2	Benzene	0.004	1
	1330-20-7	Xylene (mixed isomers)	0.004	1
	100-42-5	Styrene	0.004	1
	91-20-3	Naphthalene	0.004	1
	108-88-3	Toluene	0.004	1
		Total	33.915	13
Lake Huron, ON	7664-93-9	Sulfuric acid	129.641	1
		Total	129.641	1

Surface Water Name and Province	CAS #	Substance Name	Releases (Tonnes)	# of Reports
Lake Ontario, ON	7664-93-9	Sulfuric acid	112.382	3
	107-21-1	Ethylene glycol	27.740	1
	NA	Manganese (and its compounds)	9.300	1
	NA	Zinc (and its compounds)	9.145	2
	123-91-1	1,4-Dioxane	4.643	1
	NA	Copper (and its compounds)	1.064	4
	67-56-1	Methanol	1.000	1
	7664-41-7	Ammonia	0.990	2
	7782-50-5	Chlorine	0.700	1
	NA	Nickel (and its compounds)	0.659	1
	NA	Chromium (and its compounds)	0.638	1
	NA	Lead (and its compounds)	0.205	3
	7664-39-3	Hydrogen fluoride	0.132	1
	92-52-4	Biphenyl	0.060	1
	120-12-7	Anthracene	0.037	1
	91-20-3	Naphthalene	0.007	1
	NA	Antimony (and its compounds)	0.003	1
	71-43-2	Benzene	0.002	1
	1330-20-7	Xylene (mixed isomers)	0.000	1
	100-41-4	Ethylbenzene	0.000	1
Lake Ontario, ON (continued)	100-42-5	Styrene	0.000	1
	1163-19-5	Decabromodiphenyl oxide	0.000	1
	108-88-3	Toluene	0.000	1
		Total	168.707	32
Lake Superior, ON	67-56-1	Methanol	2,271.000	1
	67-64-1	Acetone	0.315	1
	78-93-3	Methyl ethyl ketone	0.010	1
		Total	2,271.325	3
La Lièvre River, PQ	67-56-1	Methanol	192.600	1
	7664-38-2	Phosphoric acid	2.175	1
	NA	Chromium (and its compounds)	0.003	1
		Total	194.778	3
Lim Lake, ON	NA	Cyanides (ionic)	0.878	2
	NA	Copper (and its compounds)	0.395	1
		Total	1.273	3
Little River, NB	67-56-1	Methanol	64.400	1
	7664-41-7	Ammonia	43.000	1
		Total	107.400	2
Little South Tomogonops River, NB	NA	Zinc (and its compounds)	13.880	1
	NA	Copper (and its compounds)	1.425	1
	NA	Lead (and its compounds)	0.732	1
		Total	16.037	3
Magusi River, ON	NA	Cyanides (ionic)	0.006	1
		Total	0.006	1
Malaspina Strait, BC	10049-04-4	Chlorine dioxide	4.800	1
		Total	4.800	1
McLarens Creek, ON	NA	Nickel (and its compounds)	0.095	1
	NA	Chromium (and its compounds)	0.027	1
		Total	0.122	2
Muchalat Inlet, BC	7664-41-7	Ammonia	2.500	1
		Total	2.500	1
Myra Creek, BC	NA	Zinc (and its compounds)	3.330	1
	NA	Copper (and its compounds)	0.380	1
	NA	Lead (and its compounds)	0.000	1
		Total	3.710	3
Nepisiguit River, NB	7664-93-9	Sulfuric acid	51.900	1
	7664-41-7	Ammonia	3.900	1
		Total	55.800	2
Neroutsos Inlet, BC	67-56-1	Methanol	1,600.000	1
		Total	1,600.000	1
Niagara River, ON	7782-50-5	Chlorine	0.390	6
	NA	Manganese (and its compounds)	0.002	1
		Total	0.392	7
Nith River, ON	50-00-0	Formaldehyde	1.248	1
		Total	1.248	1

Surface Water Name and Province	CAS #	Substance Name	Releases (Tonnes)	# of Reports
North Saskatchewan River, AB	7783-20-2	Ammonium sulfate (solution)	450.000	2
	6484-52-2	Ammonium nitrate (solution)	441.000	2
	NA	Nickel (and its compounds)	3.610	1
	7664-41-7	Ammonia	3.597	7
	NA	Cobalt (and its compounds)	1.120	1
	NA	Zinc (and its compounds)	0.780	1
	NA	Chromium (and its compounds)	0.600	1
	NA	Copper (and its compounds)	0.270	1
	7664-93-9	Sulfuric acid	0.200	1
	107-21-1	Ethylene glycol	0.050	1
	91-20-3	Naphthalene	0.026	1
	75-01-4	Vinyl chloride	0.003	1
	67-66-3	Chloroform	0.002	1
		Total	901.258	21
North Unnamed Creek, BC	NA	Cyanides (ionic)	0.857	1
		Total	0.857	1
Northumberland Channel, BC	7664-41-7	Ammonia	4.340	1
	7782-50-5	Chlorine	1.982	3
		Total	6.322	4
Onaping River, ON	NA	Copper (and its compounds)	0.380	1
	NA	Cyanides (ionic)	0.136	1
	NA	Nickel (and its compounds)	1.160	1
		Total	1.676	3
Otonabee River, ON	7664-41-7	Ammonia	122.600	1
	7782-50-5	Chlorine	2.600	1
		Total	125.200	2
Ottawa River, PQ/ON	7664-93-9	Sulfuric acid	2,500.000	1
	7664-41-7	Ammonia	77.900	1
	75-07-0	Acetaldehyde	13.200	1
	50-00-0	Formaldehyde	1.767	1
	108-95-2	Phenol	0.481	1
		Total	2,593.348	5
Peace River, AB/BC	107-21-1	Ethylene glycol	9.680	2
	7664-41-7	Ammonia	2.305	2
	7782-50-5	Chlorine	0.394	1
	1330-20-7	Xylene (mixed isomers)	0.009	1
	108-88-3	Toluene	0.008	1
	71-43-2	Benzene	0.008	1
		Total	12.404	8
Pell Creek, ON	NA	Chromium (and its compounds)	0.010	1
		Total	0.010	1
Porcupine River, ON	7664-41-7	Ammonia	32.304	1
	NA	Zinc (and its compounds)	4.341	1
	NA	Manganese (and its compounds)	4.282	1
	NA	Selenium (and its compounds)	3.896	1
	NA	Arsenic (and its compounds)	0.824	1
	NA	Lead (and its compounds)	0.781	1
	NA	Copper (and its compounds)	0.283	1
	NA	Cadmium (and its compounds)	0.201	1
		Total	46.912	8
Quevillon River, PQ	67-56-1	Methanol	57.380	1
		Total	57.380	1
Red Deer River, AB	107-21-1	Ethylene glycol	9.000	1
		Total	9.000	1
Richelieu River, PQ	6484-52-2	Ammonium nitrate (solution)	170.000	1
	67-64-1	Acetone	30.000	1
	7664-41-7	Ammonia	18.000	1
	7647-01-0	Hydrochloric acid	10.000	1
	NA	Lead (and its compounds)	0.140	1
	NA	Zinc (and its compounds)	0.131	2
		Total	228.271	7
Rouyn Lake, PQ	NA	Zinc (and its compounds)	4.420	1
	NA	Manganese (and its compounds)	3.660	1
	7429-90-5	Aluminum (fume or dust)	3.390	1
	NA	Copper (and its compounds)	1.850	1

Surface Water Name and Province	CAS #	Substance Name	Releases (Tonnes)	# of Reports
	NA	Nickel (and its compounds)	0.660	1
	NA	Lead (and its compounds)	0.600	1
	NA	Arsenic (and its compounds)	0.240	1
		Total	14.820	7
Rupert Inlet, BC	NA	Copper (and its compounds)	13,439.526	1
		Total	13,439.526	1
Saint John River, NB	67-56-1	Methanol	3,129.039	1
	7664-41-7	Ammonia	51.360	1
	7647-01-0	Hydrochloric acid	6.442	1
		Total	3,186.841	3
Saskatchewan River, MB	6484-52-2	Ammonium nitrate (solution)	3.600	1
		Total	3.600	1
Shawinigan River, PQ	50-00-0	Formaldehyde	221.430	1
		Total	221.430	1
Similkameen River, BC	NA	Copper (and its compounds)	0.003	1
		Total	0.003	1
South Little River, NB	NA	Zinc (and its compounds)	4.800	1
	NA	Lead (and its compounds)	0.880	1
	NA	Copper (and its compounds)	0.380	1
		Total	6.060	3
South Porcupine River, ON	NA	Cyanides (ionic)	0.446	1
		Total	0.446	1
South Saskatchewan River, SK/AB	7664-93-9	Sulfuric acid	87.000	1
	7664-41-7	Ammonia	31.250	2
		Total	118.250	3
St. Charles River, PQ	55-63-0	Nitroglycerin	9.000	1
	7697-37-2	Nitric acid	7.700	1
	121-14-2	2,4-Dinitrotoluene	4.000	1
	67-64-1	Acetone	0.500	1
	NA	Zinc (and its compounds)	0.128	1
		Total	21.328	5
St. Clair River, ON	6484-52-2	Ammonium nitrate (solution)	39.500	1
	7664-41-7	Ammonia	38.421	5
	67-56-1	Methanol	3.188	1
	NA	Zinc (and its compounds)	1.400	1
	75-65-0	*tert*-Butyl alcohol	1.100	1
	74-87-3	Chloromethane	0.880	1
	110-82-7	Cyclohexane	0.500	2
	1634-04-4	Methyl *tert*-butyl ether	0.400	1
	75-00-3	Chloroethane	0.370	1
	NA	Lead (and its compounds)	0.360	1
	75-01-4	Vinyl chloride	0.160	1
	1330-20-7	Xylene (mixed isomers)	0.140	3
	75-05-8	Acetonitrile	0.130	1
	71-43-2	Benzene	0.128	5
	100-41-4	Ethylbenzene	0.106	3
	107-06-2	1,2-Dichloroethane	0.091	1
	108-88-3	Toluene	0.075	4
	7782-50-5	Chlorine	0.063	1
	105-99-0	1,3-Butadiene	0.058	1
	107-13-1	Acrylonitrile	0.040	1
	100-42-5	Styrene	0.020	2
		Total	87.130	38
St. François River, PQ	67-56-1	Methanol	22.000	1
	7783-20-2	Ammonium sulphate (solution)	2.000	1
	NA	Lead (and its compounds)	1.000	1
		Total	25.000	3
St. Germain River, PQ	67-63-0	Isopropyl alcohol	0.544	1
		Total	0.544	1
St. Lawrence River, PQ/ON	7664-93-9	Sulfuric acid	15,010.200	5
	67-56-1	Methanol	460.970	3
	7664-41-7	Ammonia	159.302	8
	50-00-0	Formaldehyde	85.457	2
	NA	Manganese (and its compounds)	77.500	2

Surface Water Name and Province	CAS #	Substance Name	Releases (Tonnes)	# of Reports
	7647-01-0	Hydrochloric acid	44.000	2
	7697-37-2	Nitric acid	28.500	2
	NA	Chromium (and its compounds)	27.435	3
	NA	Nickel (and its compounds)	20.300	1
	111-42-2	Diethanolamine	19.740	1
	71-36-3	*n*-Butyl alcohol	18.680	1
	NA	Zinc (and its compounds)	14.989	5
	75-15-0	Carbon disulphide	10.405	1
	78-83-1	*i*-Butyl alcohol	6.440	1
	103-23-1	*Bis*(2-ethylhexyl) adipate	5.600	1
	56-23-5	Carbon tetrachloride	4.732	2
	108-95-2	Phenol	4.319	3
	NA	Selenium (and its compounds)	3.560	1
	NA	Copper (and its compounds)	2.751	3
	107-21-1	Ethylene glycol	1.382	2
	1330-20-7	Xylene (mixed isomers)	1.315	3
	7782-50-5	Chlorine	1.178	2
	71-43-2	Benzene	0.619	3
	79-06-1	Acrylamide	0.610	1
	100-42-5	Styrene	0.380	1
	108-88-3	Toluene	0.357	3
	7664-39-3	Hydrogen fluoride	0.290	1
	95-63-6	1,2,4-Trimethylbenzene	0.146	1
	100-41-4	Ethylbenzene	0.133	2
	107-13-1	Acrylonitrile	0.122	1
	67-64-1	Acetone	0.081	1
	127-18-4	Tetrachloroethylene	0.074	2
	NA	Lead (and its compounds)	0.060	2
	7723-14-0	Phosphorus (yellow or white)	0.030	1
	NA	Mercury (and its compounds)	0.008	1
		Total	16,011.665	74
St. Mary River, BC	NA	Zinc (and its compounds)	2.566	1
	NA	Lead (and its compounds)	0.489	1
		Total	3.055	2
St. Mary's River, ON	7664-41-7	Ammonia	448.952	1
	NA	Zinc (and its compounds)	1.274	1
	NA	Cyanides (ionic)	1.162	1
	1330-20-7	Xylene (mixed isomers)	0.002	1
	108-88-3	Toluene	0.001	1
	71-43-2	Benzene	0.001	1
		Total	451.392	6
St. Maurice River, PQ	67-56-1	Methanol	3,174.800	1
	79-06-1	Acrylamide	0.316	1
	NA	Zinc (and its compounds)	0.130	1
		Total	3,175.246	3
Strait of Canso, NS	67-56-1	Methanol	895.230	1
	7664-93-9	Sulfuric acid	137.800	1
	7647-01-0	Hydrochloric acid	2.336	1
	10049-04-4	Chlorine dioxide	0.473	1
		Total	1,035.839	4
Sunday Lake, ON	NA	Cyanides (ionic)	0.510	1
	NA	Lead (and its compounds)	0.130	1
		Total	0.640	2
Sydney Harbour, NS	107-21-1	Ethylene glycol	5.000	1
		Total	5.000	1
Thornbrough Channel of Howe Sound, B.C.	7664-41-7	Ammonia	0.100	1
		Total	0.100	1
Thousand Islands River, PQ	NA	Zinc (and its compounds)	1.757	1
	7664-41-7	Ammonia	1.095	1
	50-00-0	Formaldehyde	0.730	1
	7664-38-2	Phosphoric acid	0.037	1
		Total	3.619	4
Twelve Mile Creek, ON	107-21-1	Ethylene glycol	5.000	1
	7664-41-7	Ammonia	0.630	1

Surface Water Name and Province	CAS #	Substance Name	Releases (Tonnes)	# of Reports
	NA	Manganese (and its compounds)	0.012	1
		Total	5.642	3
Twin Lakes Creek, NT	NA	Zinc (and its compounds)	0.016	1
		Total	0.016	1
Vanze Creek, PQ	NA	Zinc (and its compounds)	0.803	1
	NA	Arsenic (and its compounds)	0.158	1
	NA	Lead (and its compounds)	0.103	1
	NA	Copper (and its compounds)	0.095	1
	NA	Nickel (and its compounds)	0.064	1
		Total	1.223	5
Wabamun Ash Lagoons, AB	7664-41-7	Ammonia	12.200	1
		Total	12.200	1
Wabigoon River, ON	107-21-1	Ethylene glycol	4.800	1
	6484-52-2	Ammonium nitrate (solution)	3.000	1
		Total	7.800	2
Wapiti River, AB	7664-41-7	Ammonia	0.830	1
		Total	0.830	1
Wascana Creek, SK	7782-50-5	Chlorine	26.100	1
		Total	26.100	1
Wawagosic River, PQ	NA	Zinc (and its compounds)	1.300	1
		Total	1.300	1
Wekusko Lake, MB	7664-41-7	Ammonia	8.135	1
	NA	Zinc (and its compounds)	3.288	3
	NA	Copper (and its compounds)	0.199	3
		Total	11.622	7
Welland Canal, ON	108-95-2	Phenol	2.085	1
	NA	Manganese (and its compounds)	0.220	2
	NA	Zinc (and its compounds)	0.067	1
	NA	Chromium (and its compounds)	0.019	1
	107-21-1	Ethylene glycol	0.004	1
		Total	2.395	6
Welland River, ON	7664-41-7	Ammonia	19.807	1
	7723-14-0	Phosphorus (yellow or white)	4.556	1
	NA	Nickel (and its compounds)	0.610	1
	NA	Zinc (and its compounds)	0.360	1
	NA	Chromium (and its compounds)	0.230	1
	75-01-4	Vinyl chloride	0.001	1
		Total	25.564	6
Whitesand River, ON	67-56-1	Methanol	120.000	1
	7664-93-9	Sulfuric acid	30.003	1
	NA	Zinc (and its compounds)	0.089	1
	NA	Copper (and its compounds)	0.024	1
		Total	150.116	4
Wolf Lake, SK	7664-41-7	Ammonia	0.044	1
		Total	0.044	1
Winnipeg River, ON/MB	67-56-1	Methanol	120.000	1
	7664-93-9	Sulfuric acid	30.003	1
		Total	150.003	2
Wye River, ON	7664-41-7	Ammonia	10.120	1
	NA	Zinc (and its compounds)	0.089	1
	NA	Lead (and its compounds)	0.079	1
	79-01-6	Trichloroethylene	0.022	1
		Total	10.310	4
York River, PQ	NA	Manganese (and its compounds)	6.130	1
	NA	Copper (and its compounds)	3.561	1
	NA	Nickel (and its compounds)	1.687	1
	NA	Lead (and its compounds)	1.580	1
	NA	Zinc (and its compounds)	1.423	1
	NA	Cadmium (and its compounds)	0.380	1
	NA	Arsenic (and its compounds)	0.212	1
	NA	Mercury (and its compounds)	0.008	1
		Total	14.981	8

Appendix 8: Substances transferred by category (tonnes)

CAS #	Substance Name	Physical	Chemical	Biological	Incineration	Landfill	Storage	MSTP*	Underground	Total Transfers	# of Reports
117-81-7	*Bis*(2-ethylhexyl) phthalate	32.073	0.000	0.000	6.858	14,852.205	0.003	0.000	0.000	14,891.139	33
7429-90-5	Aluminum (fume or dust)	1.000	0.000	0.000	8.940	7,756.462	0.000	0.000	0.000	7,766.402	35
NA	Copper (and its compounds)	6,652.737	58.004	0.000	17.881	293.937	9.712	5.376	0.772	7,038.419	243
NA	Zinc (and its compounds	880.377	260.877	0.000	46.238	4,069.363	15.229	14.761	0.000	5,286.846	283
67-56-1	Methanol	107.263	0.000	2,613.660	596.741	59.956	0.000	58.175	960.070	4,395.865	284
NA	Manganese (and its compounds)	8.017	1,174.712	0.000	0.000	2,517.297	87.725	4.023	0.000	3,791.774	192
7664-93-9	Sulfuric acid	49.710	1,138.948	0.012	15.150	368.306	250.263	100.785	800.000	2,723.174	387
108-88-3	Toluene	326.132	65.262	8.228	1,485.159	14.153	10.313	1.047	7.455	1,917.749	313
107-21-1	Ethylene glycol	14.598	46.060	801.546	148.792	13.479	7.748	550.352	76.846	1,659.420	232
1330-20-7	Xylene (mixed isomers)	155.860	0.000	0.688	1,156.631	44.562	21.719	56.705	3.157	1,439.323	285
7440-66-6	Zinc (fume or dust)	1.280	0.000	0.000	10.140	1,420.836	0.000	0.032	0.000	1,432.288	36
NA	Lead (and its compounds)	558.036	4.737	0.000	3.246	733.318	0.000	2.314	0.000	1,301.650	160
7647-01-0	Hydrochloric acid	52.600	225.508	0.000	0.731	26.427	22.310	681.770	0.000	1,009.346	211
NA	Chromium (and its compounds)	85.806	410.816	0.000	2.156	296.718	199.058	8.092	0.000	1,002.646	189
1332-21-4	Asbestos	0.000	0.000	0.000	0.000	935.063	0.000	0.000	0.000	935.063	37
67-63-0	Isopropyl alcohol	12.024	37.723	1.914	521.907	2.104	208.731	64.898	0.024	849.325	195
78-93-3	Methyl ethyl ketone	301.218	2.000	6.321	379.595	2.965	0.253	0.181	0.000	692.533	141
7664-38-2	Phosphoric acid	0.031	57.550	0.000	0.375	183.755	255.000	63.088	0.000	559.799	177
67-64-1	Acetone	16.235	0.200	0.000	431.260	48.851	0.010	43.611	0.000	540.167	118
NA	Nickel (and its compounds)	4.220	183.751	0.000	0.000	193.905	121.513	3.529	0.000	506.918	114
108-05-4	Vinyl acetate	0.000	0.000	0.000	464.347	0.001	0.000	0.678	0.000	465.026	14
111-42-2	Diethanolamine	0.080	0.000	0.000	3.904	2.501	0.015	20.337	384.370	411.207	75
7664-41-7	Ammonia	0.000	211.400	20.400	21.271	1.376	0.000	138.607	0.000	393.054	161
108-95-2	Phenol	0.000	136.200	124.770	58.744	22.091	0.000	49.785	0.000	391.590	50
100-42-5	Styrene	0.000	1.700	0.002	254.314	54.084	0.000	0.528	0.065	310.694	78
80-62-6	Methyl methacrylate	0.000	0.000	0.000	258.100	5.200	0.000	0.006	0.000	263.306	13

CAS #	Substance Name	Physical	Chemical	Biological	Incineration	Landfill	Storage	MSTP*	Underground	Total Transfers	# of Reports
71-36-3	*n*-Butyl alcohol	25.075	3.000	15.200	185.152	7.074	0.603	13.530	0.000	249.634	79
7697-37-2	Nitric acid	0.000	133.245	0.000	6.600	16.125	0.000	65.870	10.000	231.840	101
50-00-0	Formaldehyde	0.000	0.000	1.345	164.220	41.993	0.000	4.893	0.000	212.451	80
7440-62-2	Vanadium (fume or dust)	0.000	181.640	0.000	0.000	25.512	0.000	1.790	0.000	208.942	11
108-10-1	Methyl isobutyl ketone	20.618	0.000	0.500	127.300	21.223	0.000	0.000	0.000	169.641	71
71-43-2	Benzene	0.000	0.000	0.000	124.286	0.948	0.000	0.078	26.277	151.589	95
106-99-0	1,3-Butadiene	0.000	0.000	0.000	128.687	2.100	0.000	0.524	0.000	131.311	13
110-82-7	Cyclohexane	0.000	0.000	0.000	116.903	0.154	0.000	0.000	0.000	117.057	82
NA	Selenium (and its compounds)	0.000	0.000	0.000	0.000	93.305	0.000	4.565	0.000	97.870	7
127-18-4	Tetrachloroethylene	28.830	0.000	0.000	26.097	39.000	0.000	0.002	0.000	93.929	29
1344-28-1	Aluminum oxide (fibrous forms)	0.000	0.000	0.340	9.560	46.920	0.000	0.000	0.000	56.820	14
7783-20-2	Ammonium sulphate (solution)	0.000	0.000	0.000	0.000	0.000	0.000	52.058	0.000	52.058	19
95-63-6	1,2,4-Trimethylbenzene	0.404	0.000	0.000	50.828	0.074	0.602	0.043	0.014	51.966	75
100-41-4	Ethylbenzene	0.020	0.000	0.000	33.891	1.058	0.039	0.015	1.149	36.171	68
75-09-2	Dichloromethane	0.156	15.390	0.000	19.759	0.002	0.000	0.002	0.000	35.309	50
85-68-7	Butyl benzyl phthalate	0.000	0.000	0.000	25.099	8.188	0.000	0.000	0.000	33.287	11
NA	Mercury (and its compounds)	0.000	0.000	0.000	0.000	32.633	0.000	0.000	0.000	32.633	5
101-68-8	Methylene*bis* (phenylisocyanate)	20.800	2.046	0.000	0.383	2.890	6.200	0.000	0.000	32.319	48
78-83-1	*i*-Butyl alcohol	0.000	0.000	0.000	28.173	2.910	0.000	0.000	0.050	31.133	25
6484-52-2	Ammonium nitrate (solution)	0.000	0.000	0.000	0.000	0.000	30.000	0.030	0.000	30.030	22
NA	Arsenic (and its compounds)	2.200	0.000	0.000	0.000	26.625	0.100	0.192	0.000	29.117	45
91-20-3	Naphthalene	0.000	0.017	0.000	3.292	25.768	0.000	0.007	0.000	29.084	40
156-62-7	Calcium cyanamide	0.000	28.740	0.000	0.000	0.000	0.000	0.290	0.000	29.030	1
79-01-6	Trichloroethylene	5.000	13.355	0.000	8.652	0.040	0.000	0.181	0.000	27.228	45
95-50-1	*o*-Dichlorobenzene	0.000	0.000	0.000	22.700	0.000	0.000	0.000	0.000	22.700	3
107-13-1	Acrylonitrile	0.000	0.000	0.000	18.930	0.000	0.000	0.000	0.000	18.930	8
73-35-4	Vinylidene chloride	0.000	0.000	0.000	0.000	11.400	0.000	0.000	0.000	11.400	1

CAS #	Substance Name	Physical	Chemical	Biological	Incineration	Landfill	Storage	MSTP*	Underground	Total Transfers	# of Reports
56-23-5	Carbon tetrachloride	0.000	0.000	0.000	11.107	0.000	0.000	0.000	0.000	11.107	6
1319-77-3	Cresol (mixed isomers)	0.000	0.000	0.000	9.200	0.000	0.000	0.000	0.000	9.200	4
7664-39-3	Hydrogen fluoride	0.000	7.950	0.000	0.000	0.000	0.000	0.000	0.000	7.950	19
103-23-1	*Bis*(2-ethylhexyl) adipate	0.000	0.000	0.000	4.545	2.471	0.003	0.000	0.000	7.019	14
NA	Antimony (and its compounds)	0.000	0.000	0.000	0.000	5.603	0.485	0.000	0.000	6.088	30
NA	Cobalt (and its compounds)	2.919	0.000	0.000	0.000	2.718	0.000	0.029	0.000	5.666	23
84-74-2	Dibutyl phthalate	0.000	0.004	0.000	5.137	0.460	0.000	0.000	0.000	5.601	17
117-84-0	Di-*n*-octyl phthalate	0.000	0.000	0.000	3.001	2.088	0.000	0.047	0.000	5.136	14
96-33-3	Methyl acrylate	0.000	0.000	0.000	4.900	0.000	0.000	0.000	0.000	4.900	2
85-44-9	Phthalic anhydride	0.000	0.000	0.000	2.330	1.424	0.001	0.000	0.000	3.755	15
120-12-7	Anthracene	0.000	0.000	0.000	0.000	3.466	0.000	0.000	0.000	3.466	12
92-52-4	Biphenyl	0.000	0.000	0.000	2.899	0.154	0.000	0.000	0.000	3.052	10
75-56-9	Propylene oxide	0.000	0.000	0.000	2.906	0.000	0.000	0.000	0.000	2.906	4
79-06-1	Acrylamide	0.000	0.000	0.000	0.000	2.845	0.000	0.000	0.000	2.845	10
26471-62-5	Toluene-diisocyanate (mixed isomers)	0.000	0.720	0.268	1.496	0.207	0.000	0.000	0.000	2.691	31
62-53-3	Aniline	0.000	0.000	2.610	0.023	0.000	0.000	0.000	0.000	2.633	2
NA	Cadmium (and its compounds)	0.804	0.000	0.000	0.000	1.635	0.000	0.045	0.000	2.484	18
108-31-6	Maleic anhydride	0.000	0.000	0.000	2.460	0.008	0.000	0.000	0.000	2.468	15
109-86-4	2-Methoxyethanol	0.000	0.000	0.000	2.121	0.000	0.000	0.000	0.000	2.121	4
1163-19-5	Decabromo-diphenyl oxide	0.000	0.000	0.000	0.000	1.995	0.001	0.000	0.000	1.996	3
584-84-9	Toluene-2,4-diisocyanate	0.000	0.000	0.000	1.590	0.000	0.000	0.000	0.000	1.590	3
108-38-3	*m*-Xylene	0.000	0.000	0.000	0.000	0.004	0.000	1.348	0.000	1.352	7
75-01-4	Vinyl chloride	0.000	0.000	0.000	0.130	0.827	0.000	0.000	0.000	0.957	10
75-21-8	Ethylene oxide	0.000	0.000	0.000	0.877	0.000	0.000	0.000	0.000	0.877	12
110-80-5	2-Ethoxyethanol	0.000	0.000	0.000	0.782	0.000	0.000	0.000	0.000	0.782	7
75-15-0	Carbon disulphide	0.000	0.000	0.000	0.350	0.230	0.000	0.000	0.100	0.680	7
1313-27-5	Molybdenum trioxide	0.000	0.223	0.000	0.000	0.280	0.000	0.169	0.000	0.672	15

CAS #	Substance Name	Physical	Chemical	Biological	Incineration	Landfill	Storage	MSTP*	Underground	Total Transfers	# of Reports
95-47-6	*o*-Xylene	0.000	0.000	0.000	0.019	0.007	0.000	0.526	0.000	0.552	7
139-13-9	Nitrilotriacetic acid	0.000	0.004	0.000	0.000	0.050	0.000	0.495	0.000	0.549	11
106-42-3	*p*-Xylene	0.000	0.000	0.000	0.000	0.004	0.000	0.516	0.000	0.520	4
106-46-7	*p*-Dichlorobenzene	0.000	0.000	0.000	0.000	0.500	0.000	0.000	0.000	0.500	5
7550-45-0	Titanium tetrachloride	0.000	0.000	0.000	0.442	0.018	0.000	0.000	0.000	0.460	4
107-06-2	1,2-Dichloroethane	0.000	0.000	0.000	0.220	0.000	0.000	0.000	0.000	0.220	4
74-85-1	Ethylene	0.000	0.000	0.000	0.000	0.000	0.000	0.160	0.000	0.160	43
141-32-2	Butyl acrylate	0.000	0.000	0.000	0.111	0.000	0.000	0.031	0.000	0.142	9
1634-04-4	Methyl *tert*-butyl ether	0.000	0.000	0.000	0.100	0.000	0.000	0.000	0.000	0.100	9
NA	Silver (and its compounds)	0.000	0.000	0.000	0.000	0.066	0.000	0.033	0.000	0.099	11
NA	Cyanides (ionic)	0.000	0.087	0.000	0.000	0.000	0.000	0.000	0.000	0.087	38
7782-50-5	Chlorine	0.000	0.000	0.000	0.000	0.000	0.000	0.065	0.000	0.065	175
75-05-8	Acetonitrile	0.000	0.000	0.000	0.064	0.000	0.000	0.000	0.000	0.064	3
534-52-1	4,6-Dinitro-*o*-cresol	0.000	0.000	0.037	0.000	0.000	0.000	0.000	0.000	0.037	1
123-31-9	Hydroquinone	0.000	0.000	0.000	0.017	0.000	0.000	0.006	0.000	0.023	3
79-10-7	Acrylic acid	0.000	0.000	0.000	0.021	0.000	0.000	0.000	0.000	0.021	9
98-82-8	Cumene	0.000	0.000	0.000	0.000	0.017	0.000	0.000	0.000	0.017	16
80-05-7	*p,p′*-Isopropylidene diphenol	0.000	0.000	0.000	0.000	0.014	0.000	0.000	0.000	0.014	2
75-65-0	*tert*-Butyl alcohol	0.000	0.000	0.000	0.000	0.010	0.000	0.000	0.000	0.010	2
67-66-3	Chloroform	0.000	0.000	0.000	0.008	0.000	0.000	0.000	0.000	0.008	3
111-15-9	2-Ethoxyethyl acetate	0.000	0.000	0.000	0.002	0.000	0.000	0.000	0.000	0.002	4
75-07-0	Acetaldehyde	0.000	0.000	0.000	0.000	0.001	0.000	0.000	0.000	0.001	5
	Total	**9,366.123**	**4,401.869**	**3,597.841**	**7,049.850**	**34,351.959**	**1,247.636**	**2,016.220**	**2,270.349**	**64,301.847**	**5,786**

* Municipal sewage treatment plant.

Appendix 9: Anticipated transfers in 1994 (tonnes)

CAS #	Substance Name	Reported 1994	Anticipated 1995	Anticipated 1996	Anticipated 1997	# of Reports
117-81-7	*Bis*(2-ethylhexyl) phthalate	14,891.139	7,383.282	7,744.311	8,127.039	33
7429-90-5	Aluminum (fume or dust)	7,766.402	7,773.203	7,769.403	7,770.303	35
NA	Copper (and its compounds)	7,038.419	7,426.786	7,394.882	7,391.904	243
NA	Zinc (and its compounds)	5,286.846	7,294.211	6,991.567	7,105.072	283
67-56-1	Methanol	4,395.865	4,348.491	4,220.336	4,162.771	284
NA	Manganese (and its compounds)	3,791.774	3,972.976	3,989.268	4,036.078	192
7664-93-9	Sulfuric acid	2,723.174	5,250.951	5,047.393	5,047.903	387
108-88-3	Toluene	1,917.749	2,090.693	2,042.001	1,965.132	313
107-21-1	Ethylene glycol	1,659.420	1,049.766	999.990	987.329	232
1330-20-7	Xylene (mixed isomers)	1,439.323	1,479.167	1,385.135	1,336.726	285
7440-66-6	Zinc (fume or dust)	1,432.288	1,410.502	1,414.558	1,330.558	36
NA	Lead (and its compounds)	1,301.650	1,492.703	1,444.971	1,391.900	160
7647-01-0	Hydrochloric acid	1,009.346	987.275	872.275	877.375	211
NA	Chromium (and its compounds)	1,002.646	1,031.374	887.663	800.124	189
1332-21-4	Asbestos	935.063	1,147.136	850.819	755.625	37
67-63-0	Isopropyl alcohol	849.325	779.929	741.555	744.371	195
78-93-3	Methyl ethyl ketone	692.533	623.174	633.857	623.986	141
7664-38-2	Phosphoric acid	559.799	502.368	496.678	494.053	177
67-64-1	Acetone	540.167	398.755	363.591	354.327	118
NA	Nickel (and its compounds)	506.918	558.001	505.214	448.592	114
108-05-4	Vinyl acetate	465.026	401.551	401.151	401.051	14
111-42-2	Diethanolamine	411.207	372.741	372.853	372.978	75
7664-41-7	Ammonia	393.054	360.819	380.819	402.819	161
108-95-2	Phenol	391.590	231.801	228.103	228.606	50
100-42-5	Styrene	310.694	306.770	220.213	219.570	78
80-62-6	Methyl methacrylate	263.306	78.020	63.840	59.860	13
71-36-3	*n*-Butyl alcohol	249.634	237.669	229.335	198.472	79
7697-37-2	Nitric acid	231.840	206.622	151.122	150.722	101
50-00-0	Formaldehyde	212.451	105.004	105.826	101.167	80
7440-62-2	Vanadium (fume or dust)	208.942	203.400	203.400	203.400	11
108-10-1	Methyl isobutyl ketone	169.641	171.643	168.627	171.291	71
71-43-2	Benzene	151.589	159.322	112.332	76.342	95
106-99-0	1,3-Butadiene	131.311	87.500	12.600	12.210	13
110-82-7	Cyclohexane	117.057	117.987	117.988	117.989	82
NA	Selenium (and its compounds)	97.870	27.800	27.800	27.800	7
127-18-4	Tetrachloroethylene	93.929	98.394	98.394	99.193	29
1344-28-1	Aluminum oxide (fibrous forms)	59.820	54.630	54.430	54.230	14
7783-20-2	Ammonium sulphate (solution)	52.058	56.000	56.000	56.000	19
95-63-6	1,2,4-Trimethylbenzene	51.966	53.514	53.424	53.423	75
100-41-4	Ethylbenzene	36.171	46.750	47.037	46.831	68
75-09-2	Dichloromethane	35.309	49.897	57.345	63.848	50
85-68-7	Butyl benzyl phthalate	33.287	21.111	21.111	21.111	11
NA	Mercury (and its compounds)	32.633	16.000	16.000	6.100	5
101-68-8	Methylene*bis*(phenylisocyanate)	32.319	27.038	26.398	25.908	48
78-83-1	*i*-Butyl alcohol	31.133	23.574	24.863	14.959	25
6484-52-2	Ammonium nitrate (solution)	30.030	30.030	0.030	0.030	22
NA	Arsenic (and its compounds)	29.117	27.852	25.485	25.104	45
91-20-3	Naphthalene	29.084	18.136	11.336	10.136	40
156-62-7	Calcium cyanamide	29.030	30.000	30.000	30.000	1
79-01-6	Trichloroethylene	27.228	31.103	30.627	24.201	45
95-50-1	*o*-Dichlorobenzene	22.700	11.350	11.350	11.350	3
107-13-1	Acrylonitrile	18.930	8.000	2.500	2.300	8
75-35-4	Vinylidene chloride	11.400	0.000	0.000	0.000	1
56-23-5	Carbon tetrachloride	11.107	8.900	6.000	6.000	6
1319-77-3	Cresol (mixed isomers)	9.200	4.640	4.640	4.640	4
7664-39-3	Hydrogen fluoride	7.950	5.100	0.100	0.050	19
103-23-1	*Bis*(2-ethylhexyl) adipate	7.019	8.074	8.174	5.075	14
NA	Antimony (and its compounds)	6.088	4.488	3.671	3.551	30
NA	Cobalt (and its compounds)	5.666	5.496	5.358	5.158	23
84-74-2	Dibutyl phthalate	5.601	5.541	5.702	5.866	17

CAS #	Substance Name	Reported 1994	Anticipated 1995	Anticipated 1996	Anticipated 1997	# of Reports
117-84-0	Di-*n*-octyl phthalate	5.136	6.423	6.667	6.935	14
96-33-3	Methyl acrylate	4.900	3.400	3.200	3.000	2
85-44-9	Phthalic anhydride	3.755	1.215	1.204	1.293	15
120-12-7	Anthracene	3.466	1.866	0.066	0.066	12
92-52-4	Biphenyl	3.052	2.802	2.802	2.802	10
75-56-9	Propylene oxide	2.906	2.750	2.500	0.000	4
79-06-1	Acrylamide	2.845	2.845	2.845	2.845	10
26471-62-5	Toluenediisocyanate (mixed isomers)	2.691	2.321	2.401	2.501	31
62-53-3	Aniline	2.633	3.140	3.140	3.140	2
NA	Cadmium (and its compounds)	2.484	0.921	0.789	0.685	18
108-31-6	Maleic anhydride	2.468	2.110	2.110	2.110	15
109-86-4	2-Methoxyethanol	2.121	1.253	1.253	1.253	4
1163-19-5	Decabromodiphenyl oxide	1.996	1.075	0.157	0.157	3
584-84-9	Toluene-2,4-diisocyanate	1.590	0.150	0.150	0.170	3
108-38-3	*m*-Xylene	1.352	5.404	5.404	5.404	7
75-01-4	Vinyl chloride	0.957	0.830	0.830	0.830	10
75-21-8	Ethylene oxide	0.877	15.000	9.500	0.000	12
110-80-5	2-Ethoxyethanol	0.782	1.638	1.638	1.638	7
75-15-0	Carbon disulphide	0.680	1.100	0.100	0.100	7
1313-27-5	Molybdenum trioxide	0.672	0.700	0.700	0.710	15
95-47-6	*o*-Xylene	0.552	2.125	2.123	2.123	7
139-13-9	Nitrilotriacetic acid	0.549	0.100	0.100	0.100	11
106-42-3	*p*-Xylene	0.520	2.104	2.104	2.104	4
106-46-7	*p*-Dichlorobenzene	0.500	0.000	0.000	0.000	5
7550-45-0	Titanium tetrachloride	0.460	0.270	0.310	0.230	4
107-06-2	1,2-Dichloroethane	0.220	0.200	0.200	0.200	4
74-85-1	Ethylene	0.160	0.160	0.160	0.160	43
141-32-2	Butyl acrylate	0.142	0.200	0.200	0.200	9
1634-04-4	Methyl *tert*-butyl ether	0.100	0.100	0.100	0.100	9
NA	Silver (and its compounds)	0.099	0.036	0.030	0.030	11
NA	Cyanides (ionic)	0.087	0.100	0.100	0.100	38
7782-50-5	Chlorine	0.065	0.695	0.695	0.695	175
75-05-8	Acetonitrile	0.064	1.064	1.064	1.064	3
534-52-1	4,6-Dinitro-*o*-cresol	0.037	0.037	0.037	0.037	1
123-31-9	Hydroquinone	0.023	0.024	0.024	0.024	3
79-10-7	Acrylic acid	0.021	0.001	0.001	0.001	9
98-82-8	Cumene	0.017	0.017	0.017	0.017	16
80-05-7	*p*,*p*′-Isopropylidenediphenol	0.014	0.020	0.040	0.050	2
75-65-0	*tert*-Butyl alcohol	0.010	0.010	0.010	0.010	2
67-66-3	Chloroform	0.008	0.008	0.008	0.008	3
111-15-9	2-Ethoxyethyl acetate	0.002	0.002	0.002	0.002	4
75-07-0	Acetaldehyde	0.001	0.001	0.001	0.001	5
	Total	**64,301.847**	**60,779.227**	**59,246.233**	**59,113.404**	**5,786**

Appendix 10: Substances sent for the "3Rs" and energy recovery (tonnes)

CAS #	Substance Name	"3Rs"	Energy Recovery	Total	# of Reports
7664-93-9	Sulfuric acid	65,776.565	0.000	65,776.565	20
7647-01-0	Hydrochloric acid	41,922.140	0.000	41,922.140	9
NA	Manganese (and its compounds)	36,706.039	0.669	36,706.708	57
67-63-0	Isopropyl alcohol	33,784.297	696.065	34,480.362	50
117-84-0	Di-*n*-octyl phthalate	20,013.426	52.900	20,066.326	6
NA	Copper (and its compounds)	17,513.498	0.643	17,514.141	101
NA	Lead (and its compounds)	13,959.820	1.144	13,960.964	61

CAS #	Substance Name	"3Rs"	Energy Recovery	Total	# of Reports
115-07-1	Propylene	13,856.940	0.000	13,856.940	2
NA	Zinc (and its compounds)	13,594.043	93.108	13,687.151	88
108-88-3	Toluene	5,891.989	1,239.520	7,131.509	94
1330-20-7	Xylene (mixed isomers)	5,412.974	972.188	6,385.162	83
78-93-3	Methyl ethyl ketone	3,336.456	440.351	3,776.807	48
NA	Chromium (and its compounds)	3,445.765	0.000	3,445.765	45
NA	Nickel (and its compounds)	2,995.830	0.000	2,995.830	41
NA	Cobalt (and its compounds)	1,190.480	2.358	1,192.838	4
67-64-1	Acetone	656.112	386.565	1,042.677	34
107-21-1	Ethylene glycol	700.670	113.045	813.715	32
67-56-1	Methanol	324.031	447.362	771.393	35
7440-62-2	Vanadium (fume or dust)	712.810	0.000	712.810	4
7664-38-2	Phosphoric acid	603.566	0.000	603.566	7
7429-90-5	Aluminum (fume or dust)	590.819	4.410	595.229	12
NA	Arsenic (and its compounds)	515.607	0.000	515.607	2
108-10-1	Methyl isobutyl ketone	194.649	120.276	314.925	27
7697-37-2	Nitric acid	297.832	0.000	297.832	7
127-18-4	Tetrachloroethylene	236.241	30.340	266.581	8
NA	Antimony (and its compounds)	224.008	0.000	224.008	10
79-01-6	Trichloroethylene	188.980	10.738	199.718	24
100-42-5	Styrene	0.696	182.300	182.996	8
100-41-4	Ethylbenzene	151.512	27.442	178.954	10
71-36-3	*n*-Butyl alcohol	59.243	116.577	175.820	26
1313-27-5	Molybdenum trioxide	171.092	0.000	171.092	2
75-09-2	Dichloromethane	160.145	5.050	165.195	10
7440-66-6	Zinc (fume or dust)	110.780	0.000	110.780	6
NA	Cadmium (and its compounds)	102.937	0.650	103.587	4
95-63-6	1,2,4-Trimethylbenzene	62.242	28.311	90.553	8
7664-41-7	Ammonia	85.272	0.000	85.272	3
1344-28-1	Aluminum oxide (fibrous forms)	66.940	0.000	66.940	2
111-42-2	Diethanolamine	46.700	16.504	63.204	4
110-82-7	Cyclohexane	61.474	0.059	61.533	4
NA	Selenium (and its compounds)	58.770	0.000	58.770	1
117-81-7	*Bis*(2-ethylhexyl) phthalate	48.772	0.000	48.772	4
108-38-3	*m*-Xylene	14.666	5.424	20.090	1
108-05-4	Vinyl acetate	0.000	20.000	20.000	1
108-95-2	Phenol	15.246	1.573	16.819	5
101-68-8	Methylene*bis*(phenylisocyanate)	7.870	0.000	7.870	3
78-83-1	*i*-Butyl alcohol	4.724	2.380	7.104	3
71-43-2	Benzene	6.800	0.003	6.803	2
84-74-2	Dibutyl phthalate	0.304	3.656	3.960	3
85-68-7	Butyl benzyl phthalate	1.392	2.290	3.682	4
75-56-9	Propylene oxide	0.000	2.906	2.906	1
50-00-0	Formaldehyde	1.668	0.095	1.763	5
1319-77-3	Cresol (mixed isomers)	1.620	0.000	1.620	2
75-21-8	Ethylene oxide	0.000	0.877	0.877	1
98-82-8	Cumene	0.000	0.500	0.500	1
26471-62-5	Toluenediisocyanate (mixed isomers)	0.500	0.000	0.500	1
141-32-2	Butyl acrylate	0.000	0.500	0.500	1
110-80-5	2-Ethoxyethanol	0.369	0.008	0.377	2
140-88-5	Ethyl acrylate	0.000	0.150	0.150	1
1634-04-4	Methyl *tert*-butyl ether	0.100	0.000	0.100	1
80-62-6	Methyl methacrylate	0.000	0.100	0.100	1
109-86-4	2-Methoxyethanol	0.000	0.071	0.071	1
78-92-2	*sec*-Butyl alcohol	0.000	0.057	0.057	1
NA	Silver (and its compounds)	0.008	0.000	0.008	1
	Total	**285,887.429**	**5,029.165**	**290,916.594**	**1,045**

Appendix 11: Anticipated "3Rs" and energy recovery (tonnes)

CAS #	Substance Name	Reported 1994	Anticipated 1995	Anticipated 1996	Anticipated 1997	# of Reports
7664-93-9	Sulfuric acid	65,776.565	57,696.771	55,258.803	51,798.853	20
7647-01-0	Hydrochloric acid	41,922.140	41,069.500	41,073.000	41,068.000	9
NA	Manganese (and its compounds)	36,706.708	39,708.548	42,853.007	46,650.335	57
67-63-0	Isopropyl alcohol	34,480.362	21,183.130	20,996.355	20,998.630	50
117-84-0	Di-*n*-octyl phthalate	20,066.326	20,073.766	20,073.966	20,074.166	6
NA	Copper (and its compounds)	17,514.141	17,108.145	17,235.073	17,204.156	101
NA	Lead (and its compounds)	13,960.964	4,250.265	3,641.740	3,605.310	61
115-07-1	Propylene	13,856.940	14,560.000	14,560.000	14,560.000	2
NA	Zinc (and its compounds)	13,687.151	13,612.899	13,815.493	13,885.835	88
108-88-3	Toluene	7,131.509	6,270.499	4,773.476	4,745.719	94
1330-20-7	Xylene (mixed isomers)	6,385.162	4,961.039	4,508.673	4,315.652	83
78-93-3	Methyl ethyl ketone	3,776.807	3,178.159	2,052.329	2,027.049	48
NA	Chromium (and its compounds)	3,445.765	3,281.401	3,332.946	2,264.866	45
NA	Nickel (and its compounds)	2,995.830	2,882.309	2,914.951	2,942.554	41
NA	Cobalt (and its compounds)	1,192.838	1,175.000	1,175.000	1,175.000	4
67-64-1	Acetone	1,042.677	865.275	859.342	865.191	34
107-21-1	Ethylene glycol	813.715	719.572	682.096	671.216	32
67-56-1	Methanol	771.393	750.190	660.610	661.556	35
7440-62-2	Vanadium (fume or dust)	712.810	600.000	600.000	600.000	4
7664-38-2	Phosphoric acid	603.566	253.876	253.889	158.901	7
7429-90-5	Aluminum (fume or dust)	595.229	594.189	598.869	604.749	12
NA	Arsenic (and its compounds)	515.607	515.000	515.000	515.000	2
108-10-1	Methyl isobutyl ketone	314.925	224.780	205.720	202.790	27
7697-37-2	Nitric acid	297.832	249.000	249.000	249.000	7
127-18-4	Tetrachloroethylene	266.581	223.801	160.111	113.201	8
NA	Antimony (and its compounds)	224.008	220.718	223.014	221.864	10
79-01-6	Trichloroethylene	199.718	152.326	113.737	69.079	24
100-42-5	Styrene	182.996	200.620	201.220	200.420	8
100-41-4	Ethylbenzene	178.954	86.900	84.400	81.200	10
71-36-3	*n*-Butyl alcohol	175.820	173.060	168.870	166.670	26
1313-27-5	Molybdenum trioxide	171.092	170.000	170.000	170.000	2
75-09-2	Dichloromethane	165.195	172.050	177.000	183.000	10
7440-66-6	Zinc (fume or dust)	110.780	65.100	70.100	70.100	6
NA	Cadmium (and its compounds)	103.587	101.730	101.730	101.730	4
95-63-6	1,2,4-Trimethylbenzene	90.553	93.500	93.500	91.100	8
7664-41-7	Ammonia	85.272	84.870	81.370	77.870	3
1344-28-1	Aluminum oxide (fibrous forms)	66.940	0.000	0.000	0.000	2
111-42-2	Diethanolamine	63.204	56.500	56.500	56.500	4
110-82-7	Cyclohexane	61.533	73.000	73.000	73.000	4
NA	Selenium (and its compounds)	58.770	58.000	58.000	58.000	1
117-81-7	*Bis*(2-ethylhexyl) phthalate	48.772	65.673	65.673	65.673	4
108-38-3	*m*-Xylene	20.090	14.750	14.750	14.750	1
108-05-4	Vinyl acetate	20.000	10.000	5.000	5.000	1
108-95-2	Phenol	16.819	22.848	15.779	15.417	5
101-68-8	Methylene*bis*(phenylisocyanate)	7.870	4.950	4.950	4.950	3
78-83-1	*i*-Butyl alcohol	7.104	3.569	2.409	2.409	3
71-43-2	Benzene	6.803	7.000	7.000	7.000	2
84-74-2	Dibutyl phthalate	3.960	0.304	0.334	0.364	3
85-68-7	Butyl benzyl phthalate	3.682	3.641	3.641	3.641	4
75-56-9	Propylene oxide	2.906	0.000	0.000	0.000	1
50-00-0	Formaldehyde	1.763	4.723	1.336	1.392	5
1319-77-3	Cresol (mixed isomers)	1.620	1.500	1.500	1.500	2
75-21-8	Ethylene oxide	0.877	0.000	0.000	0.000	1
141-32-2	Butyl acrylate	0.500	0.000	0.000	0.000	1
98-82-8	Cumene	0.500	0.500	0.500	0.500	1
26471-62-5	Toluenediisocyanate (mixed isomers)	0.500	0.000	0.000	0.000	1
110-80-5	2-Ethoxyethanol	0.377	0.000	0.000	0.000	2
140-88-5	Ethyl acrylate	0.150	0.000	0.000	0.000	1
1634-04-4	Methyl *tert*-butyl ether	0.100	0.000	0.000	0.000	1

CAS #	Substance Name	Reported 1994	Anticipated 1995	Anticipated 1996	Anticipated 1997	# of Reports
80-62-6	Methyl methacrylate	0.100	0.000	0.000	0.000	1
109-86-4	2-Methoxyethanol	0.071	0.000	0.000	0.000	1
78-92-2	*sec*-Butyl alcohol	0.057	0.000	0.000	0.000	1
NA	Silver (and its compounds)	0.008	0.000	0.000	0.000	1
	Total	**290,916.594**	**257,854.946**	**254,845.762**	**253,702.858**	**1,045**

Air emissions

Air emissions data are presented in several formats. Tables P-2 and P-3 are examples of how air criteria contamination data for specific provinces are tabulated.

Data from Environment Canada indicate that the annual average level of airborne particles at all NAPS monitoring sites decreased by 46 percent between 1974 and 1986. This significant improvement in urban air quality relative to particulate emissions reductions reflects the fact that most major industrial sources of particulate matter are located outside urban areas. The average levels at more than 90 percent of the sites were well within the desirable range of air quality in 1986 (Fig. P-2).

Short-term exposure to high levels of particulate matter continues to be a problem, however. For example, the maximum acceptable level objective for a 24-hour exposure period was exceeded at least 10 percent of the time at some NAPS sites in Sydney, Rouyn, Windsor, Edmonton, Hamilton, and Calgary. As well, the 24-hour maximum tolerable level ambient air quality objective was exceeded once in 1986 at NAPS monitoring sites in each of Edmonton, Calgary, and Yellowknife. High short-term particulate levels can be attributed in part to natural windblown dust, as well as to construction and industrial activity and the increasing number of motor vehicles on city streets. Over the long term, economic and social development—which has led to the use of cleaner energy sources (e.g., natural gas instead of coal and wood), modernization of older city centers, cleaner streets, more grass and asphalt cover, and the upgrading of industrial and commercial facilities, including installation of equipment to control pollution—appears to be leading to better air quality in urban areas with respect to particulate matter.

Environment Canada has collected inhalable particles (<10 μm in diameter) in 15 Canadian cities since May 1984. Results for the period from May 1984 to December 1987 show that average inhalable particulate levels ranged from 17 $\mu g/m^3$ in St. John's to 49 $\mu g/m^3$ at a site in Montreal. Analysis has shown that the finest of the inhalable particles—those with a diameter less than about 2.5 μm—are different in origin and composition from the coarser particles in the 2.5- to 10-μm-diameter range. The coarse particles are mostly of natural origin (minerals from Earth's crust, sea salt, and plant material), whereas the fine particles consist of lead, sulfates, nitrates, carbon, and a variety of organic compounds, mainly resulting from man-made pollution. At eastern Canadian sites, fine particulate matter accounted for more than 60 percent of the inhalable particles; at sites in the Prairie provinces, the fine fraction was usually less than 40 percent of the inhalable particles.

The levels of inhalable particles measured in Canada are below the recently introduced U.S. air quality standards for this size of particle, set at 150 $\mu g/m^3$ for exposure over one hour and 50 $\mu g/m^3$ for exposure over one year. Canadian air quality objectives for inhalable particulate matter are being developed.

TABLE P-2 Case Study: 1990 Emissions of Criteria Contaminants for Ontario

	Emissions in Tonnes				
Category/Sector	Part	SO_2	NO_x	VOC	CO
Industrial Processes					
Abrasives manufacture	126	21	10	42	1
Aluminum production	235	5	49	5	1
Asphalt production	11,255	255	131	2,176	61
Bakeries			4	1,374	
Cement and concrete manufacture	5,422	12,373	9,603	155	503
Clay products	4,900	2	98	13	31
Crude oil production				46	
Grain industries	12,178				
Iron and steel production	22,562	48,125	21,719	25,275	548,265
Mining and rock quarrying	53,550	897	946	73	455
Nonferrous mining and smelting	6,264	692,302	52,668	277	26
Other chemicals	6,185	834	1,233	401	1,018
Paint & varnish manufacturing	40	1	9	912	2
Petrochemical industry	1,033	1,012	3,397	11,802	9,529
Petroleum refining	3,226	63,687	14,360	35,047	5,020
Plastics & synthetic resins fabrication	128	296	316	5,964	814
Pulp and paper industry	22,024	29,398	9,632	7,913	22,158
Wood industry	8,680	279	2,073	6,557	522
Other industries*	80,635	75,051	35,023	57,455	27,069
Category total	**238,440**	**924,539**	**151,269**	**155,489**	**615,474**
Fuel Combustion					
Commercial fuel combustion	517	4,050	8,323	317	1,657
Electric power generation	8,208	195,101	77,235	396	3,353
Residential fuel combustion	467	8,146	13,800	952	5,305
Residential fuelwood combustion	28,779	348	2,342	114,664	206,411
Category total	**37,971**	**207,645**	**101,700**	**116,329**	**216,725**
Transportation					
Aircraft	328	409	5,527	2,157	13,116
Heavy duty diesel vehicles	9,336	14,646	116,814	13,164	52,846
Heavy duty gasoline trucks	45	37	3,241	3,159	41,696
Light duty diesel trucks	35	1,146	789	324	622
Light duty diesel vehicles	31	1,012	605	207	490
Light duty gasoline trucks	392	1,134	32,226	44,150	422,853
Light duty gasoline vehicles	1,366	3,094	113,666	165,492	1,497,666
Marine	1,737	19,316	10,526	12,628	38,406
Motor cycles	6	7	217	1,358	3,612
Off-road use of diesel	5,785	6,970	70,651	6,335	21,296
Off-road use of gasoline	1,728	336	10,957	34,812	388,211
Propane powered vehicles	101	7	2,803	1,674	2,601
Railroads	1,783	2,663	33,042	1,604	13,192
Tire wear	15,088			305	
Category total	**37,759**	**50,775**	**401,065**	**287,369**	**2,496,607**
Incineration					
Other incineration	52	750	962	1,104	2,012
Wood waste incineration	2,034	35	351	3,856	45,565
Category total	**2,086**	**785**	**1,313**	**4,959**	**47,577**
Miscellaneous					
Application of surface coatings	16		4	121,628	
Cigarette smoking	1,203				480
Dry cleaning			1	5,933	
Fuel marketing				35,309	
General solvent use	4		9	124,225	2
Pesticides and fertilizer application	4,883		1	9,482	
Structural fires	2,062		424	2,062	4,123
Category total	**8,168**		**440**	**298,639**	**4,606**
TOTAL FOR ONTARIO	**324,424**	**1,183,745**	**655,786**	**862,784**	**3,380,990**

NOTE: Due to rounding, totals may not add exactly.

* Industrial sectors such as Coal Industry, Iron Ore Mining and Beneficiation, Carbon Black, Ferrous Foundries, and Other Petroleum and Coal Products were included in the Other Industries sector to protect the confidentiality of the information.

TABLE P-3 Case Study: 1990 Emissions of Criteria Contaminants for Quebec

Category/Sector	Emissions in Tonnes				
	Part	SO_2	NO_x	VOC	CO
Industrial Processes					
Aluminum production	14,790	25,669	602	691	240,024
Asbestos production	88	963	220	1	20
Asphalt production	1,629				
Bakeries				1,094	
Cement and concrete manufacture	3,829	14,061	3,941	13	825
Coal industry	1,729		156	2,041	2,236
Crude oil production				1,093	
Ferrous foundries	1,027	1,846	15	1,053	515
Grain industries	4,346				
Iron and steel production	24,547	19,477	5,465	1,142	21,313
Mining and rock quarrying	33,266				
Nonferrous mining and smelting	2,306	194,158	561	7	119
Other chemicals	1,945	11,470	2,985	185	232
Petrochemical industry	105	862	1,238	6,835	247
Petroleum refining	893	15,630	5,059	20,100	5,217
Plastics & synthetic resins fabrication	44	127	29	4,520	4
Pulp and paper industry	21,736	41,945	17,061	2,449	13,731
Wood industry	13,034	104	51	22	37
Other industries	18,662	7,244	5,218	3,130	4,466
Category total	**143,976**	**333,556**	**42,600**	**44,377**	**288,986**
Fuel combustion					
Commercial fuel combustion	354	4,271	3,930	145	800
Electric power generation	932	11,313	3,527	39	291
Residential fuel combustion	624	8,088	5,237	207	1,318
Residential fuelwood combustion	63,377	515	1,393	51,188	274,476
Category total	**65,287**	**24,186**	**14,087**	**51,579**	**276,884**
Transportation					
Aircraft	220	291	4,302	1,176	8,715
Heavy duty diesel vehicles	5,765	7,753	72,139	8,130	32,635
Heavy duty gasoline trucks	26	39	1,917	1,846	24,696
Light duty diesel trucks	16	464	364	147	285
Light duty diesel vehicles	37	1,041	722	243	583
Light duty gasoline trucks	185	959	15,095	20,346	196,767
Light duty gasoline vehicles	883	3,376	72,241	103,971	954,911
Marine	1,897	17,338	10,397	4,777	13,520
Motor cycles	3	6	117	670	1,950
Off-road use of diesel	3,881	4,087	48,846	4,198	14,365
Off-road use of gasoline	632	302	6,051	15,177	206,968
Railroads	569	731	10,542	512	4,209
Tire wear	9,500			192	
Category total	**23,615**	**36,387**	**242,731**	**161,385**	**1,459,604**
Incineration					
Other incineration	91	915	624	432	5,763
Wood waste incineration	4,355	40	396	4,355	51,466
Category total	**4,446**	**955**	**1,020**	**4,787**	**57,229**
Miscellaneous					
Application of surface coatings				40,356	
Cigarette smoking	892				356
Dry cleaning				2,548	
Fuel marketing				17,765	
General solvent use				77,112	
Structural fires	98		36	127	1,530
Category total	**990**		**36**	**137,908**	**1,885**
TOTAL FOR QUEBEC	**238,314**	**395,084**	**300,476**	**400,036**	**2,084,588**

NOTE: Due to rounding, totals may not add exactly.

* Industrial sectors such as Abrasives Manufacture were included in the Other Industries sector to protect the confidentiality of the information.

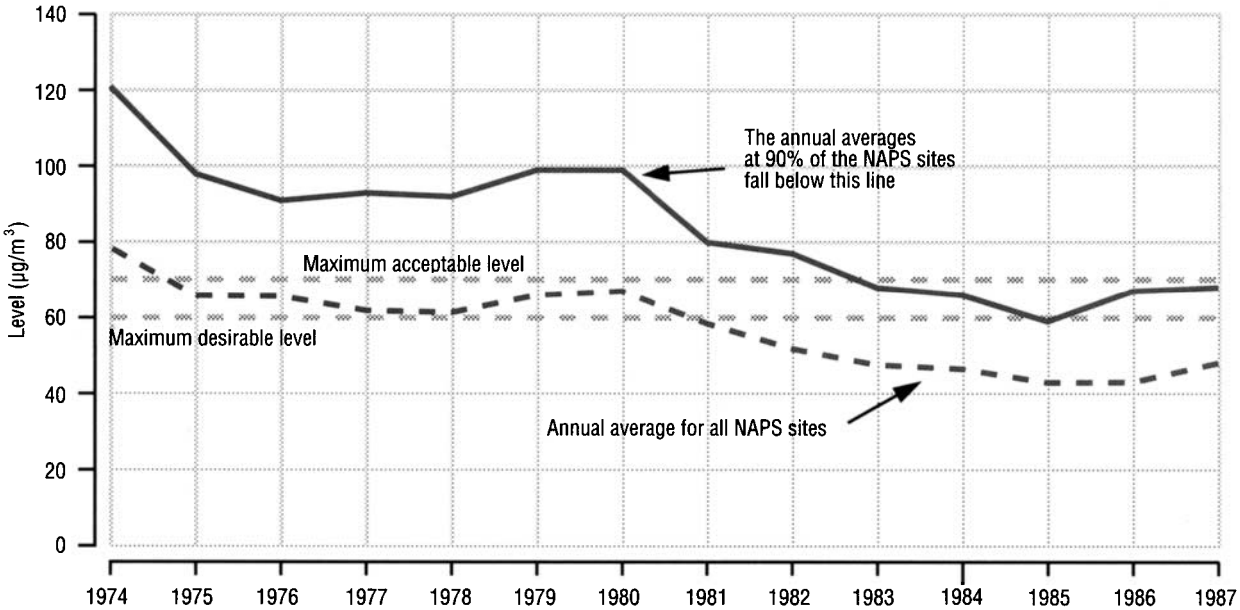


FIG. P-2 Trends in ambient air levels of total suspended particulate matter in Canada, 1974–1987. Note: Particulate matter is collected over a 24-hour period on every sixth day throughout the year. (Source: Environment Canada.)

Sulfur dioxide

About 70 percent of total sulfur dioxide emissions in 1985 in Canada came from industrial processes—69 percent of this was from copper, nickel, lead, zinc, gold, and aluminum production, and a further 21 percent was from oil and natural gas recovery and processing. Fuel combustion, mainly by power plants and industries, accounted for 28 percent of total Canadian emissions of sulfur dioxide. Between 1970 and 1985, sulfur dioxide emissions declined by almost 45 percent (Fig. P-3), mainly because of modifications to industrial processes and technology, the capture and use of sulfur dioxide to make commercially useful sulfuric acid, and the increased use of low-sulfur fuels.

Environmental levels of sulfur dioxide can adversely affect both human health and vegetation. In epidemiological and laboratory-controlled human health studies, effects on the lung and the induction of chronic lung disease have been recorded. Although no clear threshold has been identified, short-term exposures to sulfur dioxide at concentrations of up to 1 ppm have not induced severe or irreversible effects; however, significant reductions in lung function have been observed in healthy exercising adults after exposure to this level. Mild respiratory symptoms related to airway dysfunction and transient bronchoconstriction have also been observed in exercising asthmatic subjects.

In epidemiological studies, short-term exposures to sulfur dioxide that lasted a day or so have been correlated with deaths, although there was concomitant exposure to high particulate levels during these pollution episodes. Long-term or chronic exposures to levels of up to 50 ppb of sulfur dioxide–induced respiratory symptoms and disease (coughs and bronchitis), especially in young children and smokers.

The earliest sign of injury to vegetation is damage to foliage; other plant parts appear to be more resistant. The eastern white pine is a particularly sensitive

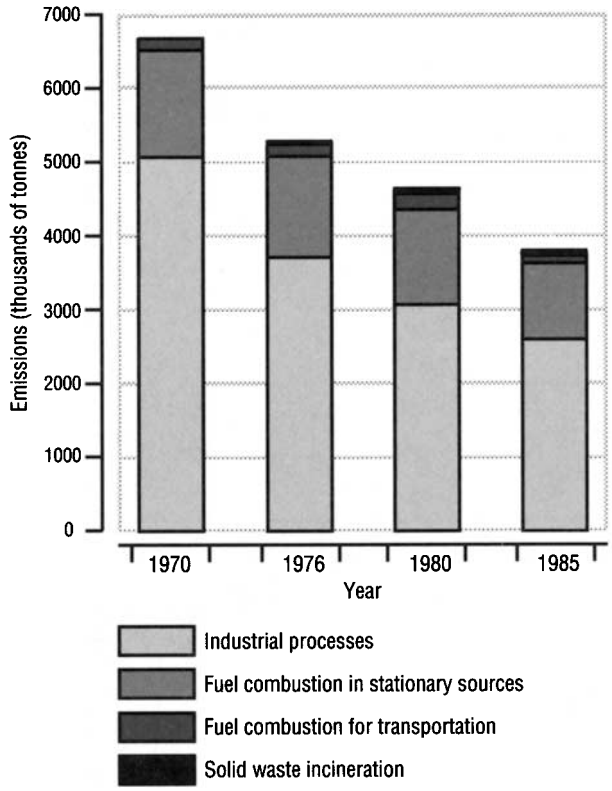


FIG. P-3 Sulfur dioxide emissions in Canada, 1970–1985. (Source: Environment Canada.)

species, showing signs of acute injury in a matter of hours at a sulfur dioxide concentration in air as low as 25–30 ppb. Many plant species are damaged within hours when exposed to a sulfur dioxide concentration between 100 and 1000 ppb, whereas some hardier plant species show acute damage only above 1 ppm.

Long-term injury to vegetation, usually observed as a yellowing of foliage, is generally assessed in terms of the average concentration of sulfur dioxide to which a plant is exposed over the growing season or over a one-year period, because sulfur dioxide concentrations can be quite variable from day to day. The response observed in plants is also strongly influenced by such environmental factors as sun, rain, wind, and drought. Studies of chronic forest damage suggest that effects are prominent when the average sulfur dioxide concentration is about 17 ppb but slight when the concentration is 8 ppb. Lichen species diversity and abundance are affected at 15–20 ppb.

The recommended air quality objectives were based on both human health effects and effects on vegetation. The maximum acceptable limits for a one-hour, 24-hour, and annual average were 340, 110, and 20 ppb, respectively; maximum desirable limits were 170, 60, and 10 ppb for the same time periods. These levels were retained following a review of the more recent literature.

In urban areas of Canada, the annual average level of sulfur dioxide measured at NAPS monitoring sites decreased by 54 percent between 1974 and 1986—from 13 ppb to 6 ppb—and the annual average levels at 90 percent of the NAPS monitoring sites are now well below the maximum desirable level annual air quality objective of 10 ppb (Fig. P-4). In 1986, the one-hour maximum desirable level

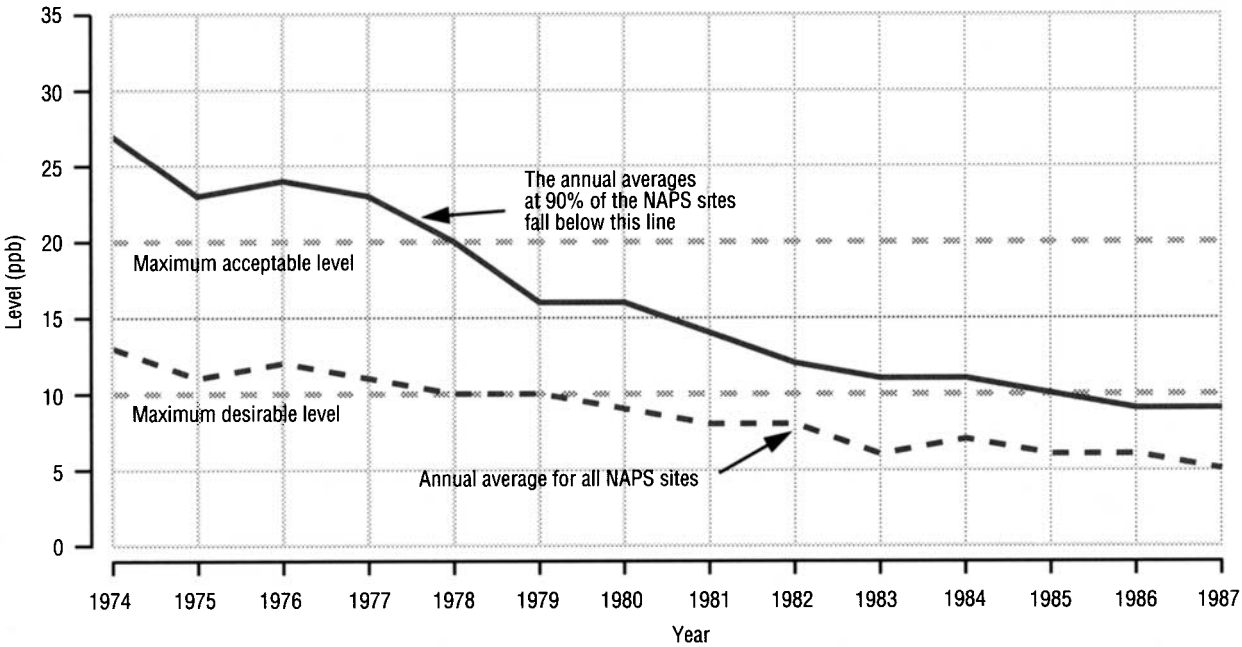


FIG. P-4 Trends in ambient air levels of sulfur dioxide in Canada, 1974–1987. (Source: Environment Canada.)

TABLE P-4 Air Quality Objectives for Sulfur Dioxide

Exposure Period	Maximum Desirable Concentration (ppb)	Maximum Acceptable Concentration (ppb)	Maximum Tolerable Concentration (ppb)
One hour	170	340	—
24 hours	60	110	310
One year	10	20	

objective of 170 ppb was met or bettered 99.9 percent of the time at 93 percent of the NAPS monitoring sites. Monitoring sites in Montreal, Quebec City, Rouyn, Trois-Rivières, Shawinigan, and Sudbury all recorded hourly average sulfur dioxide concentrations that exceeded the maximum acceptable level one-hour air quality objective of 340 ppb, but in all cases the objective was exceeded less than 1 percent of the time. At all but the Quebec City, Rouyn, and Shawinigan sites, however, the 24-hour maximum acceptable level air quality objective of 110 ppb was not exceeded in 1986. The 24-hour maximum tolerable level was exceeded 12 times in Quebec City in 1986.

See Table P-4 for air quality objectives for sulfur dioxide.

Carbon monoxide

Any combustion process where carbon-containing organic material is burned without sufficient oxygen will produce carbon monoxide. Motor vehicles, especially poorly tuned ones, are a major source of carbon monoxide because of the great number on the road at any given time. In fact, more than 66 percent of the carbon monoxide emitted in Canada in 1985 was from the internal combustion engines of motor vehicles, trains, aircraft, and boats, our principal means of transport (Fig. P-5).

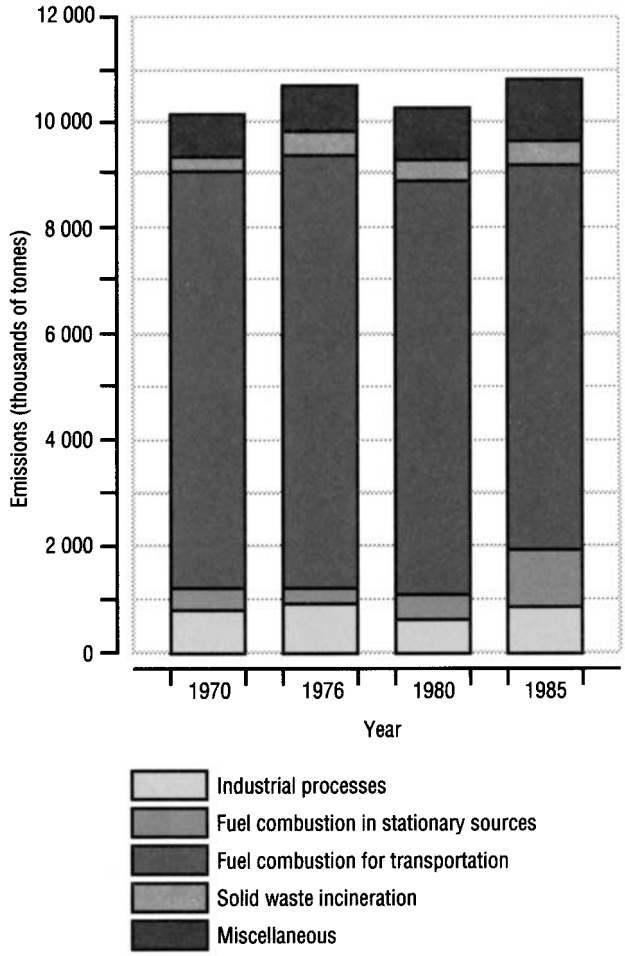


FIG. P-5 Carbon monoxide emissions in Canada, 1970–1985. (Source: Environment Canada.)

The amounts of carbon monoxide emitted from industrial processes, from the use of fuel for generating electric power and for heating homes, offices, and commercial and industrial structures, and from burning wood waste and debris from logging operations are each small in comparison with emissions from transportation, but they are nevertheless important. For example, the concentrated emissions of carbon monoxide from home heating with firewood, although less than 6 percent of total national emissions, are a concern quite distinct from carbon monoxide emissions from motor vehicles, and the potential health effects should not be underestimated.

The more than 10 million gasoline-powered automobiles and trucks that traveled Canada's roads and highways in 1985 emitted about 5.8 million tonnes of carbon monoxide, marginally less than the amount emitted in the early 1970s. Considering that in 1985 there were more than four million additional motor vehicles in use than there were in 1970—i.e., about 70 percent more—achieving a reduction in total emissions over this period is significant. Most of the credit must be attributed to the use of catalytic converters in order to meet federal motor vehicle emissions standards introduced in 1971. These standards were tightened in September 1987, which is expected to lead to even greater emissions reductions.

The toxic effects of carbon monoxide are due to its preferential combination with



FIG. P-6 Eutrophication causes loss of marine life. (Source: Environment Canada.)

the heme component of red blood cells to form carboxyhemoglobin, which reduces the capacity of the red blood cells to carry oxygen to the tissues. Because humans produce carbon monoxide in their bodies as a consequence of normal respiration, levels of carboxyhemoglobin of about 0.05–1.0 percent are found in healthy humans. In humans with hemolytic disease, the level is normally higher. Physiological effects are known to occur when carboxyhemoglobin levels reach 2.5–3.0 percent, and this induces a deficiency of oxygen in the tissues, increased oxygen debt, and even fatal myocardial infarction at exposures of about 5000 ppm for a few minutes or chronic exposure to about 14 ppm.

Chronic exposures of animals (rats and mice) to carbon monoxide have revealed other physiological and behavioral changes. These include an impairment in time discrimination and a consistent trace metal loss, but further work on this aspect is needed before firm conclusions can be drawn.

Special groups at risk when exposed to relatively high levels of carbon monoxide (>2.0%) include those with heart function and blood circulation problems, smokers, anemics, and, possibly, those living at high altitudes.

Only one-hour and eight-hour objectives were recommended to Environment Canada by the Federal–Provincial Committee on Air Pollution (1976); these were 13 and 5 ppm for the desirable levels, and 30 and 13 ppm for the acceptable levels, respectively. These recommendations remained unchanged in the more recent appraisal (1987), except for the eight-hour acceptable level, for which 11 ppm was recommended. Environment Canada is considering this recommended change.

Water pollution

Eutrophication. All lakes have a life cycle; it is perfectly correct to speak of a young lake or an old lake. Within any lake, there is a delicate balance among oxygen supply; animal, plant, and aquatic life; and sediments. As we saw with the nymphs in Lake Erie, if there are too many nutrients, such as phosphorus, the lake becomes over-fertilized. Aquatic life proliferates, places too much demand on the limited amounts of oxygen the water can hold, and then begins to die and settle to the bottom. (See Fig. P-6.) Phosphorus, from agricultural runoff and inadequately

treated sewage, is one of the major causes of overfertilization. Stream runoff and suspended matter in sewage also load large amounts of particles into the lakes. When these settle, along with the dead and decaying aquatic life that has been bred as a result of overfertilization, the sedimentation rate of a lake increases rapidly. Oxygen is exhausted and fish begin to die. The lake silts up and its aging process accelerates; when it reaches its final stages it becomes a swamp and at last, dry land. Lakes Ontario and Erie were already "middle-aged" when the white man arrived. It is estimated that in the first 50 years of the twentieth century, we aged Lake Erie the equivalent of 15,000 years.

Fortunately, once the alarm was sounded, Ontario awakened to the dangers. A program of provincially funded municipal sewage treatment plant construction was begun in the 1950s. This program continues today with the financial assistance of the federal government.

In Ontario, the battle against phosphorus and untreated municipal sewage has been largely successful. Treatment plants were built at a great rate across the province, and in the 1970s, the staggering sum of $991 million was spent or committed for sewage treatment plant construction. Rapid strides have been made in the United States, although some major centers, such as Niagara Falls, still lack proper treatment facilities. The progress that has been made in reducing phosphorus has been dramatic, and the rate of eutrophication has been reduced to something more near normal. Perhaps the greatest source of nutrients now entering the lake is agricultural runoff, with industrial pollution from the air, water, and land also significant. The Canada Water Act of 1972 limited the use of phosphates in household detergents, a major source of phosphorus pollution, to 5 percent by weight. However, other products, such as wetting agents, water softeners, and industrial cleaners, may still contain up to 45 percent phosphates by weight.

The fight that is being fought and is so far being won against phosphates and unnaturally fast eutrophication demonstrates that society and government have had the will, the desire, and the capacity to halt the deterioration of the Great Lakes. That same will must be applied to the other major problem facing the Great Lakes: the menace of toxic chemicals.

Toxic chemicals in the Great Lakes

The increasing detection of toxic chemicals in the Great Lakes (Figs. P-7 through P-10) is the latest challenge we face. Here again we find ourselves having to make up for years of neglect. Over 70,000 chemicals are in use today, and 35,000 have been classified by the EPA as definite or potential threats to human health. New concoctions are being developed at the rate of 1000 per year. Chemicals such as PCBs can be extremely dangerous if uncontrolled. There is a need for the development and establishment of efficient methods of disposal for the PCB wastes that are now in storage in Ontario and American jurisdictions. The problem is compounded by the fact that, once in the food chain, many chemicals accumulate in the bodies of organisms that eat or drink substances that have been contaminated by them. The higher a species is in the food chain, the more the effect is magnified. Humans are at the top of the chain.

With the development of sophisticated monitoring equipment, we have discovered that the Great Lakes, Erie and Ontario in particular, are contaminated with hazardous substances. Trace elements in extremely small quantities are now detectable. Unfortunately in many cases, it is virtually impossible to say how much of a given chemical is too much. The exhaustive and expensive testing necessary to determine the amounts of a given chemical that are harmful has only been carried out on a fraction of the substances now in use. Some chemicals are so harmful that

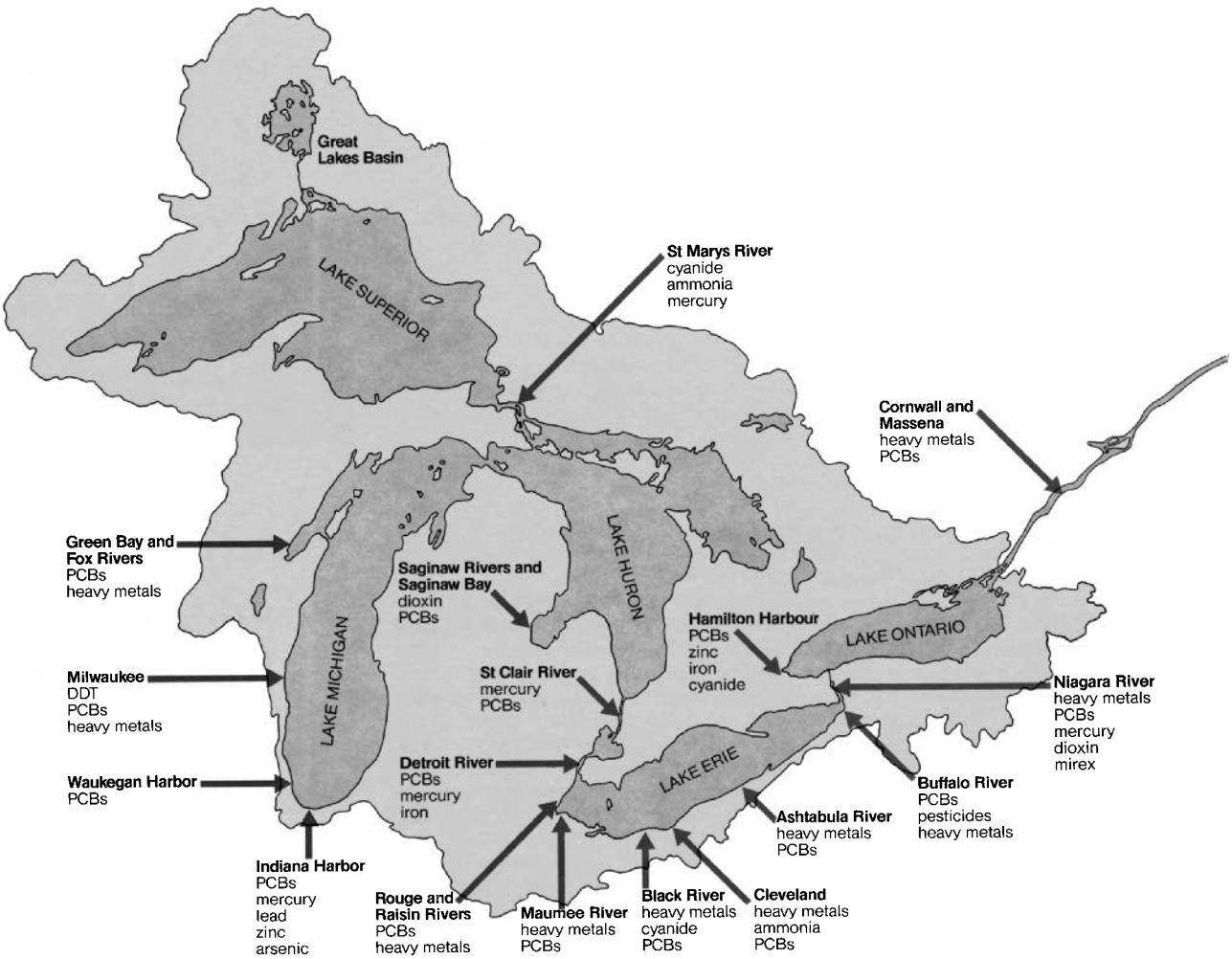


FIG. P-7 Toxic substances in the Great Lakes. (Source: Environment Canada.)

only a tiny bit, or a small accumulation, could cause serious illness or even death.

Furthermore, chemicals that are harmless or relatively harmless by themselves can become dangerous when they come in contact with other substances. These effects are extremely hard to predict, and there are so many potential combinations, that it would be impractical as well as prohibitively expensive for scientists to test all the possibilities. Far better to keep the lakes from becoming contaminated with hazardous substances in the first place.

The toxics problem is also international. It has been estimated that the overwhelming portion of the chemical contamination of the Niagara River comes from sources on the United States side of the border. But toxics also find their way into the Great Lakes from sources on the Canadian side. Agricultural pesticides from land runoff and chemicals from industry are examples.

The seriousness of the trouble, and the need for fast and effective action, has been



FIG. P-8 Illegal chemical dumping. (Source: Environment Canada.)



FIG. P-9 Monitoring water quality is a year-round job. (Source: Environment Canada.)

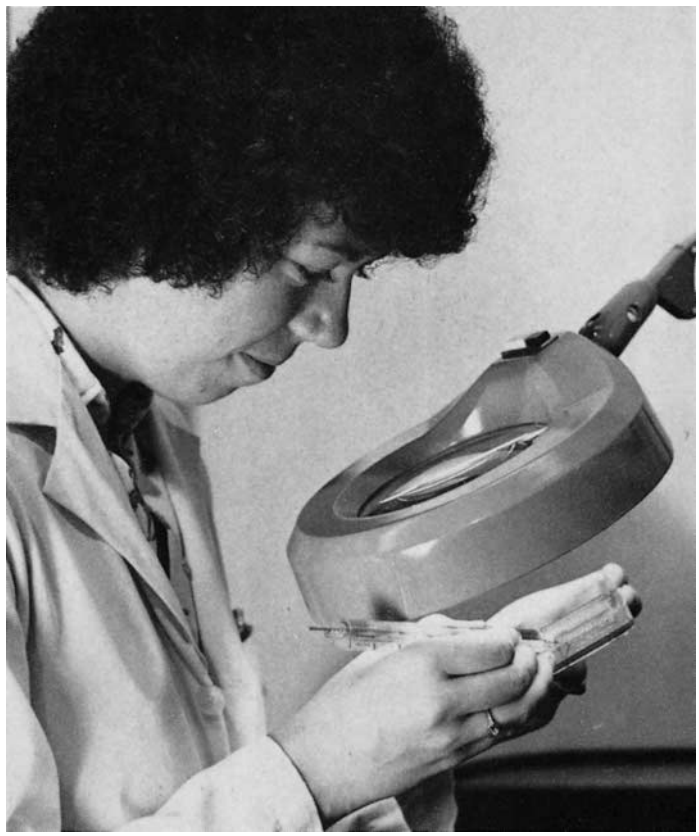


FIG. P-10 Monitoring water quality is a year-round job. (Source: Environment Canada.)

recognized by all. Canada has been in close consultation with the United States on such matters as the cleanup of the Niagara River. Both the federal and Ontario governments have developed or announced plans for the control and management of toxic chemicals. Inventories of toxic substances, crisis planning, cradle-to-grave tracking of toxics, and extensive investigative programs are a few of the measures being taken or advocated. Canada and Ontario together spend nearly $10 million per year monitoring and diagnosing the condition of the Great Lakes, with special attention paid to toxics. Great pains are taken to ensure that the United States has access to extensive Canadian environmental data, and the two nations cooperate through the Great Lakes International Surveillance Plan. Although drinking water supplies meet present guidelines, these guidelines cover only a small percentage of the many chemicals that occur in the Great Lakes. Clearly, the toxics situation cannot be allowed to deteriorate further.

The toxics issue must be solved in two basic ways. First, as much as possible of the hazardous wastes that already exist must be destroyed or recycled, and when necessary, safe methods must be used to store wastes that cannot be destroyed. Second, industrial wastes must be eliminated or reduced, *at their source*, to the fullest extent possible. Plants must be designed to produce no waste, or as little as possible; greater efficiency means less waste. This is the long-term answer to the problem—human ingenuity. Such things as closed-loop systems and waste exchange (whereby the wastes of one process become the raw materials of another) can minimize or eliminate entirely the need for the disposal of toxic wastes. Systems like this already exist, and they are economically feasible.

Society already has the technology necessary to reduce and eliminate toxics entering the Great Lakes. Whether we use it is a matter of will. If we want the benefits that chemicals can give us, then we must act responsibly in their use and disposal. If we don't clean up our act, we'll poison ourselves. It's as simple as that.

Demographic predictions. The present population in the Great Lakes basin is around 37 million; this is expected to double in the next 40 years. Sixty percent of Ontario's population now lives in the six major urban centers of Toronto, Hamilton, Ottawa, Kitchener-Waterloo, London, and Windsor, all within the Great Lakes watershed. It is forecast that by 2020 this will rise to 80 percent.

By the year 2020, the United States will require in the Great Lakes basin:

- For power generation, 15 times more land than at present.
- For power generation, 13 times more cooling water.
- Eight times more industrial water.
- Five times more irrigation water.
- Twice as much sewage capacity.
- Twice the present amount of land devoted to urban use.

Wastewater flows

Case study: British Columbia townships. Sewage treatment plants (STPs), which are operated by the Greater Vancouver Sewerage and Drainage District, discharge enough wastewater each year to fill B.C. Place Stadium 160 times. Flows have increased by 60 percent since 1976 for the Annacis plant and are expected to double by 2036. Flows from the Iona plant, which are now discharged to a deep sea outfall in Georgia Strait, are expected to remain at their current level, while steady growth is expected for the Lulu Island plant. Primary sewage treatment removes suspended particles from the waste stream and the remaining waste water is chlorinated in the summer months. STPs also produce sludges that can be contaminated with heavy metals.

Between Kanaka Greek and Hope, six municipal STPs discharge approximately 35,000 m^3/day of secondary-treated* effluent. Steady growth is expected for these areas. Throughout the Lower Fraser River Basin there are approximately 20 small private STPs treating effluent from schools, marinas, trailer parks, or other developments. Almost one-half the 1360 m^3/day of effluent discharges from these sources are to ground disposal systems. (See Figs. P-11 and P-12.)

Effluents from the Annacis and Lulu Island STPs frequently contain higher levels of contaminants than permitted by the provincial government. For the Annacis plant permit, noncompliance is most apparent for Biochemical Oxygen Demand (BOD),† toxicity, oil and grease, and dissolved oxygen. For example, in 1985 toxicity levels were exceeded 50 percent of the time for Annacis and 66.7 percent of the time for Lulu Island STPs. The toxic compounds identified in municipal STP effluent include un-ionized ammonia, cyanide, sulfides, chlorine, chloramines, phenols, anionic surfactants, heavy metals, and organic compounds. Table P-5 provides a

---

* After primary treatment, secondary treatment involves using either anaerobic bacteria (which do not use oxygen) or aerobic bacteria (which use oxygen) to treat the sewage.

† BOD—the oxygen required for the biochemical breakdown of organic material and the oxidation of inorganic materials such as sulfides and iron.

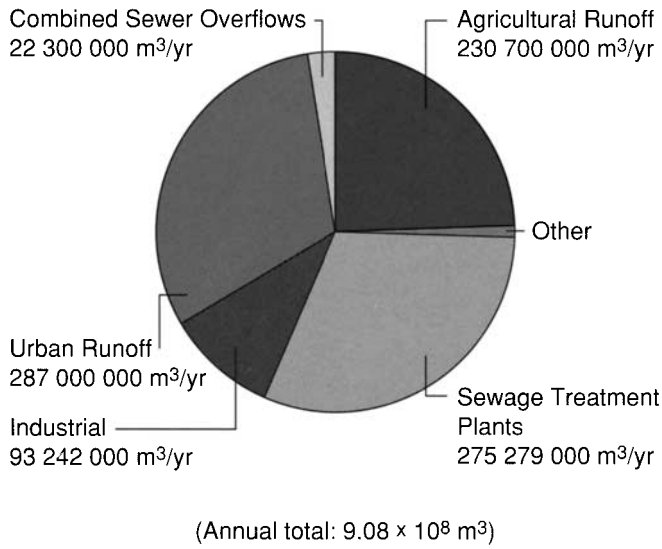


FIG. P-11 Summary of wastewater flows to the Fraser River estuary and Boundary Bay, 1987. (Source: Environment Canada.)

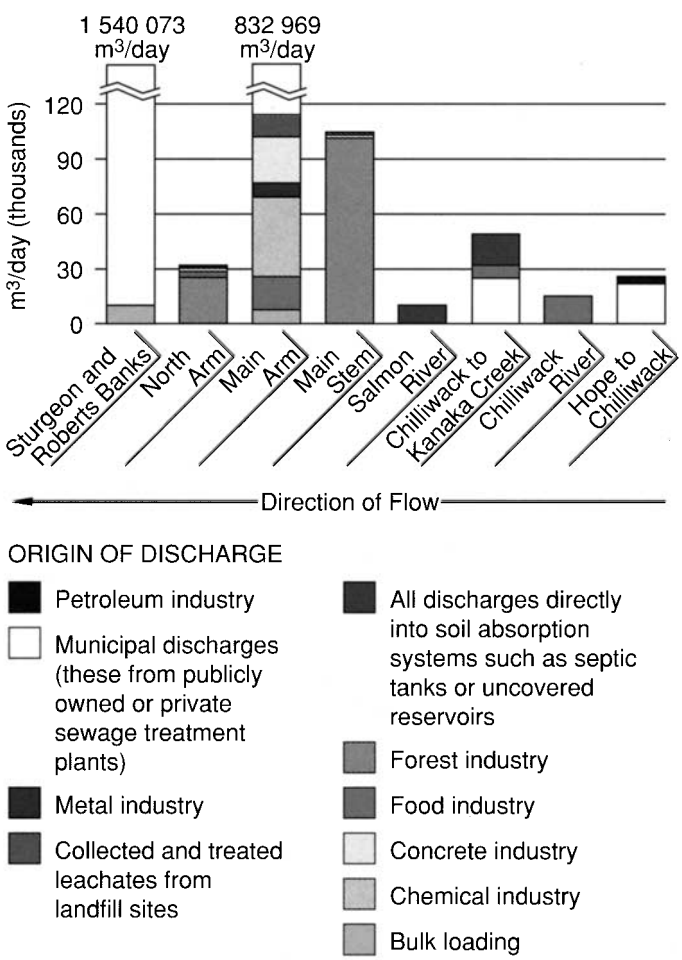


FIG. P-12 Distribution of discharges in the Lower Fraser River Basin authorized by B.C. Ministry of Environment Permits in 1987. (Source: Environment Canada.)

TABLE P-5 Sewage Treatment Plant Contaminant Loadings, 1985

Parameter (kg/day)	Iona	Annacis	Lulu
Discharge (m^3/day)	466,789	291,791	41,230
Oxygen demand	37,810	45,519	5,731
Suspended solids	26,607	20,717	2,639
Kjeldahl nitrogen	7,469	7,587	1,237
Ammonia	4,108	4,669	817
Fluoride	75	44	5.8
MBAS*	420	554	87
Sulfate	12,137	7,878	1,484
Calcium carbonate	33,609	30,930	4,535
pH	3,361	2,013	284
Phosphorus (diss)	840	905	136
Phosphorus (total)	1,354	1,314	219
Oil and grease	7,469	8,462	1,278
Phenol	14	15	1.6
Boron (diss)	75	85	13
Aluminum (diss)	bdl	bdl	33
Aluminum (total)	ai	233	115
Cadmium (total)	ai	ai	0.07
Chromium (total)	bdl	bdl	6.2
Copper (diss)	19	15	2.1
Copper (total)	47	41	6.6
Iron (diss)	135	236	41
Iron (total)	415	525	111
Lead (diss)	ai	3.8	0.7
Lead (total)	20	12	2.4
Manganese (diss)	23	20	2.5
Manganese (total)	28	29	3.7
Nickel (diss)	bdl	ai	4.9
Nickel (total)	bdl	ai	6.2
Zinc (diss)	37	29	6.2
Zinc (total)	56	50	14

NOTES:
Derived by multiplying finite effluent concentrations by the STP average reported flows for 1987.
ai: average indeterminate
bdl: below detection level
*MBAS: methylene blue active substances; ingredient in detergents and foaming agents.
SOURCE: Environment Canada.

summary of annual contaminant loadings and characteristics for the Annacis, Lulu, and Iona STPs.

Due to tidal conditions in the Fraser River, this effluent can pool and spread across the river within two hours at slack tide, exposing millions of juvenile salmon and eulachon larvae during downstream migrations. During low river flows, the effluent from Annacis STP, for example, can reside in the river for up to 1.7 days. Major concerns exist regarding the lethal and sublethal effects of the toxicity of the effluent on both anadromous and nonanadromous fish in terms of bioaccumulation, stress, disease, reproduction, feeding behavior, etc. Despite these concerns there are no techniques currently in place to link these effects to overall impacts on fish populations.

Industrial effluent. Authorized discharges from chemical, concrete, food, forest, gravel washing, metal fabricating and finishing, port industries, and other industrial sectors in the Lower Fraser River Basin total almost 300,000 m^3/day, 90

percent of which occur in the estuary. This is a drop from discharges of 351,571 m^3/day in 1973 probably due to industrial hookups to Annacis Island STP in 1975. Of 116 authorized waste management permits, 11 contribute about 80 percent of total industrial effluent flows.

Pollutant loadings include oil, grease, solids, metals, and organics. Total loadings are difficult to determine as permit requirements may not include all parameters, reporting periods and sampling methodology vary between permits, a few permit holders are in noncompliance situations, and unauthorized discharges may be occurring. Analysis of data for the 63 industrial permit holders on the Fraser River below Kanaka Creek show reported loadings of 4739 kg/day of BOD, 7226 kg/day of solids, 342 kg/day of oil and grease, and 5780 kg/day of nutrients. High priority industrial dischargers have been identified based on their flows and contaminant loadings.

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.

Portland Cement (*see* Cement)

Power Production; Power Production In-House; IPP; SPP

The acronym IPP generally refers to firms that existed for the sole purpose of investing in and building power plants and selling the power to a national governing body or other large customers. IPP ranks are swelling to include "small" power producers, whose production of power is secondary to their main purpose.

Small producers include large industrial entities, such as refineries and manufacturing plants, that buy their own power production machinery (sometimes to avoid expensive brownouts or outages) and make their own power. In most countries they can sell their excess power back to the national grid. The limits of this sale are generally set by the size of the distribution lines available. This small power producer generally gets less of a tariff for its power than it pays for national grid-supplied power. As such power producers increase, they lessen demand growth and therefore the required size of new, large power plants.

National power authorities traditionally move with a sluggishness that struggles to keep up with increasing power demand and changes in environmental legislation. However, the nature of the contemporary power business forces certain other optimization measures as the following paragraph indicates.

A power station in Dagenham, England, with both Alberta (Canada) power and English partners for owners, is an IPP. In anticipation of CO_2 (carbon dioxide) emissions legislation, the firm ordered a high-precision condition-monitoring system for its power-generation turbines. Their logic: the system would optimize fuel consumption and cut down on CO_2 emissions. Current technology made the cost of the system initially ordered unnecessarily high; nonetheless, the trend is clear.

Note also that, generally, IPPs have to have the mental flexibility to see the return on investment of such a system. The potential effect on the national power to IPP power production ratio in the future is evident. If national power producers do not react swiftly to changing environmental pressures, their profitability margins could decrease to the point where IPPs can further encroach on their territory.

One lesson learned from the severe ice storms suffered by Canada and the United States early in 1998 is that smaller IPP installations might prove less of an "Achilles heel" (weak link) to overall power demand than a few large national power plants.

Sagging national nuclear industries in Canada, the United States, and Japan are testament to overly optimistic life prognoses of nuclear fission reactors. They have been, and will continue to be, decommissioned. This can result in several smaller IPPs taking up the slack.

If the process plant does not want to "go it alone" to be an IPP, other willing partners may be available. IPP ranks are further being swelled by IPP joint-venture companies that can have one of the turbine manufacturers, such as Alstom [formerly ABB (Asea Brown Boveri)] or Siemens, as a major or controlling-interest partner. Interesting variations on a theme can be arranged contractually with original equipment manufacturers (OEMs). Alstom had a turnkey arrangement on the Kuala Langat, Malaysia, plant with the Genting Corporation, Malaysia, and the Lumut, Malaysia, plant with Segari Ventures, Malaysia. The pulp mill next to the Kuala Langat plant exchanges steam with its power-generation neighbor. Transmission and distribution systems in the vicinity also provide scope for minimizing hardware.

IPP Trends Globally

IPP conglomerates that include an OEM or large contractor, such as Enron, will continue to increase. Partners for many of these ventures include major oil and process firms. The advantage gained by joining forces with an OEM can include bargaining the terms of comprehensive maintenance contracts. If they team up with a major contractor, they may thus have negotiated a plant expansion for optimized dollars per unit of capacity. Alstom, for instance, is starting to increase its ownership of power facilities, even partially owned state or municipality ones, in the United States, such as the massive Midland plant. Alstom's participation in long-term comprehensive maintenance contracts in power projects, such as Deeside in the United Kingdom, serve to illustrate how entwined OEMs now are with the IPP sector. Quite apart from the return-on-investment figures that a "plain" investor might consider, the profit margin on spare parts and the markup on maintenance or construction services further add to the attraction of IPP projects for OEMs and contractors.

Deregulation of the power industry, increasing environmental legislation, and the increased difficulty of maintaining profit margins serve to accelerate the gradual turnover of national power authorities' territory to IPPs. The nuclear industries in Canada, the United States, and Japan are likely to provide further illustration of this fact in the near future.

More and more, "IPP" can mean "small IPP." As tax incentives for internal power production rise, some countries that were formerly opposed to SPPs are now lifting their objection. Singapore is one such example. The corresponding number of firms who then qualify to invest as IPPs increases correspondingly.

Oil and gas companies increasingly make their own power. They then become their own best customer. This trend is further stepped up as technology makes viable fuel selections of many of the "unwanted" by-products these facilities produce. The resultant economic benefits of producing their own power escalate further over time. An excellent example of this is seen under Stepper Motor Valves (a subsection of Control Systems) with the example of the PCS plant in Singapore.

Power Transmission

Power transmission is the act of taking power from a driving piece of equipment (such as a gas turbine, steam turbine, or motor) and transmitting it to a driven piece of equipment (such as a compressor or a pump). Power-transmission

equipment then includes gears and gearboxes, couplings, and other systems that transmit power from the "driver" to the "driven."

In this section, model numbers used by the information source companies will appear, as in other sections in this book. Care was taken to get source information from suppliers with the widest product ranges currently available, so the reader can then use this information as a basis for comparison with other OEMs being considered.

Gears*

Helical gears

Gears are associated with nearly every human activity in the modern world. They come in all sizes, shapes, and materials. They go by such names as spur, helical, bevel, hypoid, worm, skew, internal, external, epicyclic, and so on.

The following material is presented to assist an engineer who is not a gear specialist in determining the basic size and requirements of a gearset for one specific type of gearing: high-speed, high-power parallel-axis gears. The industry definition of high speed is 3600 rpm and/or 5000 ft/min pitch-line velocity. In this instance, *high power* means from 1000 to 2000 hp at the low end and upward of 50,000 hp at the high end. The kinds of applications that generally require high-speed gearing are those involving steam and gas turbines, centrifugal pumps and compressors, and marine propulsion equipment.

High-speed gears. Gears for high-speed service are usually of the helical type. They can be either single- or double-helical and can be used in either single or double stages of increase or reduction, depending on the required ratio. The ratio of a single stage is usually limited to about 8 to 1. There is a very small difference in frictional loss at the teeth, depending on whether the pinion or the gear is driven, but for all practical purposes no distinction need be made between speed increasers and speed reducers.

Most high-speed gearing operates at pitch-line velocities of 25,000 ft/min or less. At higher speeds, up to about 33,000 ft/min, special consideration must be given to many aspects of the gearset and housing. Speeds of over 33,000 ft/min should be considered developmental.

As gears go faster, the need for gear accuracy becomes greater. The following can be used for guidelines for high-speed gearing. Tooth-spacing errors should not exceed about 0.00015 in; tooth-profile errors, about 0.0003 in; and helix or lead error, as reflected by tooth contact over the entire face, about 0.0005 in. The usual range of helix angles on single-helical gears is between 12 and 18°. For double-helical gears, the helix is generally between 30 and 40°. Pressure angles are usually found between 20 and 25° (in the plane of rotation).

In addition to the requirement for extreme accuracy, a characteristic of high-speed helical gears that sets them apart from other helicals is the design objective of infinite life, which in turn results in fairly conservative stress levels.

Overload and distress. If a gearset is overloaded from transmitting more than the design power, or by being undersized, or as a result of misalignment, the teeth are likely to experience distress. The three most probable forms of distress are *pitting*, *tooth breakage*, and *scoring*.

---

* Source: Demag Delaval, USA.

Pitting is a surface-fatigue phenomenon. It occurs when the hertzian, or surface, compressive stresses exceed the surface-endurance strength.

Tooth breakage is exactly what the name implies: sections of gear teeth literally break out. It occurs when the bending stresses on the flank or in the root of the teeth exceed the bending-fatigue strength of the material.

Scoring, sometimes called *scuffing*, is actually instantaneous welding of particles of the pinion and gear teeth to each other. It occurs when the oil film separating the teeth becomes so hot that it flashes or so thin that it ruptures, thereby permitting metal-to-metal sliding contact. The heat generated as the pinion and gear teeth slide on each other is sufficient to cause localized welding. These tiny welds are immediately torn loose and proceed to scratch the mating surfaces—hence the name *scoring*.

Neither pitting nor scoring causes immediate shutdown. If allowed to progress, however, they can produce a deterioration of the involute profiles in addition to producing stress risers. If permitted to continue too long, pitting or scoring can lead to tooth breakage.

Basic sizing. The basic sizing of a gearset, or what can be called the *preliminary design*, is based on resistance to pitting. Since the surface endurance strength is a function of the material hardness, preliminary sizing of a gearset is relatively simple. It should be understood, however, that the final design requires the efforts of a competent gear engineer to investigate and attend to such matters as:

1. The selection of materials and processing
2. The determination of the number of teeth on the pinion and gear, which is a function of the pitch, which in turn determines the tooth bending strength
3. An investigation of the scoring resistance of the gearset, which is a function of the gear-tooth geometry, the surface finish of the teeth, and the properties of the lubricant
4. Rotor proportions and bearing design, with particular interest in related vibration characteristics
5. Gear-case features, including such things as running clearances, proper drainage, venting, mounting, doweling, and, in particular, maintenance of internal alignment
6. The many system considerations such as lateral and torsional vibration, external alignment with associated forces and moments on shaft ends, torque pulsations, etc.

The American Gear Manufacturers Association's (AGMA) fundamental equation for surface durability (pitting resistance) of helical gear teeth is

$$s_c\left[\frac{C_L C_H}{C_T C_R}\right] \leqq \left[\frac{1/\pi}{\dfrac{1-\mu_p^2}{E_p}+\dfrac{1-\mu_G^2}{E_G}}\right]^{1/2} \times \left[\frac{W_t C_o}{C_v}\frac{C_s}{dF}\frac{C_m C_f}{I}\right]^{1/2} \tag{P-1}$$

where s_c = contact-strength number
C_L = life factor
C_H = hardness-ratio factor
C_T = temperature factor
C_R = factor of safety

μ_p, μ_G = Poisson ratio for pinion and gear
E_p, E_G = modulus of elasticity for pinion and gear
W_t = transmitted tangential load at pitch diameter
C_o = overload factor
C_v = dynamic factor
C_s = size factor
d = pinion pitch diameter
F = face width
C_m = load-distribution factor
C_f = surface-condition factor
I = geometry factor

Substituting appropriate values for high-speed gears and rearranging Eq. (P-1) results in

$$K \leqq [s_c \times 10^{-4}]^2 \times \frac{1.56}{C_o C_m} \tag{P-2}$$

where K is an index of hertzian stress. It is defined mathematically as:

$$K = \frac{W_t}{Fd} \times \frac{R+1}{R} \tag{P-3}$$

where R = ratio (D/d)
D = gear pitch diameter
W_t = tangential tooth load
$$= \frac{126{,}000 \times \text{hp}}{N_p \times d}$$
and
hp = transmitted, or design, horsepower
N_p = pinion, rpm

The term C_o, the overload factor, is accounted for by application service factors (SF), shown in Table P-6.

The term C_m, the load-distribution factor, accounts for maldistribution of load across the face width of the gearset due to lateral bending and torsional twisting of the pinion, thermal distortion of the pinion and/or the gear, and centrifugal deflection of the gear. If the length-diameter ratio (L/d) of the pinion is kept within reasonable limits, usually less than 2.2 for double-helical and 1.5 for single-helical gears, and proper attention is paid to cooling and gear-band deflection, the magnitude of the C_m factor will probably lie between 1.2 and 1.4. If the higher value is used in the interest of conservatism, Eq. (P-2) can be further simplified to

$$K \leqq \frac{(s_c \times 10^{-4})^2 \times 1.11}{\text{SF}} \tag{P-4}$$

By using Eqs. (P-3) and (P-4) and Tables P-6 and P-7, the basic size of a high-speed gearset can be determined in several minutes on a hand calculator.

Table P-7 gives values of s_c for commonly used gear materials. The rating of a gearset should be based on the softer of the two members, which is normally the gear. The Brinell hardness of the pinion in through-hardened gearsets is usually about 1.2 times that of the gear.

Arrangements. Figures P-13 and P-14 show sections through a typical industrial high-speed-gear unit. See Table P-6.

TABLE P-6 Service-Factor Values

Application	Service Factor		
	Prime Mover		
	Motor	Turbine	Internal-Combustion Engine (Multicylinder)
Blowers			
Centrifugal	1.4	1.6	1.7
Lobe	1.7	1.7	2.0
Compressors			
Centrifugal: process gas except air conditioning	1.3	1.5	1.6
Centrifugal: air-conditioning service	1.2	1.4	1.5
Centrifugal: air or pipeline service	1.4	1.6	1.7
Rotary: axial flow—all types	1.4	1.6	1.7
Rotary: liquid piston (Nash)	1.7	1.7	2.0
Rotary: lobe-radial flow	1.7	1.7	2.0
Reciprocating: three or more cylinders	1.7	1.7	2.0
Reciprocating: two cylinders	2.0	2.0	2.3
Dynamometer: test stand	1.1	1.1	1.3
Fans			
Centrifugal	1.4	1.6	1.7
Forced-draft	1.4	1.6	1.7
Induced-draft	1.7	2.0	2.2
Industrial and mine (large with frequent-start cycles)	1.7	2.0	2.2
Generators and exciters			
Base-load or continuous	1.1	1.1	1.3
Peak-duty cycle	1.3	1.3	1.7
Pumps			
Centrifugal (all service except as listed below)	1.3	1.5	1.7
Centrifugal: boiler feed	1.7	2.0	. . .
Centrifugal: descaling (with surge tank)	2.0	2.0	. . .
Centrifugal: hot oil	1.5	1.7	. . .
Centrifugal: pipeline	1.5	1.7	2.0
Centrifugal: waterworks	1.5	1.7	2.0
Dredge	2.0	2.4	2.5
Rotary: axial flow—all types	1.5	1.5	1.8
Rotary: gear	1.5	1.5	1.8
Rotary: liquid piston	1.7	1.7	2.0
Rotary: lobe	1.7	1.7	2.0
Rotary: sliding vane	1.5	1.5	1.8
Reciprocating: three cylinders or more	1.7	1.7	2.0
Reciprocating: two cylinders	2.0	2.0	2.3
Marine service			
Ship's service turbine-generator sets	. . .	1.1	. . .
Turbine propulsion	. . .	1.25	. . .
Diesel propulsion	. . .	. . .	1.35

Figures P-15 and P-16 show sections through a typical turbine-driven marine propulsion reduction gear. It will be noted that the high-speed pinions each mesh with two first-reduction gears, thereby splitting the power from each turbine. These twin-power-path gears, or so-called locked-train gears, are popular in the horsepower range of 30,000 shp and up.

Figures P-17 and P-18 show sections through a typical diesel-driven marine propulsion reduction gear. In this arrangement, each pinion is fitted with a pneumatically operated clutch that permits either engine to be operated singly or one engine ahead and one astern for fast maneuvering.

TABLE P-7 s_c Values

	Gear Hardness	s_c
Through-hardened	229 BHN	112,000
	248 BHN	117,500
	302 BHN	135,000
	340 BHN	152,000
Nitrided	55 R_c	207,000
	58 R_c	218,700
	60 R_c	226,800
	63 R_c	239,400
Case-carburized	55 R_c	230,000
	58 R_c	243,000
	60 R_c	252,000
	63 R_c	266,000

NOTE: BHN = Brinell hardness number; R_c = Rockwell number.

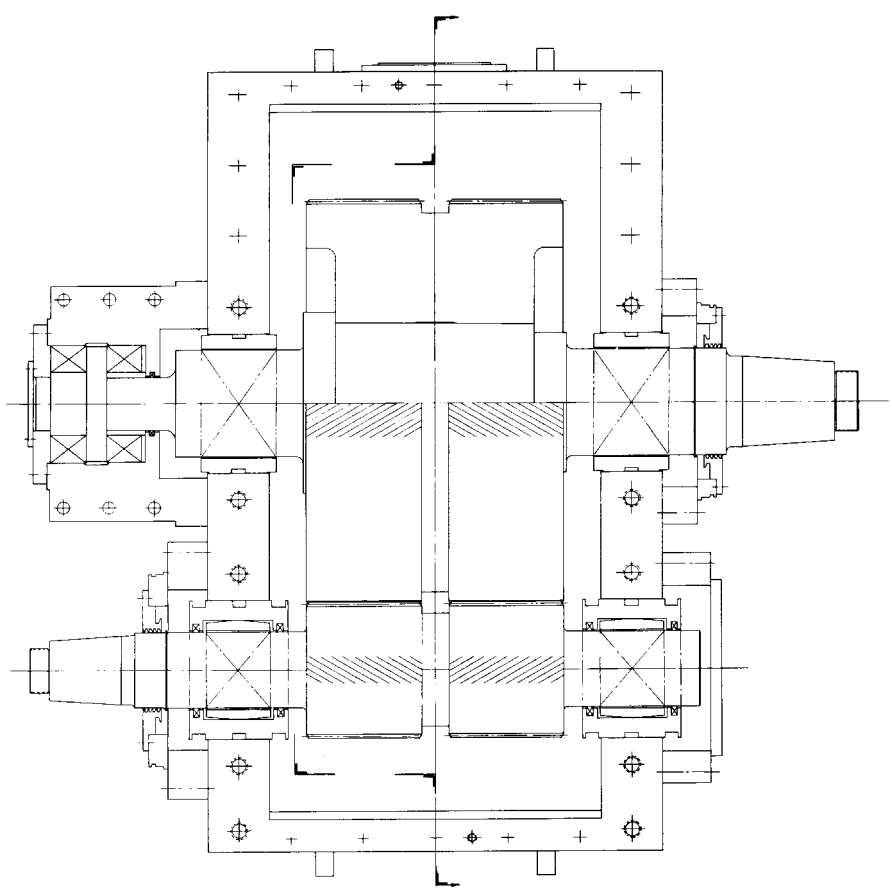


FIG. P-13 Plan cross section, typical industrial gear. (Source: Demag Delaval.)

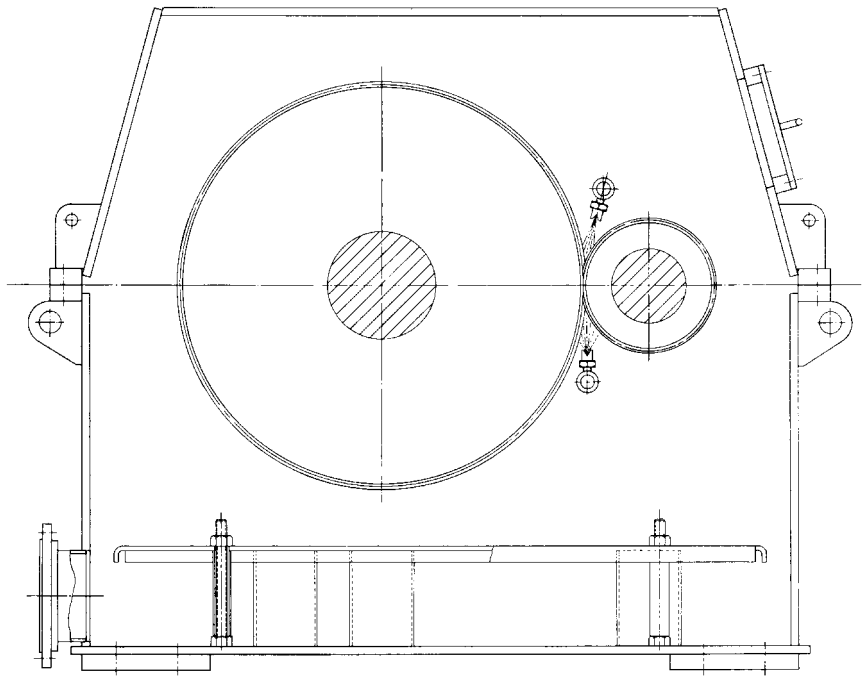


FIG. P-14 End cross section, typical industrial gear. (Source: Demag Delaval.)

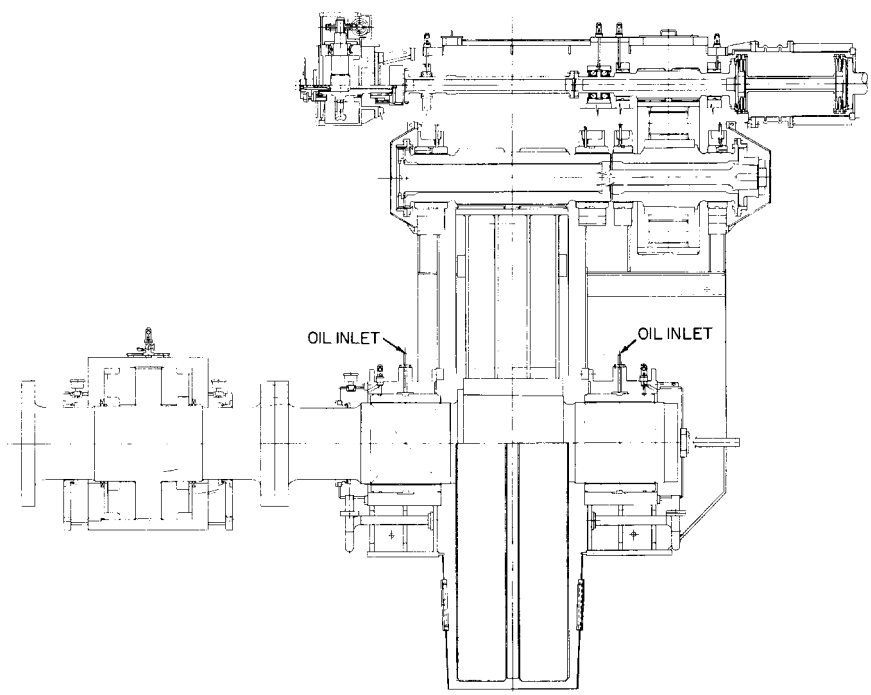


FIG. P-15 Plan cross section, typical locked-train reduction gear. (Source: Demag Delaval.)

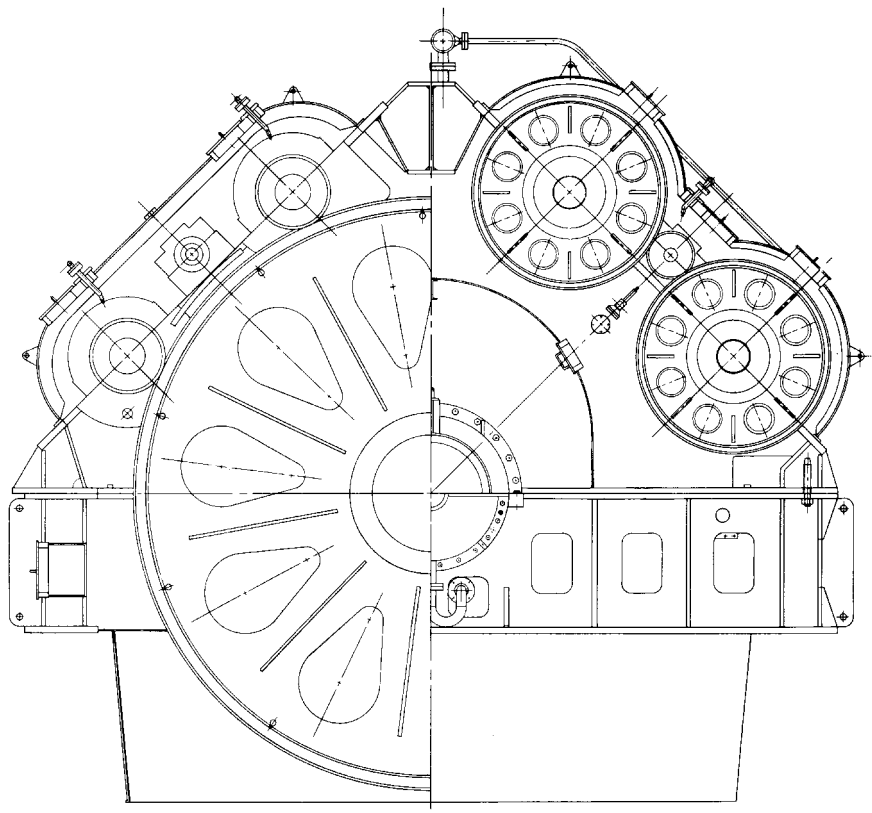


FIG. P-16 End cross section, typical locked-train reduction gear. (Source: Demag Delaval.)

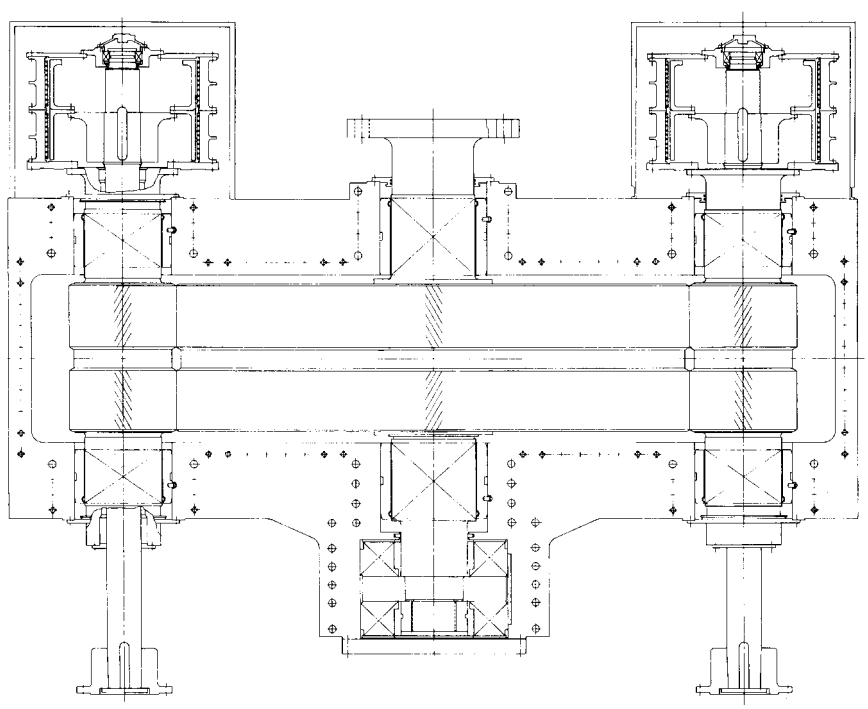


FIG. P-17 Plan cross section, typical diesel propulsion reduction gear. (Source: Demag Delaval.)

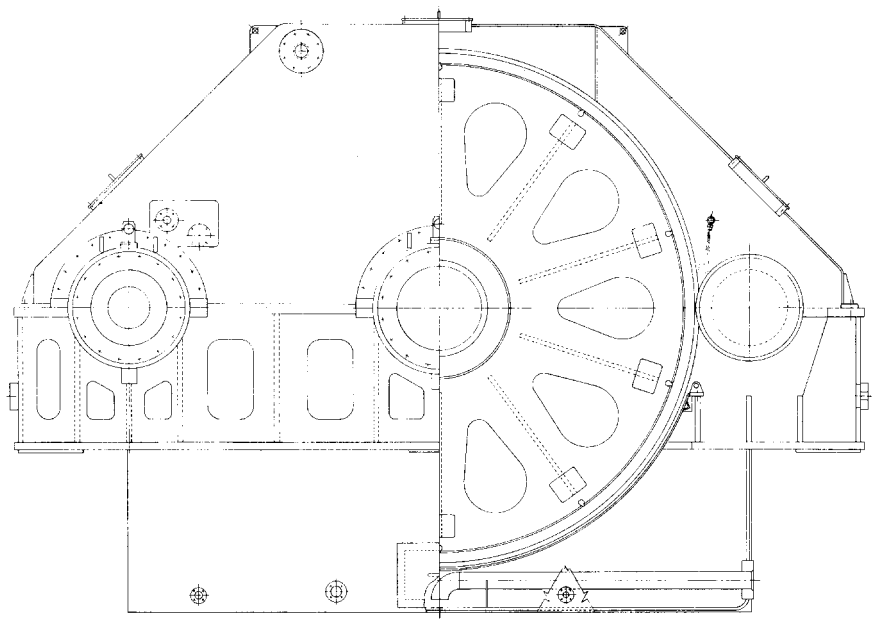


FIG. P-18 End cross section, typical diesel propulsion reduction gear. (Source: Demag Delaval.)

Horsepower losses. Prediction of gear-unit losses is an inexact science at best. The total power loss of a gear unit is made up of (1) the frictional loss in the oil film separating the teeth as they slide over one another, (2) bearing losses, and (3) windage and pumping losses.

Empirical equations have been developed for most types of gears to calculate these losses. Often rule-of-thumb estimates are as good as the calculations. Tooth-mesh losses usually amount to between 0.5 and 1 percent of the transmitted horsepower at each mesh. Bearing losses may vary a bit more, depending primarily on the bearing type, operating clearance, and sliding velocity. They usually fall into a range of 0.75 to 1.5 percent of transmitted power.

Windage losses depend primarily on the clearance between rotating parts and the housing, the smoothness of the surfaces, and the peripheral velocities.

Pumping loss, the displacement of the air-oil mixture from the tooth space as engagement takes place, is influenced by tooth size, helix angle, rotative speed, and location of the oil sprays. Losses of this type are the biggest variable and can fall anywhere from 0.5 to about 2 percent of transmitted power.

The most important consideration is that a realistic view be taken of gear losses when selecting a pump, cooler, and filters for the lubrication system. These should be large enough to do the job.

Lubrication. The oils normally used in high-speed-gear applications are rust- and oxidation-inhibited turbine oils in the viscosity range of 150 to 300 SSU at 100°F. As a general rule, the higher the pitch-line speed of the gear, the lower the viscosity oil required. In marine units, in which the propeller shaft turns at a relatively low speed, pitch-line speeds are frequently found below 5000 ft/min. In these cases, it is generally desirable to use a more viscous oil. The viscosity of the oils frequently found in turbine-driven propulsion plants is in the range of 400 to 700 SSU at 100°F. In diesel propulsion gearing, in which the engine and the gear are on separate systems, the viscosity of the gear oil is frequently in the range of 600 to 1500 SSU at 100°F.

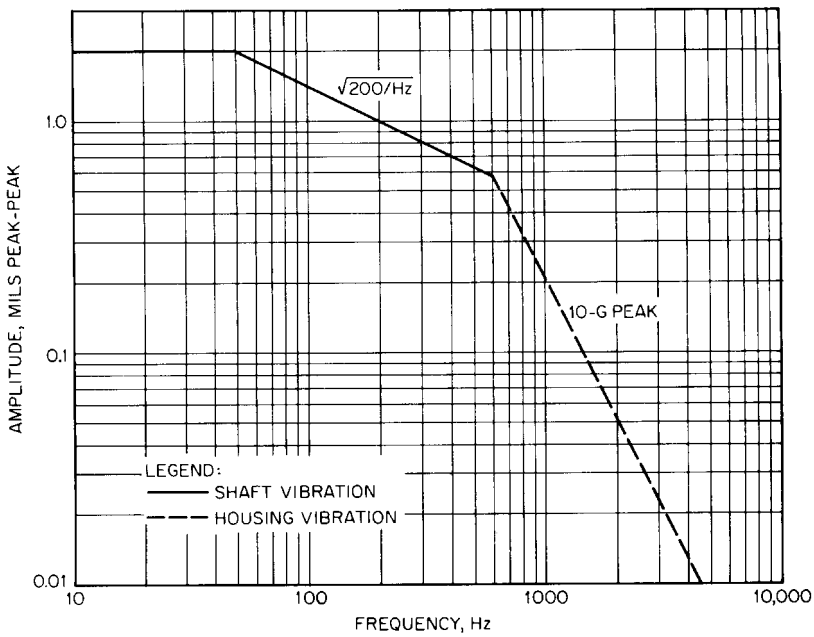


FIG. P-19 Acceptable vibration levels. (Source: Demag Delaval.)

Regardless of the application, the scoring or scuffing resistance of the gear teeth should be investigated. In many cases, it will be desirable to use an oil with appropriate extreme-pressure additives that greatly increase the antiweld or antiscoring characteristics of the lubricant.

Installation and maintenance. If a gear unit is correctly sized, properly installed, and properly maintained, it can be expected to last indefinitely. Proper installation includes (1) proper initial alignment, both internal and external, and (2) a rigid foundation that will not settle, crack, or elastically or thermally deform under operating conditions in amounts greater than the gear-alignment tolerance.

For those interested in additional information on systems considerations (overloads, system vibration, alignment, foundations, piping, and lubrication), AGMA Information Sheet 427.01, *Systems Considerations for Critical Service Gear Drives*, is recommended.

Proper maintenance consists primarily of providing a continuous supply of the correct lubricant at the right temperature, pressure, and condition. Obviously, alignment and balance must be maintained. Vibration monitoring is a good preventive-maintenance tool. Figure P-19 can be used as a guide for acceptable lateral-vibration limits. Additional information regarding vibration instruments, interpretation, tests, etc., may be found in AGMA Standard 426.01, *Specification for Measurement of Lateral Vibration on High Speed Gear Units*.

Worm gears

The use of high-speed drivers for efficient operation makes speed reduction necessary for many applications.

Worm-gear reducers are very compact, requiring less space than belts, chains, or trains of open gearing. The right-angle drive often permits compact placement of the driving and driven machines. Since three or more teeth are always in contact,

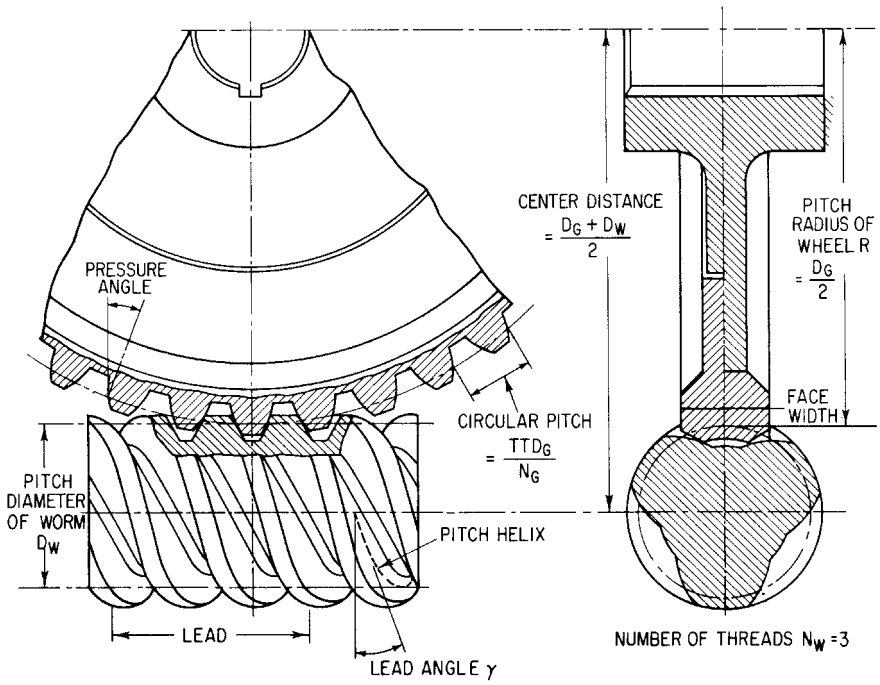


FIG. P-20 Worm gear terminology. (Source: Demag Delaval.)

there is an even flow of torque, which reduces vibration, prolongs the life of the driven machinery, and provides quiet power transmission. There are few moving parts (hence few bearings), and these are enclosed in a dustproof housing that contributes to long life and avoids danger of injury to workers.

Worm gearing consists of an element known as the *worm*, which is threaded like a screw, mating with a gear whose axis is at a 90° angle to that of the worm. The gear is throated and partially envelops the worm. The worm may have one or more independent threads, or "starts."

The ratio of speeds is determined by dividing the number of teeth in the gear by the number of threads in the worm. Since a single-threaded worm acts like a gear with one tooth and a double-threaded worm as a gear with two teeth, very large ratios can be designed into one set of gearing. Ratios between 3 to 1 and 100 to 1 are common for power transmission purposes, and even higher ratios are employed for index devices.

Mechanical elements. Dimensions of the worm and worm gear are defined as follows (see Fig. P-20):

Outer diameter of worm is the diameter of a cylinder touching the tops of the threads.

Pitch diameter of worm is the diameter of a circle that is tangent to the pitch circle of the mating gear in its midplane.

Outer diameter of gear is the diameter over the tips of the teeth at their highest points.

Throat diameter of gear is the diameter over the tips of the teeth at the middle plane that is perpendicular to the axis of the gear shaft and passes through the axis of the worm.

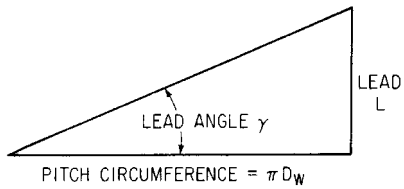


FIG. P-21 Lead angle. (Source: Demag Delaval.)

Pitch diameter of gear is the diameter of the pitch circle at the midplane of the gear that would roll upon the pitch line of the worm if the latter were used as a rack.

Circular pitch is the distance from a point on one gear tooth to the same point of the succeeding tooth measured circumferentially on the midplane pitch circle. It is equal to the axial pitch of the worm, that is, the distance from any point on a thread of the worm to the corresponding point on the next thread, measured parallel to the axis.

Lead of worm is the distance parallel to the axis of the worm from a point on a given thread to the corresponding point on the same thread after it has made one turn around the worm. If the worm has only one thread, this distance is equal to the circular pitch, but if the worm has multiple threads, it is equal to the circular pitch multiplied by the number of threads. It is the distance that a point on the pitch circle of the gear is advanced by one revolution of the worm.

One revolution of the worm advances the gear by as many teeth as there are threads on the worm. Therefore, the *ratio of transmission* is equal to the number of teeth on the gear, divided by the number of threads on the worm, without regard to the pitch.

Lead angle of the worm threads is the angle between a line tangent to the thread helix at the pitch line and a plane perpendicular to the axis of the worm. The pitch lines of the worm threads lie on the surface of a cylinder concentric with the worm and of the pitch diameter. If this cylinder is thought of as unrolled or developed on a plane, the pitch line of the thread will appear as the hypotenuse of a right-angled triangle, the base of which will be the circumference of the pitch circle of the worm and the altitude of which will be the lead of the worm. In Fig. P-21 the lead angle is γ, and the tangent of this angle is equal to the lead L divided by π times the pitch-line diameter D_w of the worm, $\tan \gamma = L/\pi D_w$.

Pressure angle is defined as the angle between a line tangent to the tooth surface at the pitch line and a radial line to that point.

Classification. A large number of arrangements are available, permitting flexibility in application to a wide variety of driven machinery. Some of the typical arrangements manufactured are shown in Figs. P-22 to P-28.

Motorized units may be furnished for:

Horizontal-shaft units
- Single worm reduction
- Helical worm reduction
- Double worm reduction

Vertical-output-shaft units
- Single worm reduction
- Helical worm reduction
- Double worm reduction

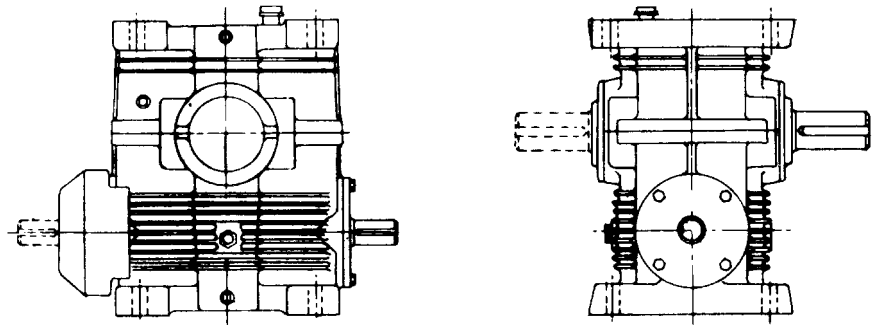


FIG. P-22 Single worm reduction. (Source: Demag Delaval.)

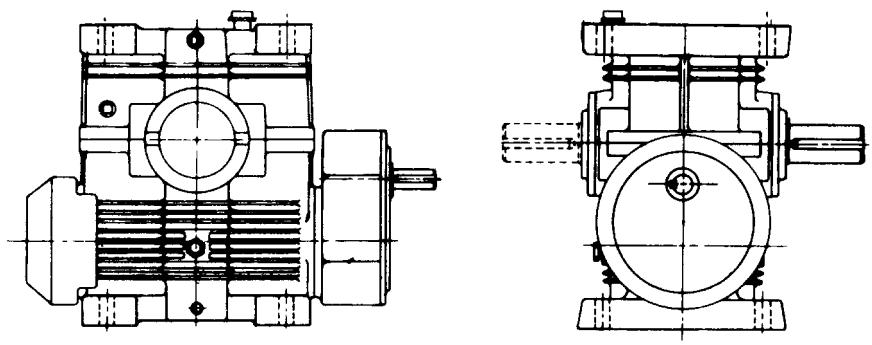


FIG. P-23 Helical worm reduction. (Source: Demag Delaval.)

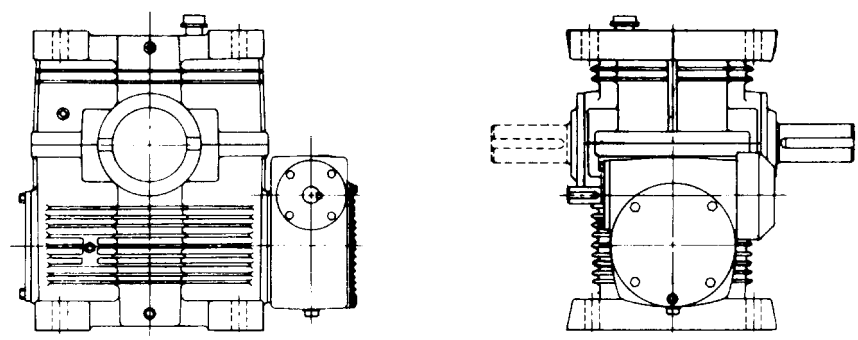


FIG. P-24 Double worm reduction. (Source: Demag Delaval.)

Shaft-mount units
- Single worm reduction
- Helical worm reduction
- Double worm reduction

Special reducers. Special reducers in various combinations are also available. An example is presented in Fig. P-29, which shows a large vertical-output-shaft unit with a single worm reduction having 38-in gear centers, which is used in pulverized-coal service.

Efficiency of worm gearset. To determine the approximate efficiency of a worm gearset in which the worm threads are of hardened and ground steel and the gear

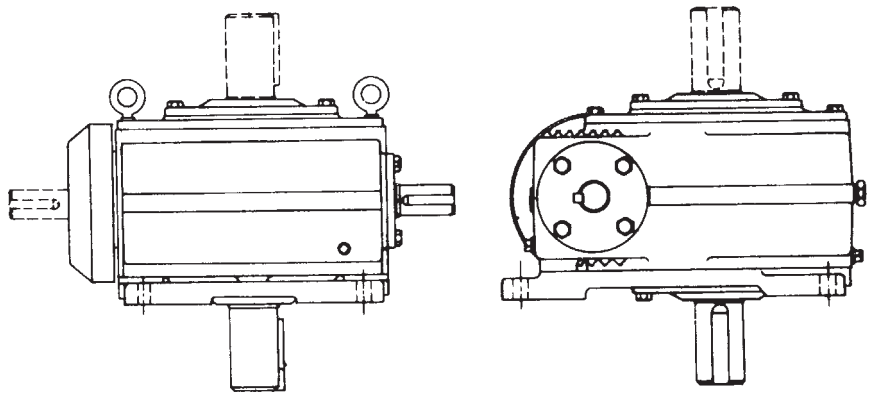


FIG. P-25 Vertical single worm reduction. (Source: Demag Delaval.)

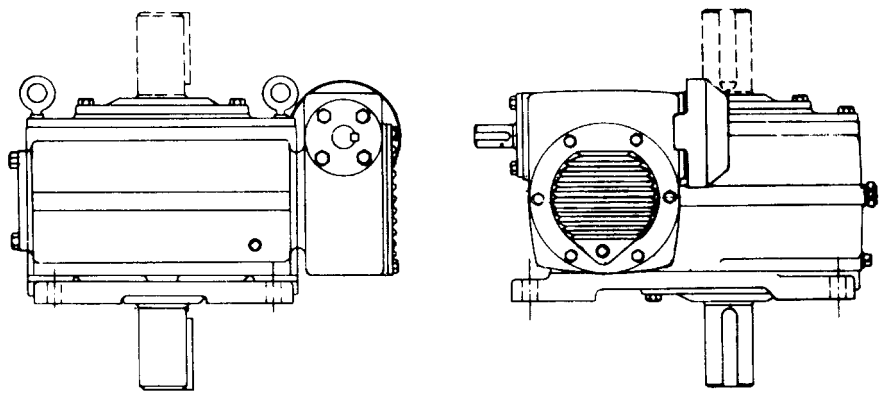


FIG. P-26 Vertical double worm reduction. (Source: Demag Delaval.)

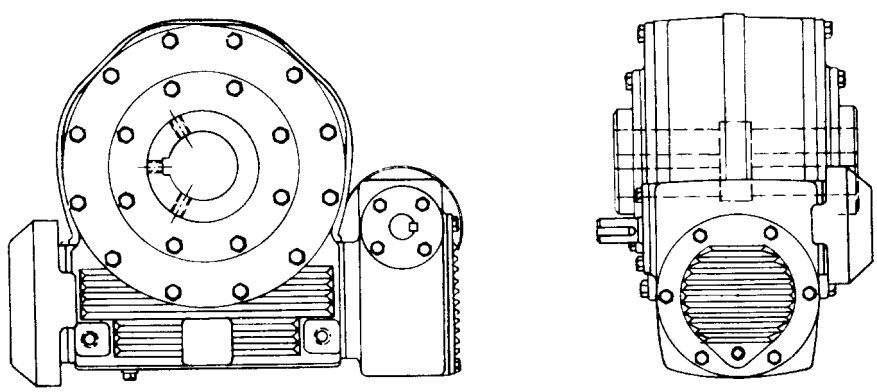


FIG. P-27 Double-worm-reduction shaft-mount unit. (Source: Demag Delaval.)

teeth of nickel bronze or phosphor bronze, lubricated with a steam-cylinder oil, Figs. P-30 and P-31 may be used. To use the coefficient-of-friction curve, calculate the rubbing speed of the worm from the following formula:

$$\text{Rubbing speed, ft/min} = \frac{\text{pitch diameter of worm} \times 0.262 \times \text{rpm}}{\cos \text{lead angle}}$$

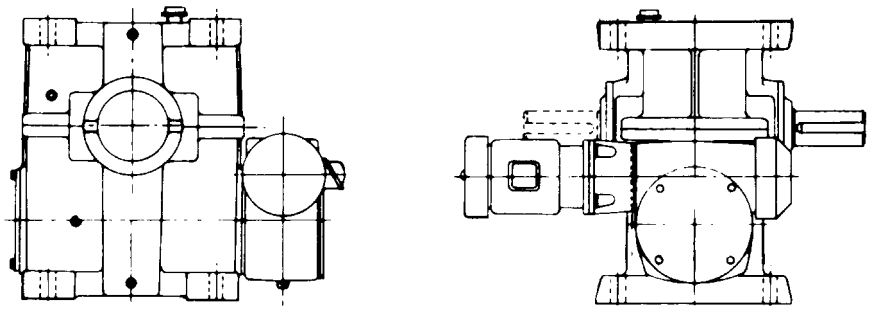


FIG. P-28 Motorized worm reduction. (Source: Demag Delaval.)

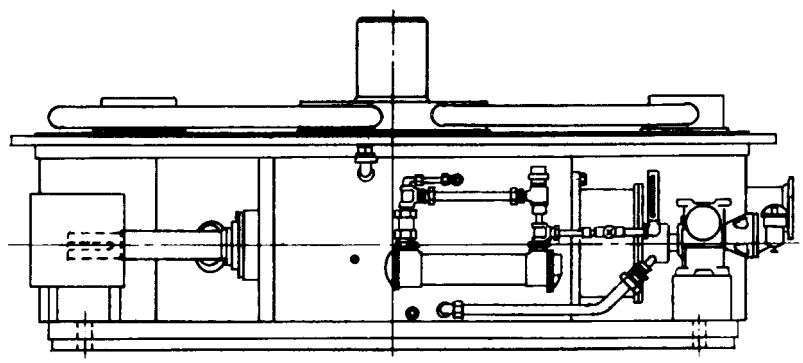


FIG. P-29 Large vertical-shaft single worm reduction. (Source: Demag Delaval.)

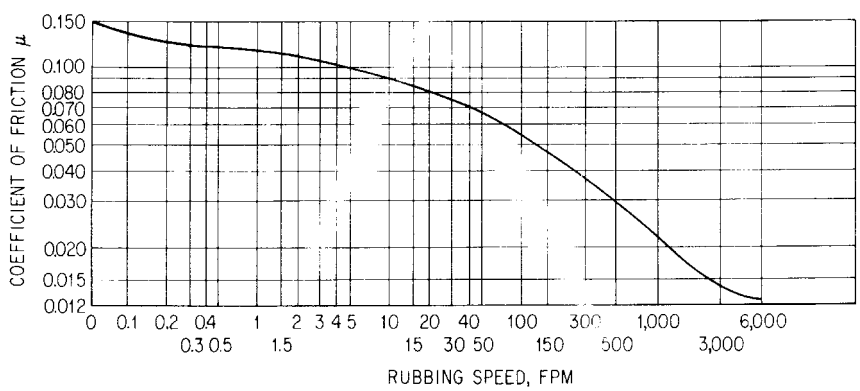


FIG. P-30 Coefficient-of-friction curve. (Source: Demag Delaval.)

(See Fig. P-21 for a definition of lead angle.) With this rubbing speed noted at the bottom of the diagram, read vertically upward until you intersect the coefficient-of-friction curve. Read the value of the coefficient of friction from the left-hand side of the diagram.

When the worm is the driver, enter the efficiency diagram with the lead angle of the worm at the bottom of the diagram. Read upward to the intersection of the curve with the correct coefficient of friction. The efficiency of the gearset may be read from the right-hand side of the diagram or the efficiency loss on the left-hand side of the diagram.

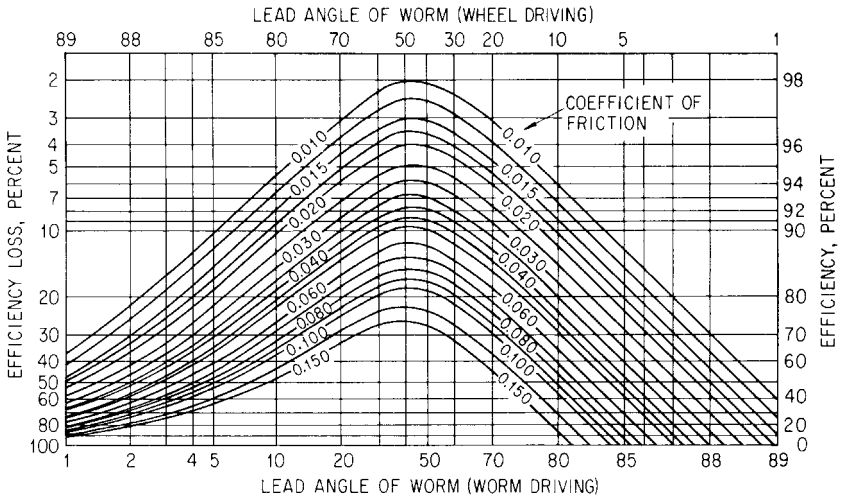


FIG. P-31 Efficiency diagram for worm gearing. (Source: Demag Delaval.)

When the gear is the driver, enter the efficiency diagram with the lead angle of the worm at the top of the diagram, reading down to the curve with the correct coefficient of friction. Find the efficiency as before.

These efficiencies, while approximate, are very close to the operating efficiency of the gearset alone. When the gearset is enclosed in a housing with bearings, seals, and oil reservoir, some allowance must be made for bearing loss, seal drag on the shaft, and churning of oil.

Self-locking gearset. A self-locking gearset is one that cannot be started in motion by applying power at the gear. Theoretically, this can be obtained when the lead angle of the worm is less than the friction angle. For normal static conditions the friction angle would be approximately 8°30′, and therefore it might be deduced that gearsets having a worm lead angle less than this value would be self-locking.

However, it is impossible to determine the point of positive self-locking for several reasons. The value of the static coefficient of friction varies considerably because of the effect of a number of variables. Furthermore, if a source of vibration is located near a self-locked gearset, a very slight motion might occur at the gear contact. Since the coefficient of friction decreases rapidly with an increase in rubbing velocity from the static condition, the friction angle may become smaller than the lead angle. Once this occurs, motion will continue and the gearing will accelerate under the action of the power applied to the gear.

Figure P-32 indicates the rapid increase in efficiency with increase in rubbing speed from the static condition for both the worm driving and the gear driving. For this particular example at a rubbing velocity of 500 ft/min, there are only a few points of efficiency difference between the two curves.

The best way to obtain locking is to use a brake, released electrically when the motor is started. With worm gears of high ratios, the braking effect need be only a fraction of full-load motor torque. A solenoid brake is usually best suited for this operation since the braking effect may be adjusted by weights that can be proportioned to stop the load gradually and avoid damage. Dashpots can be employed to ensure gradual setting of the brake.

Tooth form. The tooth form used by this information source is the involute helicoid. Figures P-33 and P-34 show the straight generating line tangent to the base circle and the convex axial section of thread.

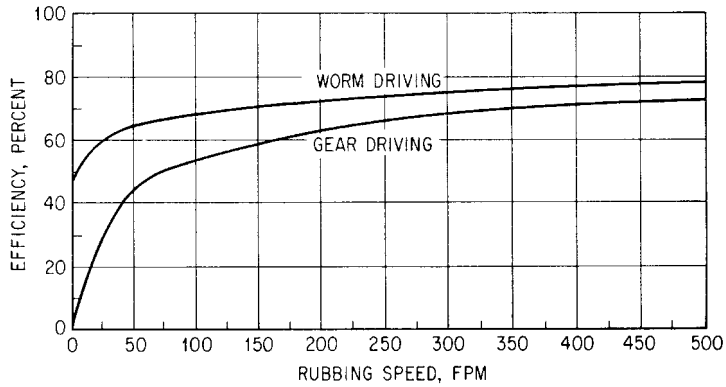


FIG. P-32 Comparison of efficiencies at tooth contact (ratio 50 on 20-in-center distance). (Source: Demag Delaval.)

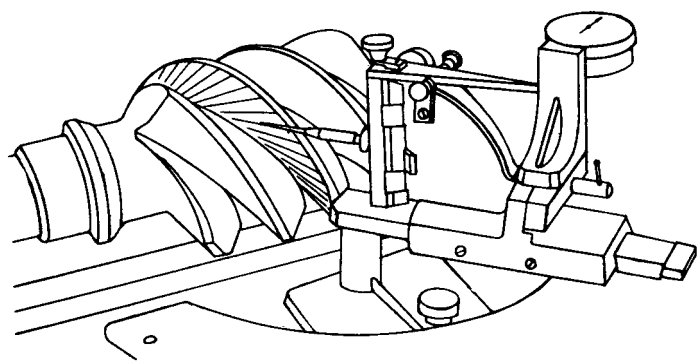


FIG. P-33 Generation of tooth form. (Source: Demag Delaval.)

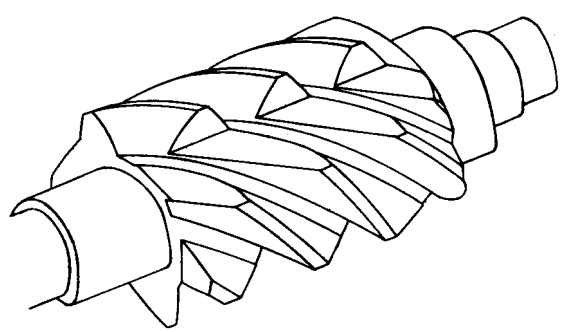


FIG. P-34 Convex axial section of thread. (Source: Demag Delaval.)

Worm-gear performance is judged in terms of load capacity, smooth, silent running, and high efficiency. The attainment of these goals requires accurate methods of producing and inspecting the worm and gear.

Since the involute helicoid worm is based on generation of a straight line tangent to the base circle, the accuracy of this line is very simple to check (Fig. P-35). This thread form lends itself to accurate manufacture, inspection, and interchangeability, as all worms can be checked to calculated measurable dimensions.

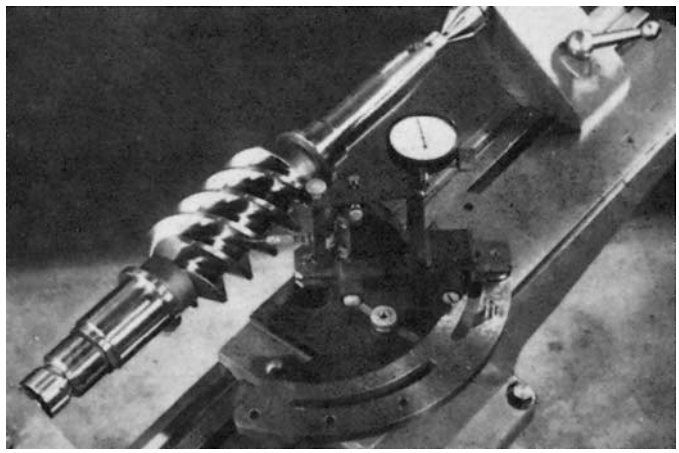


FIG. P-35 Inspection of tooth form. (Source: Demag Delaval.)

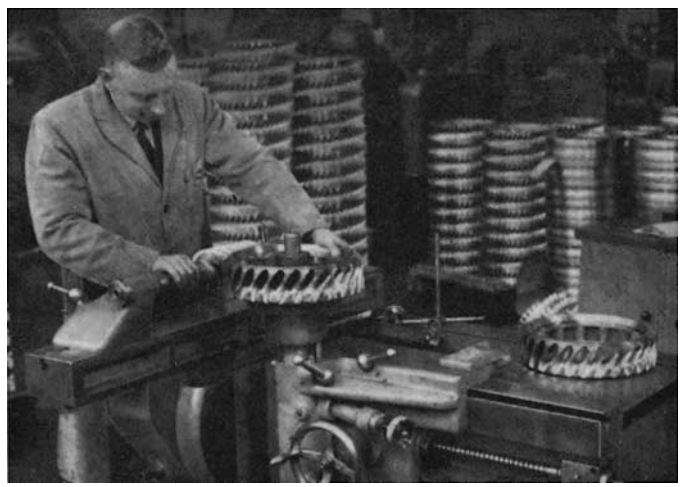


FIG. P-36 Checking with master worm. (Source: Demag Delaval.)

All wheels are checked with a master worm to ensure interchangeability and correctness of form (Fig. P-36).

Tooth contact. The involute helicoid thread form is a calculated form, and the theoretical contact is maintained more accurately and is more easily determined than that of any other worm thread, particularly a concave thread flank.

Figure P-37 shows theoretical "lines" of contact that exist between two worm threads and two gear teeth at a given angular position of the worm. As the worm rotates in the direction shown, these contact lines move progressively across the flanks of the worm and gear teeth and are inclined at an angle to the direction of sliding. This inclined effect is known to give a highly efficient form of surface lubrication and a low coefficient of friction as compared with a gear form in which the lines of contact are in the approximate direction of sliding. The contacting surfaces are always freshly lubricated and are not subject to the undesirable effects of double contact.

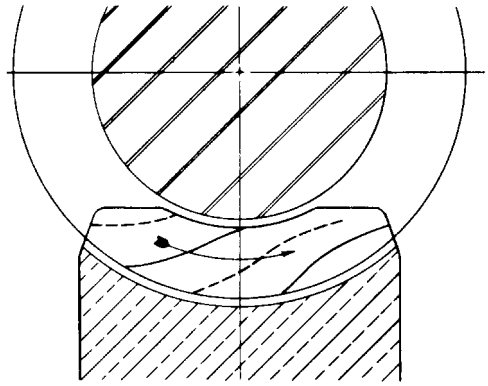


FIG. P-37 Tooth contact. (Source: Demag Delaval.)

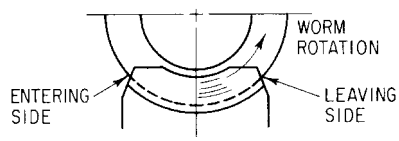


FIG. P-38 Tooth contact: good. (Source: Demag Delaval.)

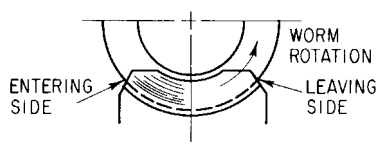


FIG. P-39 Tooth contact: poor. (Source: Demag Delaval.)

Depending on the relative radii of curvature between the two contacting surfaces and the load applied, these lines of contact actually have some width, thereby providing area contact. In spite of claims to full area contact, line contact occurs on all other thread forms including the double-enveloping thread form. Only the involute helicoid thread form provides the necessary control of the geometry of thread form in design and manufacture to obtain optimum contact conditions.

All gears, bearings, and housings deflect and distort to some extent when operating under load as compared with conditions under no load. A correction in the tooth-contact pattern is provided to ensure proper contact under loaded conditions. This correction is accomplished by producing gears with leaving-side contact as shown in Fig. P-38. This is the ideal contact pattern to aim for when assembling a worm gearset under a no-load condition. This contact pattern allows a lubricant-entry gap in tooth contact. When the gear deflects under load, the contact tends to move to a more central position on the bronze gear face, still allowing a lubricant-entry gap.

A contact pattern such as that shown in Fig. P-39 is the worst possible contact pattern under a no-load condition. This contact does not allow a lubricant-entry gap, and deflection under load will aggravate this condition. A gearset mounted in this manner may cause a temperature rise in oil 20 percent higher than that of the same gearset mounted as shown in Fig. P-38. The remedy is to move the gear axially

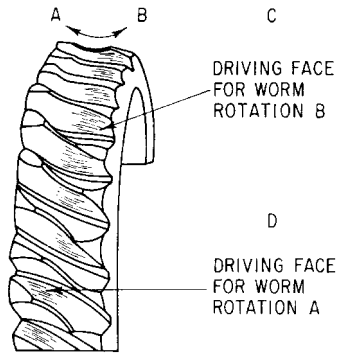


FIG. P-40 Driving face for worm notation. (Source: Demag Delaval.)

TABLE P-8 Minimum Recommended Number of Gear Teeth for General Design

Center Distance, in.	Minimum Number of Teeth
2	20
3	25
5	27
10	29
14	35
20	40
24	45

to the left (adjusting by shims or other adjustments provided) until a contact similar to that of Fig. P-38 is obtained.

When assembling worm gears that will run in both directions of rotation, it is necessary to consider both driving faces of the gear and to aim for contact as shown in Fig. P-40. When the worm is rotating in direction *A* in Fig. P-40, contact should be at *D* on the leaving side. When the worm is rotating in direction *B* in Fig. P-40, contact should be at *C* on the leaving side. For gears that will run in one direction only, it is necessary to obtain a contact pattern that is correct for the driving-side flank of gear teeth only.

Assembly adjustment. The gear should be mounted approximately on the centerline of the worm. The worm threads should be coated with prussian-blue dye. A section of the gear teeth should be coated with an orange-colored lead paste. The worm and gear should be rotated in both directions of rotation by hand. The blue markings from the worm threads will show the contact against the orange coating on the gear teeth. If the contact pattern is not as desired, the gear should be adjusted axially until a correct pattern is obtained.

Design considerations for worm and gearset. It is assumed that at the start of this design sequence the center distance for this gearset is known. (See Table P-8.)

The maximum number of teeth selected will be governed by high ratios of reduction and consideration of strength and load-carrying capacity.

Number of threads in worm. The minimum number of teeth in the gear and the reduction ratio will determine the number of threads for the worm. Generally 1 to 10 threads are used.

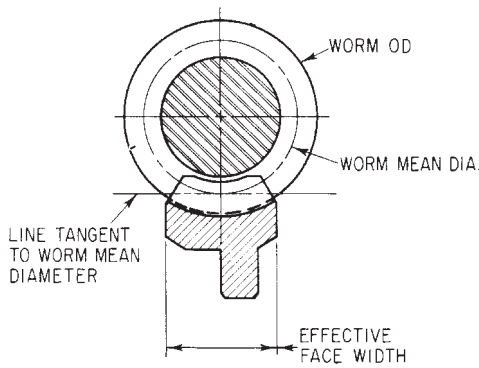


FIG. P-41 Gear-face width. (Source: Demag Delaval.)

Gear ratio

$$\text{Gear ratio} = \frac{\text{number of teeth in gear}}{\text{number of threads of worm}}$$

Pitch. Axial pitch of worm = circular pitch of gear. Keep the fraction simple so that accurate factoring can be used to determine change gears.

Worm pitch diameter. The pitch diameter of the worm is assumed to be at the mean working depth of the worm thread. The following factors should be considered when selecting worm pitch diameter:

1. Smaller pitch diameters provide higher efficiency and reduce the magnitude of the tooth loading.
2. The root diameter that results from pitch-diameter selection must be sufficiently large to prevent undue deflection and stress under load.
3. For low ratios the minimum pitch diameter is governed by the desirability of avoiding too high a lead angle. Lead angles up to 50° are practical.

Gear pitch diameter

$$\text{Gear pitch diameter} = 2 \times \text{center} - \text{pitch diameter of worm}$$

Recommended pressure angle. For general usage, pressure angles from 20 to 25° are common. Smaller values of pressure angle decrease the separating force, extend the line of action, making the amount of backlash less sensitive to change in center distance, and are used in index gearing. Larger values of pressure angle provide stronger gear teeth and assist in preventing undercutting of teeth with large lead angle. They are used in extremely heavily loaded applications.

Gear-face width (Fig. P-41). Maximum effective face width is the length of a line tangent to the mean worm diameter, to a point at which the outside diameter of the worm intersects the gear face. Any face width larger than this effective face width is of very little value and is wasteful of material.

Gear-throat diameter = gear pitch diameter + 2 × gear addenda. Gear outside diameter = gear throat diameter + 1 addendum of worm rounded off to the nearest fraction of an inch.

Gear blank under rim diameter (Fig. P-42)

$$h_t = \text{tooth depth of gear}$$

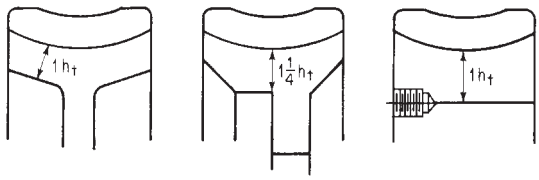


FIG. P-42 Gear-blank shapes. (Source: Demag Delaval.)

TABLE P-9 Bearing Loads

Resulting from	Bearing No. 1	Bearing No. 2	Bearing No. 3	Bearing No. 4
P	$Pa/L_W = P_1$	$Pb/L_W = P_2$	$Pr_G/L_G = U_3$	$Pr_G/L_G = U_4$
S	$Sa/L_W = S_1$	$Sb/L_W = S_2$	$Sd/L_G = S_3$	$Sc/L_G = S_4$
T	$Tr_W/L_W = U_1$	$Tr_W/L_W = U_2$	$Td/L_G = T_3$	$Tc/L_G = T_4$
Radial load	$\sqrt{P_1^2 + (S_1 - U_1)^2} = R_1$	$\sqrt{P_2^2 + (S_2 - U_2)^2} = R_2$	$\sqrt{T_3^2 + (U_3 - S_3)^2} = R_3$	$\sqrt{T_4^2 + (S_4 - U_4)^2} = R_4$
Thrust load	. . . . .	T	. . . . .	P

Underrim dimension for bronze gear block

= gear-root diameter – 2 to $2^1/_2 \times$ gear-tooth depth

Worm face

Minimum worm face

$$= z\sqrt{(\text{gear-throat diameter}/2)^2 - (\text{gear pitch diameter}/2 - \text{gear addendum})^2}$$

Allowable shaft stressed. All shafting in accord with AGMA Practice 260.01, March 1953.

Allowable bolt stressed. All bolts in accord with AGMA Practice 255.02, November 1964.

Bearing loading. All bearings selected in accord with AGMA Practice 265.01, March 1953. (See Figs. P-43 and P-44.)

Ball and roller bearings are selected on the basis of supporting loads equal to the maximum basic rating of the gear reducer and allow a minimum bearing life of 5000 h or an average life of 25,000 h. (See Table P-9.)

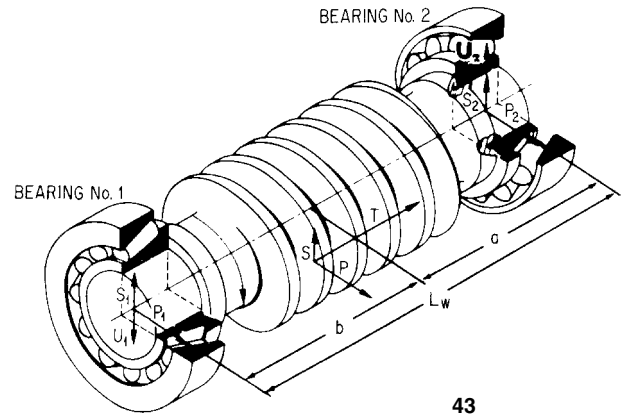
Performance

Mechanical ratings of cylindrical worm gears. The practice for this rating follows AGMA Practice 440.03, September 1959.

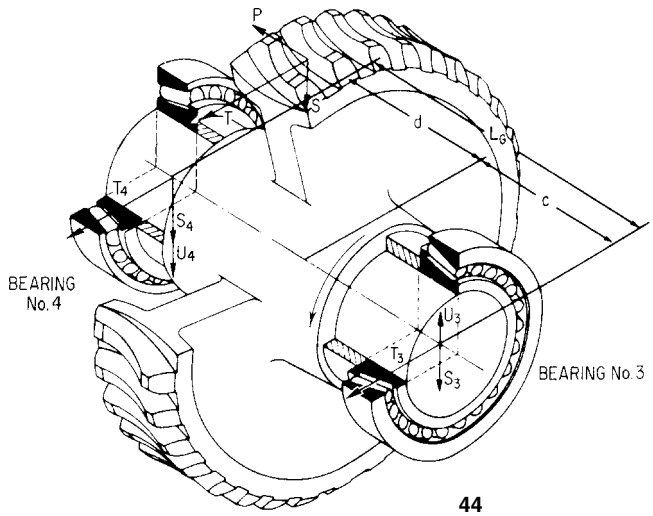
The ratings that are cataloged according to this practice are *wear* ratings that the gearset will satisfactorily permit, at the load shown, provided the driven machine has a uniform load requirement free of shock loading, 10 h/day. This is the basic rating by which worm-gear drives are selected, subject to thermal limitations.

Service factors are applied to this basic rating to factor the wear rating for shock loading or intermittent service.

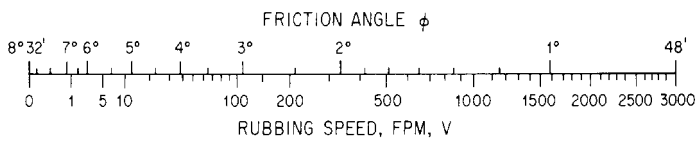
Thermal ratings of cylindrical worm gears. Thermal ratings above 100- to 200-rpm worm speed represent the input horsepower and output torque that will provide a stabilized 100°F oil-temperature rise over ambient air temperature when the machine is operated continuously. For example, if the ambient air temperature is 70°F, a reducer carrying rated thermal horsepower will operate with an average oil



43



44



Principal forces and bearing loads in a worm (FIG. P-43) and gearset (FIG. P-44). D_w = pitch diameter of worm, in; r_w = pitch radius of worm, in; r_G = pitch radius of gear, in; γ = lead angle of worm, °; P = tangential force on worm, lb; Q = torque input to worm, in · lb; S = separating force, lb; T = axial thrust of worm, lb; NPA = normal-pressure angle: ϕ = friction angle for worm driving; rpm (r/min) = worm speed; V = rubbing speed, fpm (f/min); $P = Q/r_w$; $S = P$ tan NPA/sin($\gamma + \phi$); $T = P$/tan($\gamma + \phi$); $V = 0.262\ D_w$ rpm/cos γ. (Source: Timken Roller Bearing Company.)

temperature of 170°F. Since normal worm-gear lubricants will deteriorate rapidly, require frequent replacement, and may not support the gearmesh loads when the machine is operating continuously at 210 to 220°F, the practical maximum ambient air temperature for worm-gear reducers carrying full thermal rating horsepower is 100°F.

For operation at higher ambient-air temperatures, a larger unit with a higher thermal rating must be selected for continuous operation, or a cooling system must be employed. For example, if a unit is to operate in an ambient-air temperature of 150°F, the increase in oil temperature must be limited to 50°F to keep the oil

TABLE P-10 Basic Lubricant Recommendations (AGMA)

	Size 60 Units and Smaller, Ambient Temp., °F		Size 70 Units and Larger, Ambient Temp., °F	
Worm Speed, rpm	15–60	50–125	30–60	50–125
Up to 400	7 Comp.	8 Comp.	7 Comp.	8 Comp.
Above 400			7 Comp.	7 Comp.

temperature from rising above 200°F. This means that the heat generated in the reducer must be one-half of the heat generated when the machine is operating at the catalog thermal rating; or since bearing and oil losses remain constant for a given speed, applied horsepower must be less than one-half of the catalog thermal rating.

For operation at ambients of less than a maximum of 100°F or when artificial or natural air drafts are present, catalog thermal ratings can be exceeded. For a proper evaluation, all data on ambient conditions should be determined.

Allowable starting load. Worm-gear reducers have a momentary overload strength rating + 300 percent of mechanical-wear rating. Peak starting load of the driven machine should not exceed 300 percent of the mechanical-wear rating.

Lubrication

General. Because of the nature of worm-gear sliding and rolling action, lubricants used for other types of gearing are not satisfactory. All units are shipped without oil, but reducer instructions and lubrication nameplates refer to the use of AGMA lubricants. Generally speaking, suppliers of *industrial lubricants*, *not* service stations, should be contacted and should be able to supply suitable lubricants from stock to meet these AGMA specifications (Table P-10). The units should be filled with the proper lubricant before operating.

These lubricants are basically a steam-cylinder oil. A list of trade names of the various manufacturers of oils that meet the AGMA 7 Compounded and AGMA 8 Compounded specifications is maintained by Delaval. These lubricants are basically petroleum-base oils but with 4 to 5 percent acidless tallow additives that provide additional film strength. They are heavy oils, much heavier than normal motor oils. The viscosity of AGMA 7 Compounded is approximately 135 SSU at 210°F, and that of AGMA 8 Compounded is approximately 150 SSU at 210°F. This heavy viscosity plus the plating action of the additives on the worm and gear contact surfaces is required to ensure the long trouble-free life that the gearing is designed to provide.

Lubricants not recommended. The following lubricants should never be used for worm gearing:

1. Ordinary motor oils, no matter what their viscosity.
2. Automotive rear-end oils.
3. Extreme-pressure lubricants containing compounds of sulfur or phosphorus. It may be claimed that these lubricants are noncorrosive to steel, but they are extremely corrosive to bronze and will not provide the necessary plating action required.
4. Greases of any kind. These do not flow sufficiently to provide the necessary cooling.

TABLE P-11 Cold-Weather Lubrication

For Min. Ambient Temp., °F	Use a Mild Extreme-Pressure Oil Containing Lead Naphthanate and Having a Viscosity of
0	120 SSU at 210°F
–10	100 SSU at 210°F
–20	75 SSU at 210°F
–30	53 SSU at 210°F

Cold-weather lubricants. If ambient temperatures below 15°F are expected, a winter, or cold-weather, lubricant must be selected, since the AGMA 7 Compounded or 8 Compounded will solidify and the motion of the gears will channel the solidified oil until no lubricant is present at the gear mesh. For this condition, a minimum ambient temperature to be expected must be estimated and a reputable supplier consulted to recommend an oil with a channel point well below the expected minimum ambient temperature. This will require a lighter-viscosity oil, but the oil should still contain additives. The best selection is usually the mild extreme-pressure oils containing lead naphthanate with the viscosities shown in Table P-11. The lubricant should be changed to the heavier-viscosity oils when the ambient temperature again goes above 15°F.

Frequency of oil changes. The frequency of oil changes varies with the type of service. After the initial 50 to 100 h of running, a change should normally be made to remove the particles of bronze burnished off the gear during the run-in period. Thereafter, a general rule is that the oil should be changed every 6 months of normal service and every 3 months of severe service. However, if the unit is in a dusty or moist atmosphere, dirt or water accumulation in the oil reservoir may require more frequent changes. Many oil suppliers will test a lubricant after a period of use free of charge and determine its useful life for a specific application.

Procedure for long shutdown periods. If the unit is to be idle for any length of time, particularly outdoors, something must be done to prevent rusting of the bearings, gears, and other internal parts. The easiest solution is usually to fill the unit completely with clean oil. Of course, before the unit is started again, the oil should be drained and refilled to its proper level.

Installation and operation

Installation. Normal good practice must be followed when handling the unit, choosing a foundation, checking alignment, and mounting couplings, pulleys, gears, sprockets, etc. Couplings should be pressed or shrunk on the reducer shafts. Do not drive couplings on shafts, as this may damage the bearings and also cause the shafts to spring. This, in turn, may result in failure of the bearings, vibration, and oil leakage. Sprockets, pulleys, and pinions should be mounted as close to the case as possible in order to avoid undue bearing-load and shaft deflection.

Operation. The unit is shipped from the factory *without oil* but is slushed internally with a rust-preventive compound, which need not be removed because it is oil-soluble. *Make certain that the reducer is filled to the correct level before start of operation in accordance with lubrication specifications*. The unit must be filled to, but not above, the oil-level gauge. The oil level will, of course, change with the mounting arrangement. It should be checked periodically and only at a time when the unit is not operating. A dipstick is provided in the oil-level gauge.

All units are subjected to test before shipment, but it takes additional hours of running under full gear load to attain highest efficiency. The gear may, if necessary, be put to work immediately on full load, but if circumstances permit, it is better for the ultimate life of the gear to run it under gradually increasing load. Immediate application of full load concentrates high unit pressures on tooth surfaces. When new driven equipment requires operation to achieve freedom and minimum friction loss, use precaution in the early stages of operation to prevent the reducer from taking an overload. When overload tests are specified on a machine before it is shipped, it is better to make preliminary runs under part load before building up to full load and overload. A reasonable running-in procedure is half load for a few hours, building up to full load, in two stages if possible.

Temperature rise on the initial run will be higher than that eventually attained after the gear is fully run in.

Some slight wear and/or pitting of the bronze gear teeth may be observed after a short period of initial operation. This condition is normal, as some initial wear is necessary for the hardened-steel worm to seat itself properly with the bronze gear.

Product Application Case 1: High-Speed Gears—New Developments*

High-speed gears are gears operating at high pitch line velocities up to 240 m/s and high power. The requirement to transmit significant power at extreme speeds, i.e., to have high speed *and* load on the bearings and the toothing, led to some new developments for such components.

A back-to-back test bed has been installed for research purposes. The gears transmit a power of 30,000 kW at 6380/15,574 rpm nominal speed, with the capacity of 120 percent overspeed.

A new type of radial tilting pad bearing designed by this OEM will be discussed. The face width to diameter ratio is 1.4, which enables the bearing to take higher rotational speeds at reasonable specific loads. Further, a new design for a double or multiple tilting pad thrust bearing is explained. The bearing can be used in any kind of machinery that is exposed to high axial thrusts at very high speeds or where an utmost compact design is required due to limited space.

Compressors and other turbomachines are constantly developed and improved to run at higher speeds with more power. The gear manufacturer therefore has to design gears that can transmit high toothing forces at very high rotational speeds. But the toothing and the bearings are bound to certain limits for thermal and mechanical load that must not be exceeded for safe operation. These limits have been elevated over the last years by continuous development of toothing and bearings; see Fig. P-45 for the pitch line velocity.

To solve the basic design problem of high speed and power, there are two general possibilities: to design a gear with power split, i.e., with two or more power paths or to increase the limits for toothings and bearings by development of these components.

Gears with power split reduce the load on bearings and toothing and allow the designer to stay within well-known limits. On the other side, such gears have a more complex layout, they have more tooth meshes and more bearings and they need some mechanisms for a correct power split. These mechanisms can be quill shafts, self-adjusting bearings, or others. The costs to build such a complex gear must be almost twice as much as for a simple two-shaft gearbox.

---

* Source: MAAG Gear Company, Switzerland.

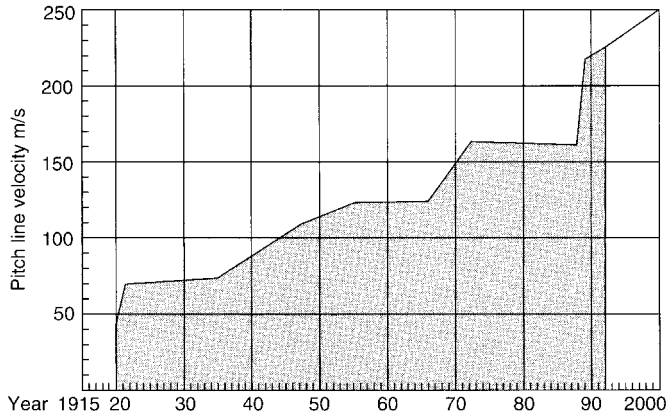


FIG. P-45 Development of toothing pitch line velocity since 1920. (Source: MAAG Gear Company.)

Therefore, the question is put: How can toothing and bearings be further developed in order to allow higher limits for load and speed? More power and speed could be transmitted in simple, reliable, and cost-effective two-shaft gear designs.

In 1985, this OEM started a research program to investigate bearings and toothings. Gears with high power and speed were designed to be carefully tested on a special back-to-back test bed. During the design phase, it was recognized that before the toothing, the pinion bearings reached their load limits. New radial and axial tilting pad bearings had to be developed for these gears to allow for safe operation with maximum white metal temperatures below 130°C. The design of these new bearings as well as the test results under full load are presented.

"Back-to-back" test bed

The mentioned back-to-back test bed is shown in Fig. P-46. It consists of two identical gearboxes that are mechanically coupled with a torque meter device on the low-speed shafts and with a special highest speed toothed coupling on the high-speed shafts.

With the single helical toothing, the gears can be loaded up to full load just by applying an axial force on the wheel. The axial shifting causes a rotary movement and the wanted closed torque circuit between the two gears is established. The axial force is produced by means of hydraulic pistons. With a variable hydraulic pressure on these pistons, any desirable load between zero and full load can be achieved. As driving power, only the total losses of the two gears have to be provided.

Some technical data of the back-to-back test bed:

Nominal power: P = 30,000 kW

Nominal speed: n = 6380/15,574 rpm

Overspeed 120%: n = 7656/18,689 rpm

Nominal pitch line velocity: v = 200 m/s

Overspeed plv.: v = 240 m/s

Center distance: a = 422 mm

Two sets of gearwheels are tested: one with a helix angle of 13°, the other with 19°. The rotors are equipped with strain gauges at the tooth root and with

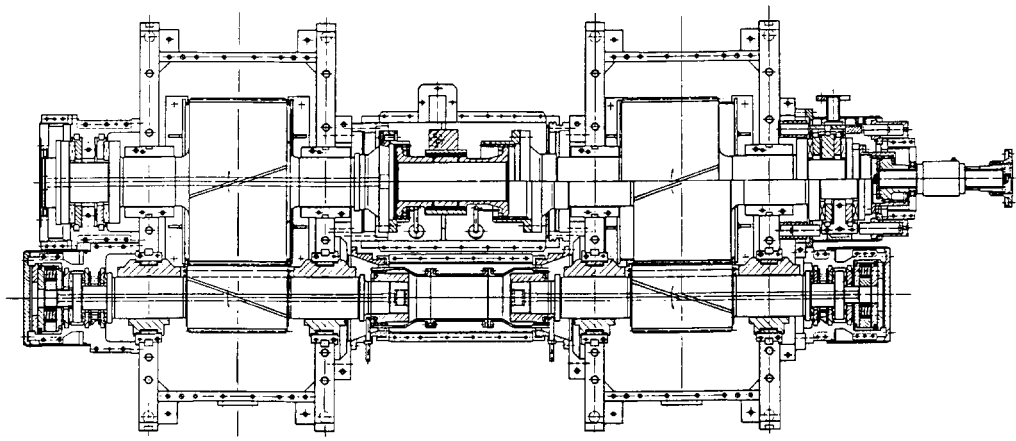


FIG. P-46 Back-to-back test bed. (Source: MAAG Gear Company.)

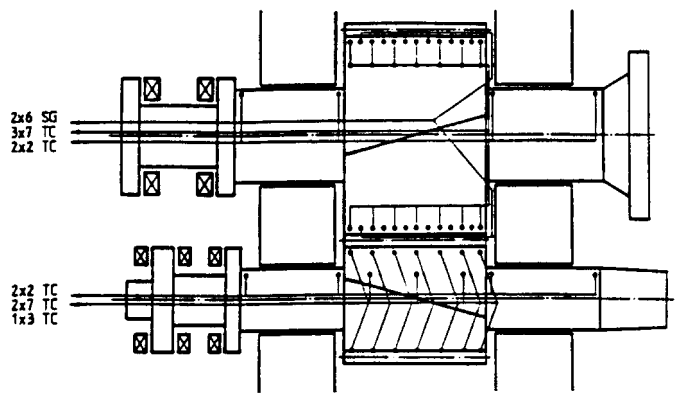


FIG. P-47 Instrumentation of test gear rotors. (Source: MAAG Gear Company.)

thermocouples near the toothing and journals. With these instruments, thermal distortions due to unequal temperature distribution and actual load distribution across the face width of the toothing are measured. The bearings are equipped with thermocouples in the hottest zones in order to determine maximum white metal temperatures and thermal deformation. Figure P-47 shows the instrumentation of the test gear rotors.

All shafts equipped with instruments are hollow to take all the cables which are connected via special high-speed slip rings to the static analysis instruments. Rotor vibrations are to be surveyed by pic-up's; all other instruments and sensors are as for a common industrial platform gear. The wheel of the slave gear is not equipped with sensors, but the hydraulic axial force device and the input motor drive are connected to it. At the input shaft, the total losses of both gears can be measured by means of a torque meter coupling.

The toothed coupling between the two pinions is a specially designed coupling for highest speeds. Its weight and overhang have been minimized in order to satisfy lateral critical speed requirements of the pinions. Without this coupling it would not have been possible to find a satisfactory solution for save operation at these speeds. Every other type of coupling, for example a disc coupling, has more weight and more overhang and is therefore only of limited use for extreme high-speed gears.

TABLE P-12 Comparison of Important Design Parameters of Conventional and New Bearing Design for Back-to-Back Gears

	Conventional	New Design
D (mm)	170	150
B/D	1.0	1.4
p (N/mm^2)	3.0	2.8
v (m/s)	139 (167)	122 (146)
T_{max} (°C)	132 (141)	121 (130)

Values in parentheses refer to overspeed 120%.

Due to thermal expansions and friction in the toothed coupling, additional axial forces will act on the gears. Earlier back-to-back tests showed that for the unloading of such a unit, it is not sufficient just to release the axial force. The gears had to be unloaded by applying an axial force in the reverse direction in order to overcome the friction in the toothed couplings. The friction in toothed couplings is existing and produces axial reaction forces that cannot be neglected.

This led to the knowledge that extreme high-speed gears, which must be equipped with toothed couplings for lateral critical speed reasons, should have a single helical toothing that is not affected by additional external thrusts.

On a double helical toothing, an external thrust would act on one helix only, which is considered a worst-case situation for the toothing.

Radial tilting pad bearings

Designs. For high-speed gears, white metal–lined slide bearings are commonly used. The known limits for such types of bearings are as follows:

- Specific load 3,2 . . . 4 N/mm^2
- Maximum white metal temperature 130°C
- With circumferential speeds above 90–100 m/s, tilting pad bearings should be used in order to avoid bearing instabilities due to oil whip

During the design phase of the back-to-back gears, it became evident that the pinion bearings could not be realized with a conventional design. A new design had to be found in order to keep the white metal temperatures within specified limits. The solution is a specially developed tilting pad bearing with the following design features:

- Ratio width/diameter for the main pad: 1:4
- Three pads, one main pad and two auxiliary pads
- The main pad has a circumferential groove in the center to evacuate the hot oil

Materials and fabrication methods are the same as for conventional bearings.

Table P-12 compares important design parameters of conventional and new bearing design for back-to-back gears. The general design of the bearing is shown in Fig. P-48. The circumferential groove is important for reduction of thermal deformations due to temperature gradients over the face width of the bearing.

Test results. The back-to-back gears have been run at full load and up to 120 percent speed for many hours. The bearings, as well as the toothing, have been

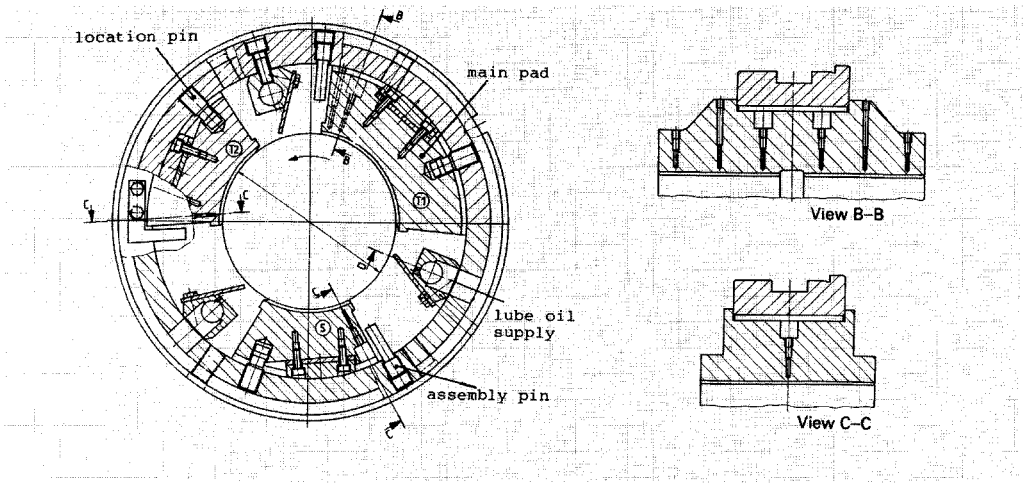


FIG. P-48 MAAG radial tilting pad bearing. (Source: MAAG Gear Company.)

operating very satisfactorily during the whole testing period. Careful inspection after test runs did not show any sign of wear or damage. This is not surprising since bearings and toothing are designed for infinite life.

The radial tilting pad bearings were equipped with thermocouples in the hottest area across the full width of the main pad (see Fig. P-49).

Measured maximum white metal temperatures are shown in Fig. P-50.

An analysis of the measured white metal temperatures and other test results leads to the following conclusions:

- The maximum values have always been lower than 125°C, at full load and full speed
- Measured values are very close to the calculated mean temperature of 121°C
- The circumferential groove stabilizes the temperature gradient at a lower level
- At overspeed 120 percent, i.e., with a circumferential speed of 146 m/s, no signs of oil whip occurred; the stability of the proposed tilting pad bearings is, as expected, very good
- Pinion lateral vibrations have always been below 1.1 mils, even at no load and overspeed; the lateral vibrations behavior of pinion and toothed coupling is good, the damping of the bearings is satisfactory

Deformation analysis. An important question for the development of a wide bearing is the heat distribution across its width and the resulting deformations. If these deformations are too large, the outer parties of the bearing will not take significant load and the heat will be concentrated in the center. This will result in overload for the bearing and consequently lead to damage. The optimal compromise for the B/D ratio must be found. After theoretical and experimental investigations, a 1.4 ratio was determined. With finite-element analysis, the deformations of the bearing were calculated. The stationary temperature distribution in the steel body of the bearing and the pressure distribution in the oil film are acted as loads on the main pad. Figure P-51 summarizes in a compromised sketch the complete results of the FE-analysis.

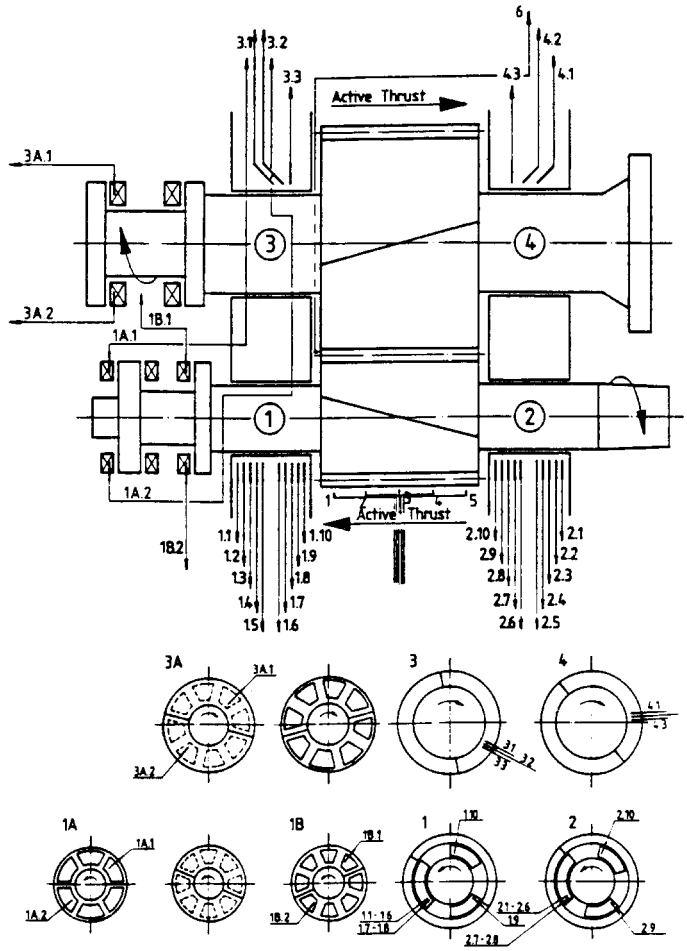


FIG. P-49 Instrumentation on bearings. (Source: MAAG Gear Company.)

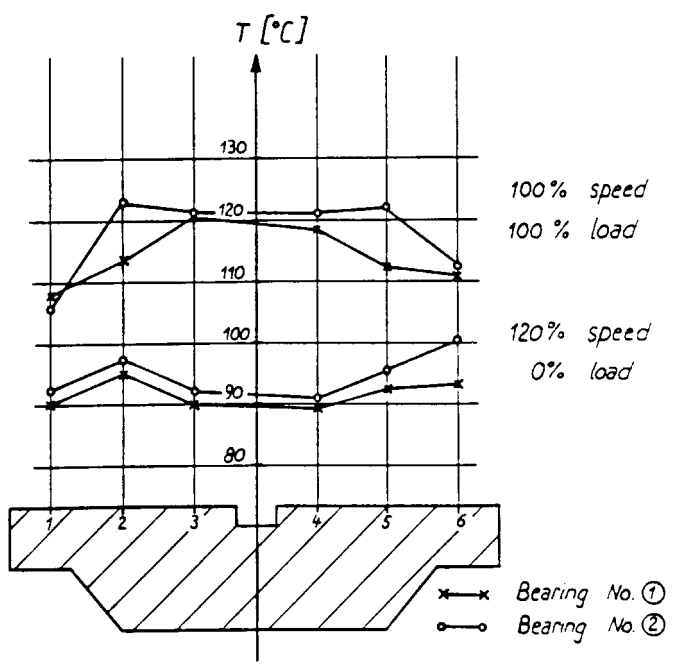


FIG. P-50 Measured white metal temperatures. (Source: MAAG Gear Company.)

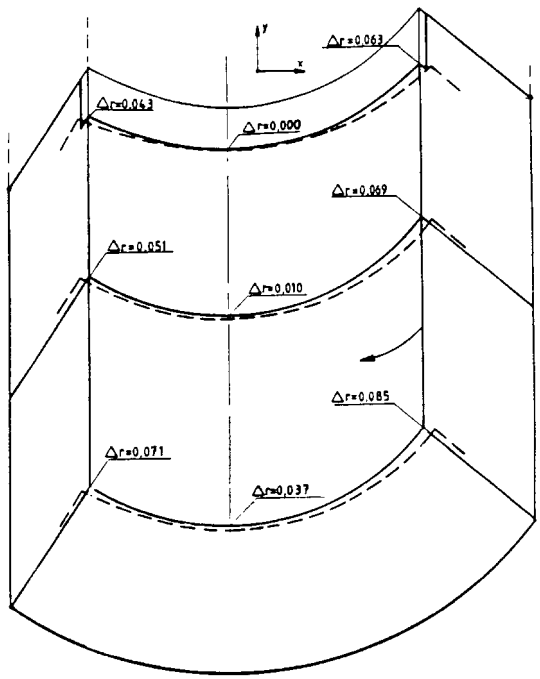


FIG. P-51 Finite-element-analysis radial deformations under thermal and mechanical load. (Source: MAAG Gear Company.)

The following conclusions can be drawn:

- 80 percent of the combined deformation is caused by unequal temperature distribution and only 20 percent by mechanical bearing load. It is, for this reason, of utmost importance to keep the temperature in the bearing low and uniform.
- The maximum deformation at the outer end of the bearing is approximately 0.08 mm. Supposing a minimum oil film thickness of 0.03 to 0.04 mm under load, it is evident that even the outer areas of the main pad will contribute to take load.

Axial tilting pad bearings

Design. High-speed gears are often designed with a single helical toothing, i.e., toothing thrust is to be compensated. In addition, external thrust from couplings can act on the high-speed shaft. The helix angle of the toothing must be high enough to limit the heat generation in the gear mesh. Thus, the question is how to absorb high axial forces at high speeds. The problem cannot be solved with a single axial tilting pad bearing or by shrunk-on trust collars because of load limits and centrifugal forces.

For the described back-to-back gears, two axial bearings have been arranged in series. It was possible to find a mechanism that allows for any power split between several axial bearings. These mechanisms are described in detail in the international patent description.

The basic principle is explained with Fig. P-52: On the upper half, the unloaded axial bearings are shown; below is the same arrangement under axial thrust. Each axial bearing (5) is supported by a ring (6) that rests on an arrangement for load adjustment (7, 8). These arrangements are supported by a rigid casing (9) that is connected to the main (gear) casing. The arrangement for load adjustment consists

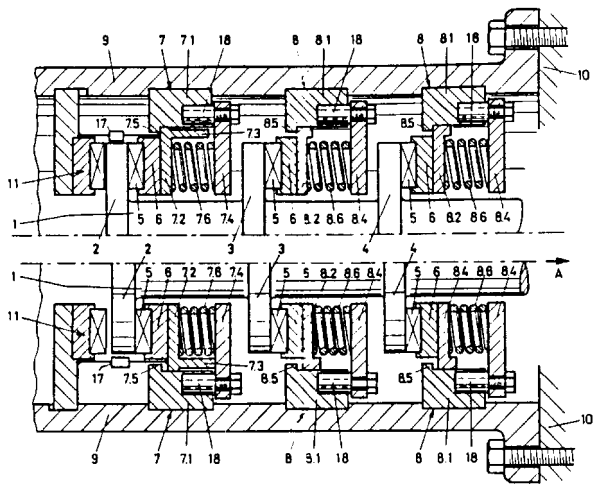


FIG. P-52 Multiple thrust bearing principle. (Source: MAAG Gear Company.)

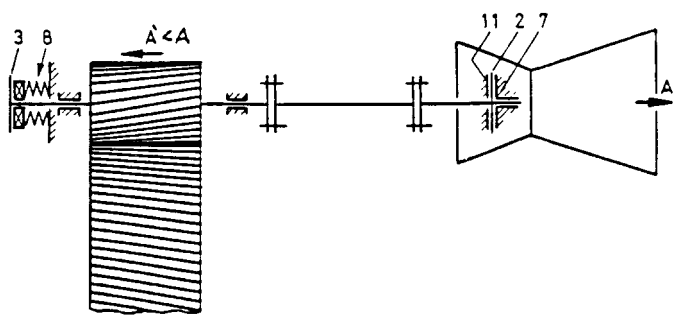


FIG. P-53 Thrust splitting between gas turbine and gearbox. (Source: MAAG Gear Company.)

of movable pistons that are preloaded by compressed springs. The preload for these springs can be chosen according to the load limit of the used axial bearing or to the preferred stage load for certain operating conditions. The piston of the last arrangement (7) has a stop (7.3) that limits the total axial mobility. The axial bearing clearances of the various stages are different; they increase from the inner to the outer bearings. With increasing axial thrust, the inner bearing starts to take load first until its spring preload limit is reached. An axial movement takes place until the second bearing has load and so on. With full axial thrust, each bearing has load according to the predetermined load sharing.

There are many different variations of the described basic principle. A very interesting one is the thrust split between rigidly coupled machines, for example, in the Fig. P-53 turbine and gearbox. Here, the toothing thrust helps to unload the turbine's axial thrust bearing. In addition, a well-defined part of the total thrust is absorbed with a bearing in the gearbox, without hindering the shafts to expand thermally. The bearing clearances have to be adjusted in a way that the gear's axial bearing is loaded even with maximum shaft expansion. The stationary bearing (7) can be positioned at the cold end of the turbine that helps to keep the thermal expansions to a minimum.

The described arrangement for axial bearings can be used not only for gears but principally for any rotating machines where high thrusts at high speeds must be

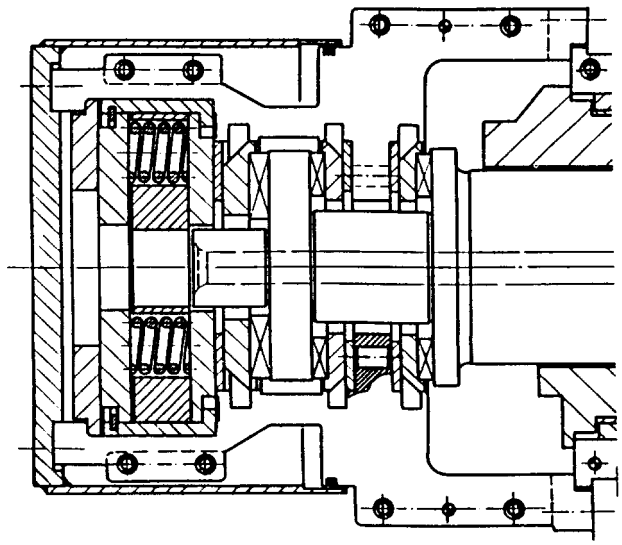


FIG. P-54 Double thrust bearing design of test gear. (Source: MAAG Gear Company.)

absorbed. It is of further interest that with a correct design, the power losses of two small bearings in series are lower than of one big bearing alone.

Test results. For the pinions of described back-to-back gears, the axial bearings have been designed according to the load-sharing principle (see Fig. P-54).

The toothing thrust is acting outward and the outer bearing will be loaded first with increasing power transmission.

Technical data are as follows:

Pinion speed: 15,574 rpm

Helix angle: 19°

Total axial thrust: 81′000 N

Preload of springs: 50′000 N

Specific bearing loads: 3.5/2.35 N/mm^2

Circumferential bearing velocities: 137/144 m/s

Bearing power losses: 34/31 kW

It would not be possible to take such a high total thrust with a single bearing at this speed. For the case of torque reversing, a "back" bearing is installed. Oil supply and instrumentation with thermocouples in the pads are realized exactly the same way as for a conventional single bearing design. The natural frequency of the preloaded piston springs was chosen to be different from potential exiting frequencies.

The results from back-to-back testing can be summarized as follows:

- No wear, no running tracks
- Maximum white metal temperatures well within permissible limits, as calculated in advance
- No axial vibrations
- Free mobility of piston, no running tracks

From these results after many hours of operation, one can conclude that the axial load sharing according to the described principle works.

Summary and outlook

The design of high-speed gears with two shafts is limited by the bearings and the toothing. Often the bearing limits are reached before the toothing becomes critical.

New radial tilting pad bearings have been developed. With these bearings, the rotor speed can be further increased at the same bearing load. In order to absorb high axial thrust at high speed, a new principle has been developed that allows for several axial bearings arranged in series. The load sharing between the stages can be chosen to any desirable values.

Both bearing designs have been tested on a back-to-back test bed with pinion speeds of more than 17,000 rpm and power up to 30,000 kW. These tests have proven that both designs are ready for industrial application.

What is the possible increase in speed? Assuming a maximum white metal temperature of 130°C and load, specific load, and circumferential bearing velocity to be constant, one can find the following relation:

$$n \text{ (increased)} \approx \sqrt{1.4} \cdot n \approx 1{,}18 \cdot n;$$

where n = max. perm. speed with "conventional" bearings.

This means that with the same thermal load on the pinion bearings, the pinion speed can be increased by approximately 20 percent when tilting pad bearings of the new type are used. Consequently, the range of high-speed gears based on a cost-effective two-shaft design is considerably increased.

Synchronous Clutch Couplings and Applications*

Synchronous clutch couplings (see Fig. P-55) are required today in a wide range of applications. These include, but do not exclusively consist of, applications in the following.

Power generation

- Alternator drives
- Peaking power stations
- Air storage power stations

Energy recovery, combined cycle technologies, cogeneration and others

- Connecting expander turbines to main drives in petrochemical plants
- Blower drives in nuclear power stations used during starting sequence
- Starting device for gas turbines
- Automatic turning gears

Synchronous clutch couplings are couplings that engage and disengage automatically. They are capable of engaging automatically at any speed within the operating range as soon as the driving machine overruns the driven machine. Basically, the synchronous clutch coupling is a disengagable coupling equipped with a mechanism (the synchronizing mechanism) that detects synchronism of both shafts and initiates the engaging movement.

The synchronous clutch coupling consists of two main elements:

---

* Source: MAAG Gear Company, Switzerland.

From these results after many hours of operation, one can conclude that the axial load sharing according to the described principle works.

Summary and outlook

The design of high-speed gears with two shafts is limited by the bearings and the toothing. Often the bearing limits are reached before the toothing becomes critical.

New radial tilting pad bearings have been developed. With these bearings, the rotor speed can be further increased at the same bearing load. In order to absorb high axial thrust at high speed, a new principle has been developed that allows for several axial bearings arranged in series. The load sharing between the stages can be chosen to any desirable values.

Both bearing designs have been tested on a back-to-back test bed with pinion speeds of more than 17,000 rpm and power up to 30,000 kW. These tests have proven that both designs are ready for industrial application.

What is the possible increase in speed? Assuming a maximum white metal temperature of 130°C and load, specific load, and circumferential bearing velocity to be constant, one can find the following relation:

$$n \text{ (increased)} \approx \sqrt{1.4} \cdot n \approx 1{,}18 \cdot n;$$

where n = max. perm. speed with "conventional" bearings.

This means that with the same thermal load on the pinion bearings, the pinion speed can be increased by approximately 20 percent when tilting pad bearings of the new type are used. Consequently, the range of high-speed gears based on a cost-effective two-shaft design is considerably increased.

Synchronous Clutch Couplings and Applications*

Synchronous clutch couplings (see Fig. P-55) are required today in a wide range of applications. These include, but do not exclusively consist of, applications in the following.

Power generation

- Alternator drives
- Peaking power stations
- Air storage power stations

Energy recovery, combined cycle technologies, cogeneration and others

- Connecting expander turbines to main drives in petrochemical plants
- Blower drives in nuclear power stations used during starting sequence
- Starting device for gas turbines
- Automatic turning gears

Synchronous clutch couplings are couplings that engage and disengage automatically. They are capable of engaging automatically at any speed within the operating range as soon as the driving machine overruns the driven machine. Basically, the synchronous clutch coupling is a disengagable coupling equipped with a mechanism (the synchronizing mechanism) that detects synchronism of both shafts and initiates the engaging movement.

The synchronous clutch coupling consists of two main elements:

---

* Source: MAAG Gear Company, Switzerland.

A. The gear coupling part for the power transmission. This is the same as in a standard tooth coupling, i.e., the external coupling teeth are hardened and ground with longitudinal corrections allowing for angular misalignment.

B. The synchronizing mechanism to detect synchronism of both shafts. This synchronizing mechanism is an assembly consisting of a number of pawls and a multiple notched ratchet wheel that act like a free-wheel drive.

This OEM has three main product lines of synchronous clutch couplings. See Fig. P-56.

It is also possible to provide each product of the synchronous clutch couplings with very distinctive features. Figure P-57 gives some examples of combination possibilities of different features available for two of the designs, Type MS and Type HS.

These features make synchronous clutches well suited for:

- Power generation
- Energy recovery, combined cycle technologies, cogeneration
- Other applications, e.g., fan drives, starter drives, turning gears
- Marine applications

Based on the foregoing, the designation of a particular type of synchronous clutch coupling gives the following information:

basic design:
M = Mechanical automatic disengagement
H = Hydraulic disengagement
S = Synchronous clutch coupling

size of coupling

M S – 26 – R . . . . . .

additional features
(see Fig. P-57)

H S – 60/7 – H . . . . . .

size of
coupling/synchronizer

See also Figs. P-56 and P-57.

Power generation

Synchronous clutch coupling (see Figs. P-58 and P-59) of the type HS-85 (the information source's model designation) can be installed in a power plant between the gas turbine and alternator.

The alternator remains on line all the time. If power is needed, the gas turbine is started by its starting system and further accelerated to full speed. When the gas turbine shaft overruns the alternator shaft, the clutch engages, allowing power transmission. When power is no longer required, the gas turbine power is reduced to approximately zero and the disengaging signal is given to the clutch, which

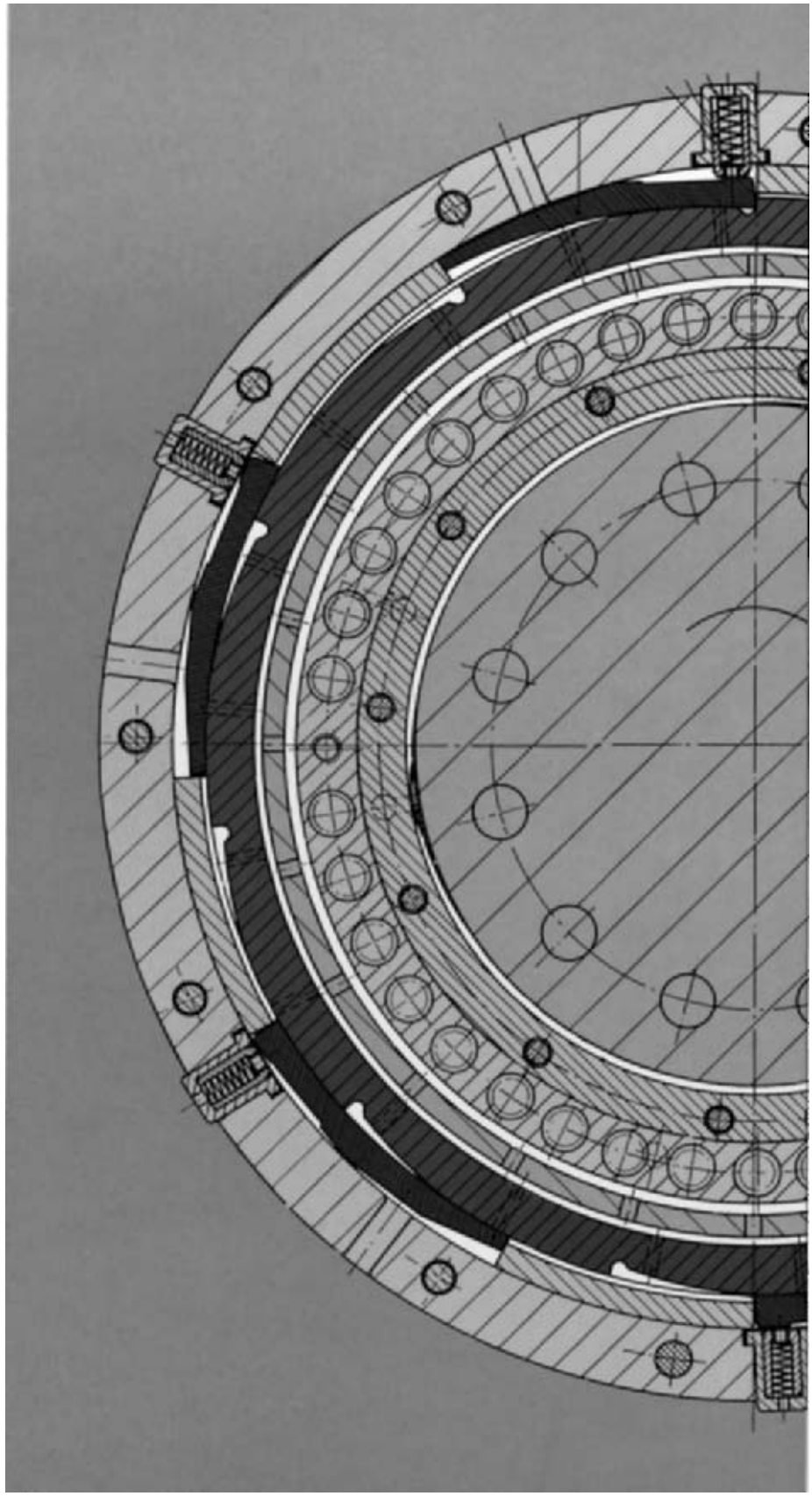


FIG. P-55 Synchronous clutch coupling section. (Source: MAAG Gear Company.)

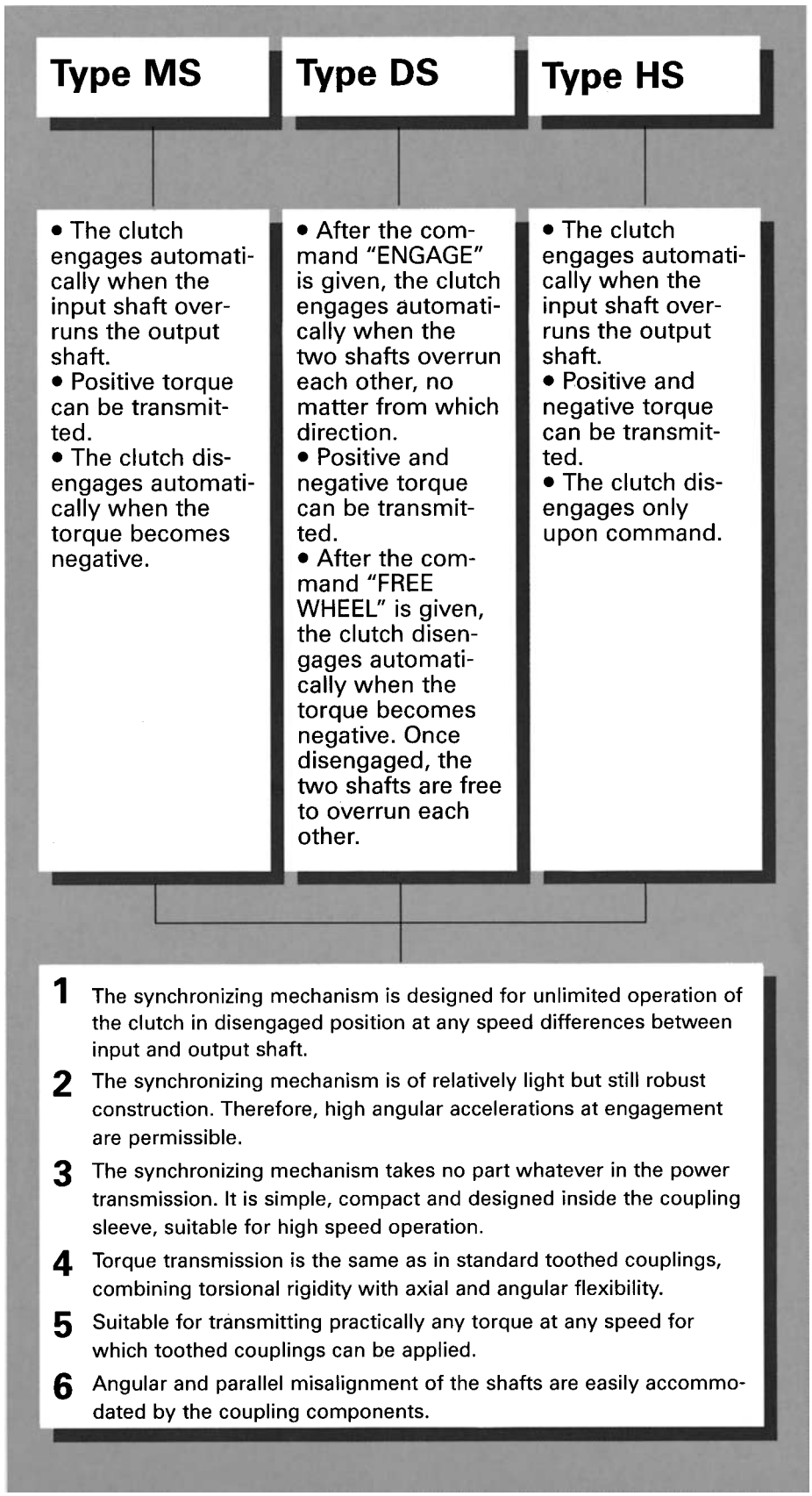


FIG. P-56 Synchronous clutch couplings (generic models). (Source: MAAG Gear Company.)

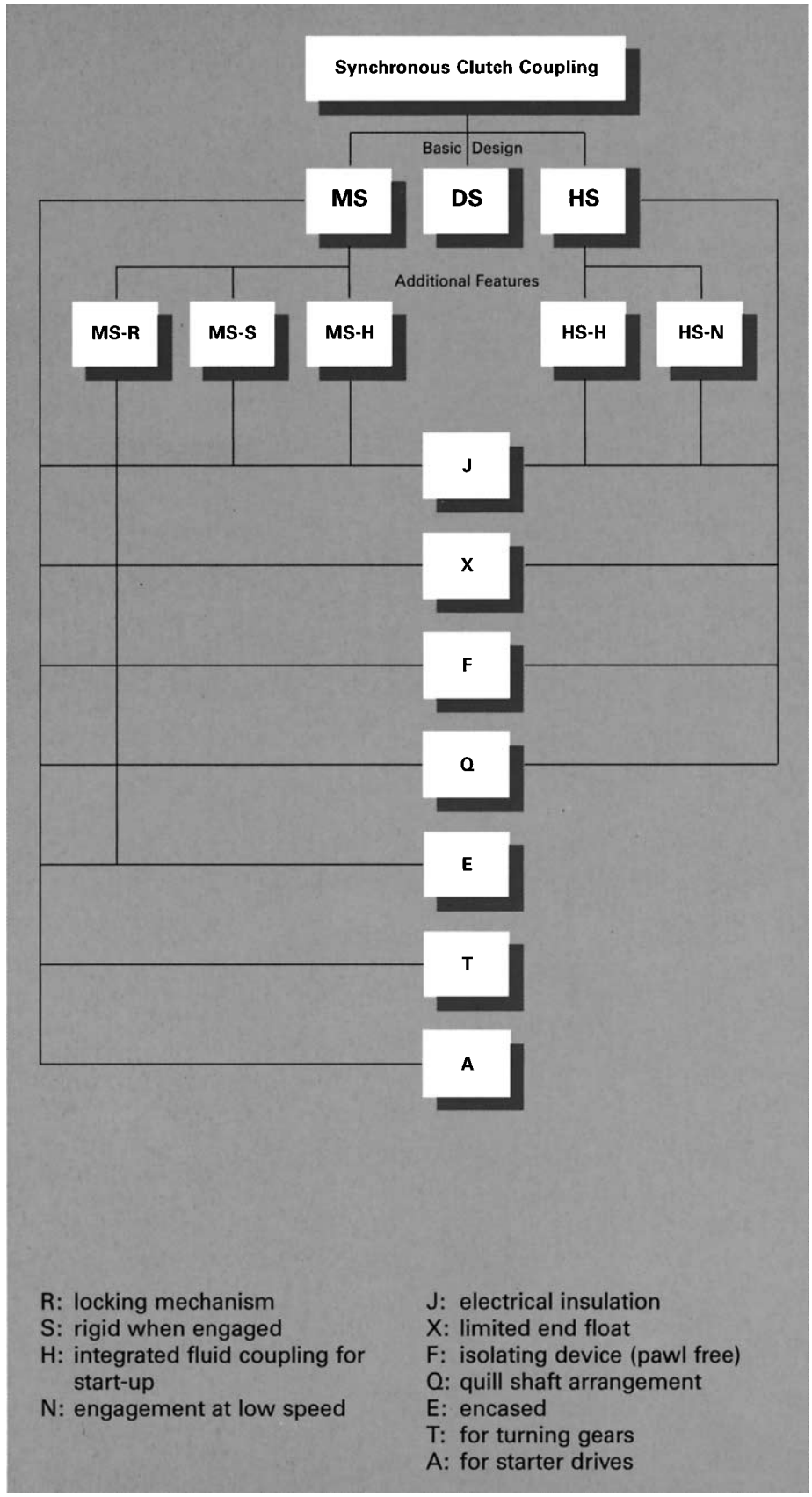


FIG. P-57 Model designation uttering for synchronous clutch coupling. (Source: MAAG Gear Company.)

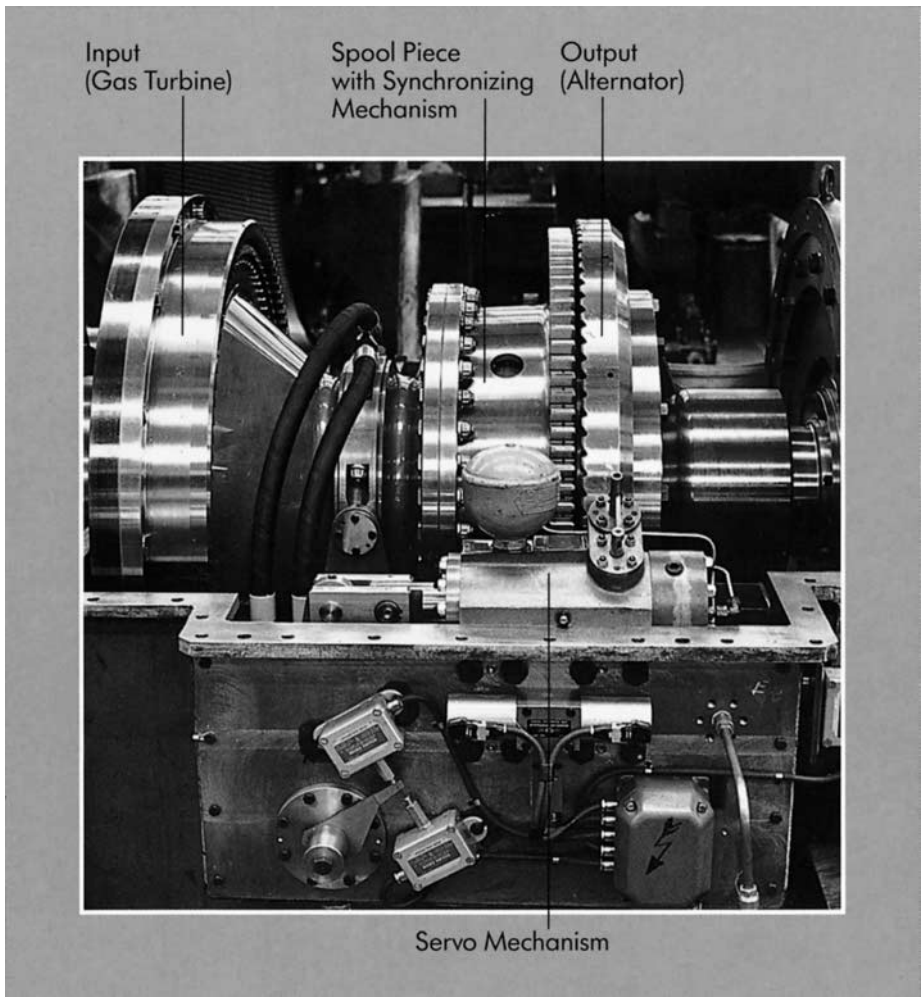


FIG. P-58 Synchronous clutch coupling HS-85 between a gas turbine and alternator. (Source: MAAG Gear Company.)

disengages immediately. The gas turbine is shut down and the alternator is rotating, operating as a synchronous condenser.

Synchronous clutch coupling of the type HS-85-H (see Figs. P-60 and P-61) can be installed in a power plant between the gas turbine and alternator.

This type HS-85-H (as the designation indicates) is combined with a fluid coupling. This fluid coupling will be used for starting the gas turbine when the alternator is working as a synchronous condenser. The alternator rotates at synchronous speed, i.e., 3000 min^{-1}; in the case of a 50-Hz grid, the clutch coupling is disengaged and the gas turbine is at a standstill.

If power is required from the gas turbine, the turbine is accelerated by filling the fluid coupling. From a certain speed on, the turbine accelerates under its own power and the fluid coupling is then emptied. The turbine accelerates further until it overruns the alternator. The clutch coupling engages automatically. Power can now be transmitted from the turbine to the alternator.

With this arrangement an expensive separate starting device for the gas turbine can be eliminated.

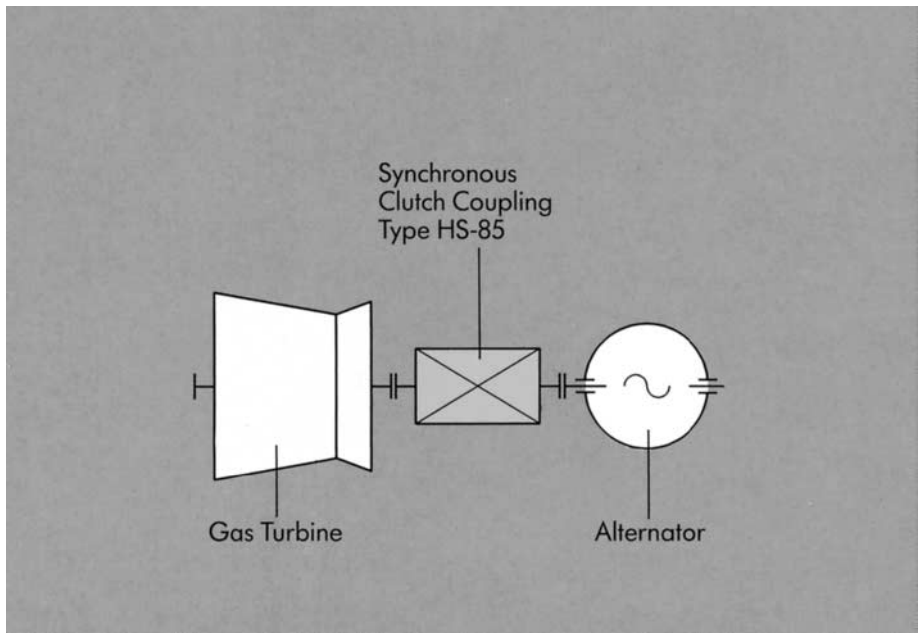


FIG. P-59 Schematic of Fig. P-58. (Source: MAAG Gear Company.)

Figures P-62 and P-63 are an example of synchronous clutch application in an air storage power plant.

The synchronous clutch coupling of the type MS-85-S is mounted between the gas turbine (power turbine only) and the alternator. In this case the MS-type coupling is provided with the S-feature, which means that the coupling will behave like a rigid coupling when engaged.

Other applications

Synchronous clutches for automatic turning gears type MS-. . .-T (this information source's model designations). Synchronous clutches for turning gears (see Fig. P-64) are of simple construction generally with outstanding reliability, high torque transmitting capacity, and suitable for turbomachinery installations.

The clutch automatically engages at the instant the input speed tends to overtake that of the output shaft. Conversely, the clutch will disengage also fully automatically when the output shaft speed exceeds the speed of the input shaft. A full range of turning gear clutches is available.

Free-standing synchronous clutch couplings type MS-. . .-E (this information source's model designations). Encased synchronous clutch couplings (see Fig. P-65) have been developed for installations where complete isolation of the driving machine is required for on-site maintenance while the driven machine continues to rotate. Typical applications for the encased clutches are fan drives with two driving machines.

The input and output shaft are supported each by two amply dimensioned bearings on each shaft. The type MS-clutch is mounted inside the casing between the two shafts. The casing is of fabricated steel plates.

Standard flexible couplings, e.g., diaphragm couplings, can be used to connect the clutch unit to the driving and driven machines.

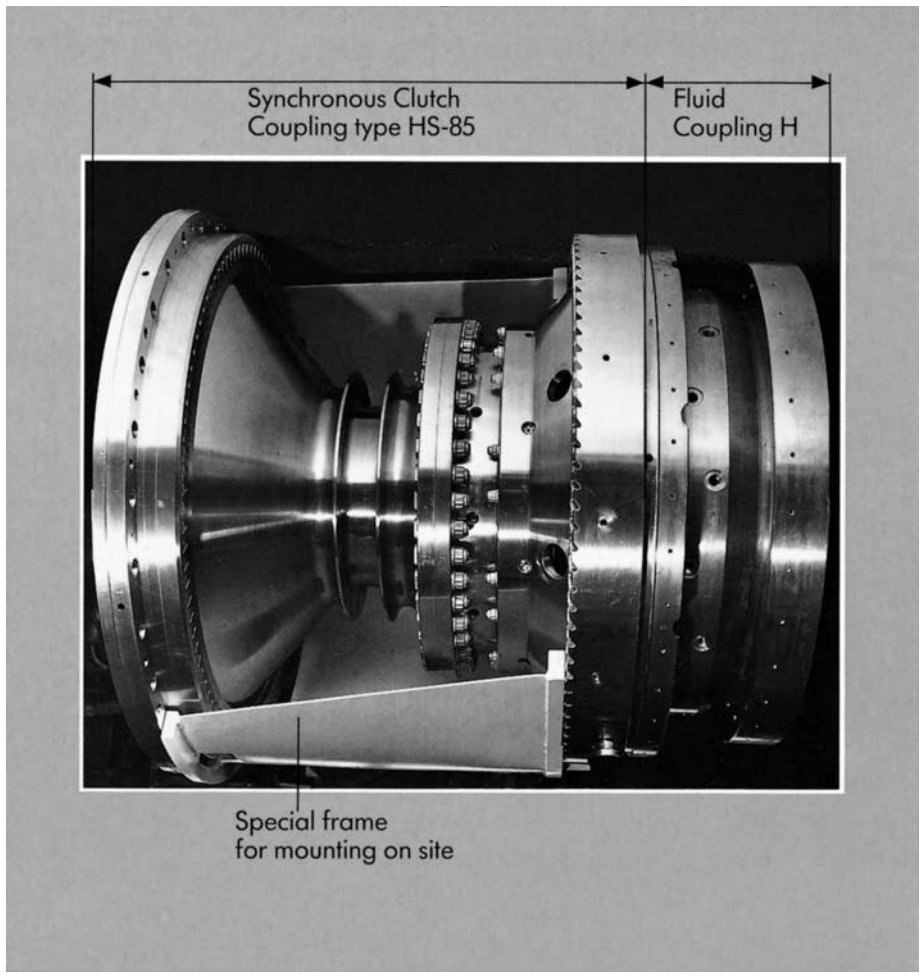


FIG. P-60 Synchronous clutch coupling HS-85-H between a gas turbine and alternator. (Source: MAAG Gear Company.)

Figures P-66 and P-67 show a typical example of synchronous clutch coupling application in the field of combined cycle technology in a chemical plant. Steam is generated by a conventional boiler which can be used for the process technology or power generation. If steam is available the steam turbine will be started and automatically coupled to the alternator.

In the original arrangement cooling steam was required for the windmilling of the steam turbine (no power generation by the steam turbine). By installing the clutch coupling the steam turbine can be shut down, hence no cooling steam is required and the overall efficiency is considerably increased. In this installation example the MS-36-J is provided with an electrical insulation and installed between the steam turbine and alternator. In this particular case the synchronous clutch coupling was installed as a replacement of a gear coupling.

Energy recovery, combined cycle technology, cogeneration

Figures P-68 and P-69 show a typical energy recovery application of the type MS-14 synchronous clutch coupling in a compressor installation where excessive process gases are available. These gases, instead of being released by a valve, are now used

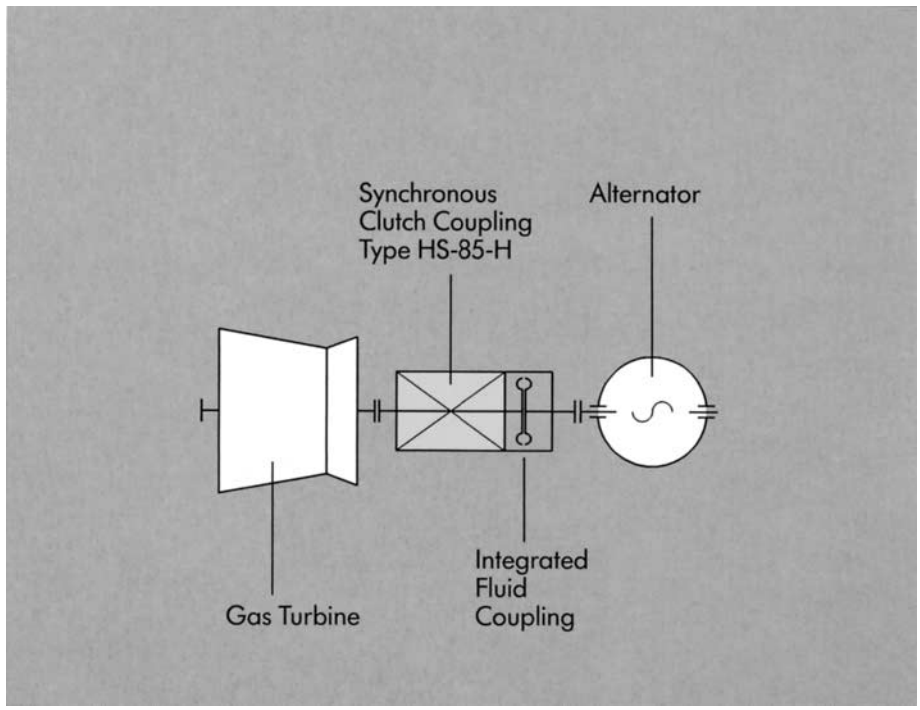


FIG. P-61 Schematic of Fig. P-60. (Source: MAAG Gear Company.)

to drive a gas expander that is coupled via the type MS-14 clutch with a centrifugal compressor. In the case when no excessive process gases are available, the gas expander will be disconnected automatically, thus avoiding windage losses in the gas expander.

Product Application Case 2: Gearboxes for Generator Drives*

These parallel shaft speed reduction gears with vertically offset shafts are

- Suitable for power stations with gas turbines of various makes and types, e.g., for power packages with Frame 6, 5, and 3 gas turbines
- Available for powers ranging from 10 to 75 MW
- Available for generator speeds of 3600, 3000, 1800, and 1500 rpm
- Designed for frequent starting and immediate transmission of full load under extreme ambient conditions
- Flexibility for system design as shown in Figs. P-70 and P-71
- Optional synchronous clutch couplings to connect or disconnect gas turbine and generator fully automatically when the generator is running as a synchronous condensor.

The gear and pinion are vertically offset. They are contained in a fabricated steel casing of rigid design comprised of three sections.

* Source: MAAG Gear Company, Switzerland.

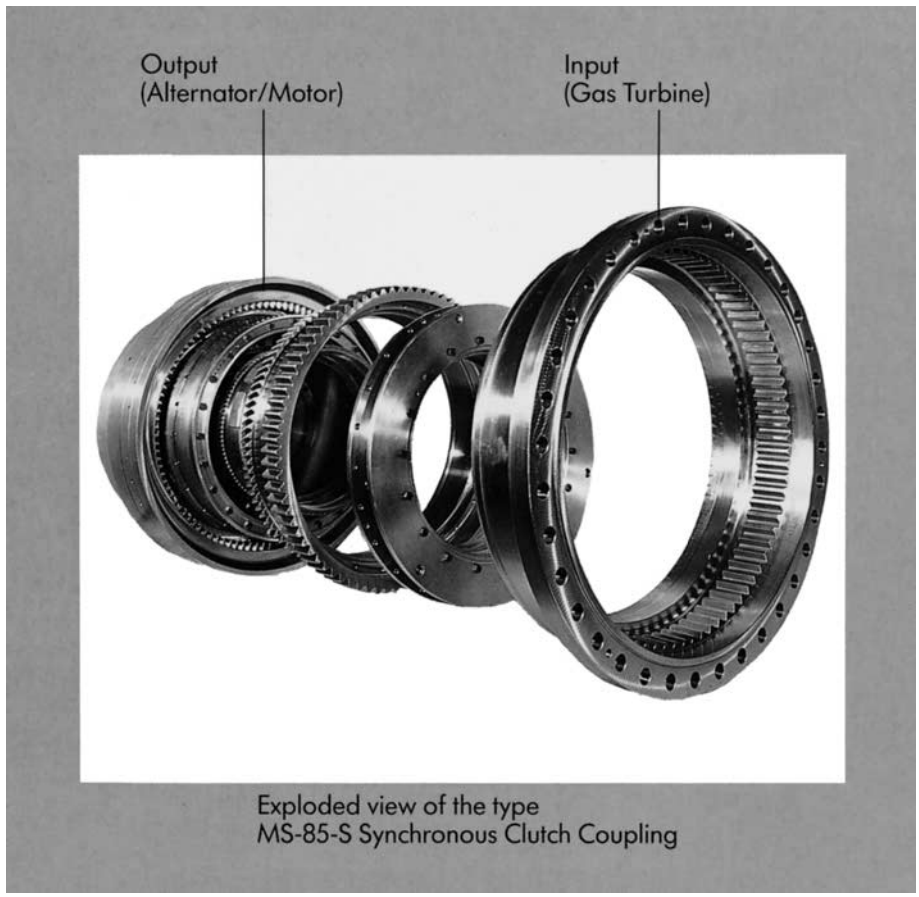


FIG. P-62 Components of the MS-85-S clutch coupling. (Source: MAAG Gear Company.)

The gears are carburized, hardened, and ground. Normally single helical gears are used. They are calculated in accordance with MAAG design criteria, or to AGMA, ISO, or API standards, with a service factor of not less than 1.1.

The shafts are supported in babbitted lined bearings. Each shaft may be provided with an integral thrust bearing. The gears can be equipped with thrust cones to compensate the gear thrust and to transfer thrust loads from one shaft to the other.

The basic gear design options are shown in Fig. P-70.

Design details

Tooth modifications. Gears and pinions under load suffer elastic deflections and their temperatures are raised unevenly. Deformations and thermal expansion have detrimental effects on the tooth engagement. The tooth flanks are therefore modified during grinding to achieve an ideal load distribution at the rated load and speed. Compensation for thermal effects is absolutely vital on high-speed gears.

Journal bearings. Pressure-lubricated three- or four-lobe bearings provide excellent load capacity and journal stability.

Gears that operate at extreme velocities are equipped with direct lubricated tilting pad radial bearings.

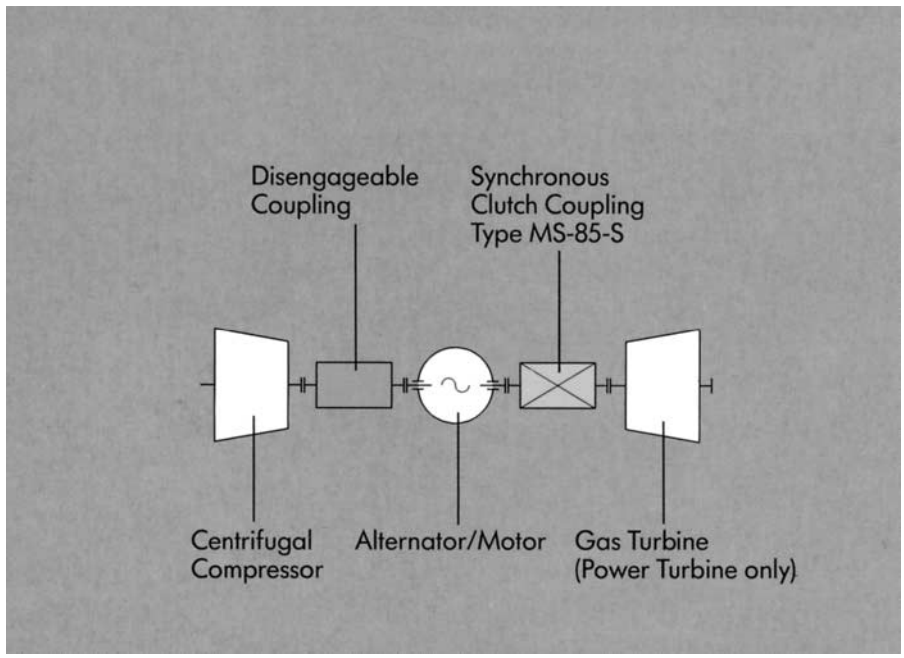


FIG. P-63 Schematic of the MS-85-S in a working assembly. (Source: MAAG Gear Company.)



FIG. P-64 Automatic turning gear clutch type MS-8-T installed in gearbox between turning gear and pinion shaft. (Source: MAAG Gear Company.)

Thrust bearings. Standard tilting pad thrust bearings with direct lubrication are provided if required. They are always located at the free shaft ends.

Thrust cones. The thrust faces are slightly cone-shaped and surface hardened and ground. They are lubricated by oil from the meshing teeth.

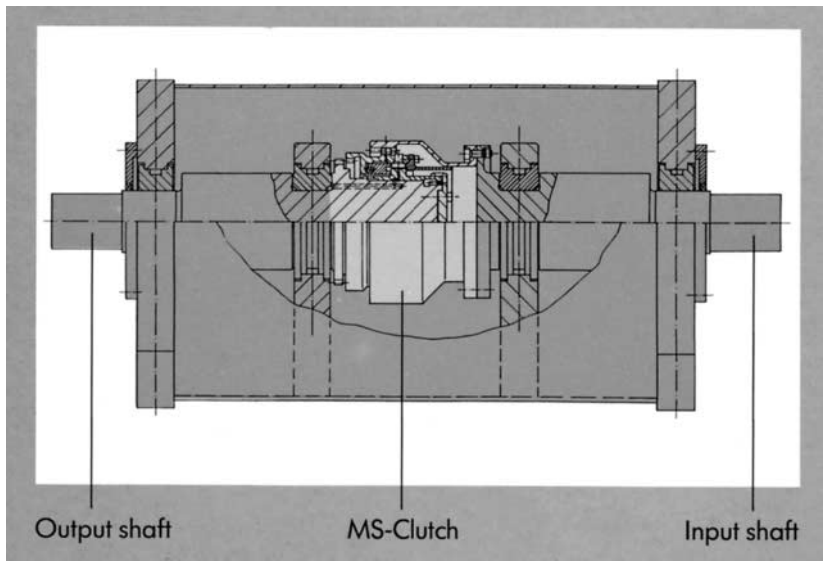


FIG. P-65 Design principle of MAAG freestanding synchronous clutch couplings. (Source: MAAG Gear Company.)

Geared systems

The choice of the basic gear design is governed by the disposition of the machinery installation and the type of couplings and clutches used. The careful choice of gear and couplings may reduce the number of thrust bearings and hence the overall losses (e.g., Fig. P-71*B* and *C*).

Flexible couplings. Gear couplings or diaphragm couplings are used to absorb shaft misalignments and axial heat expansions (Fig. P-71*A*).

Quill shafts. Flexible shafts are axially rigid and able to transmit thrust loads. They can compensate for small shaft misalignments. Where short lengths are important the quill shafts are placed in bores through the gear shafts (Fig. P-71*B* and *C*).

Clutches. Standard synchronous clutch couplings are used for automatic disengagement and reengagement. When engaged, these form-fitted geared clutches have identical characteristics to a gear-type coupling (Fig. P-71*D*).

These clutches can be quill shaft mounted to reduce length (Fig. P-71*E*).

Rigid flanges. Rigid flanges are only recommended where satisfactory shaft alignment can be maintained or with special layouts, e.g., where machinery rotors are supported at the input end by the gear bearings (Fig. P-71*F*).

Instrumentation

The standard instrumentation includes:

- One thermocouple or RTD on each radial bearing
- Two thermocouples or RTDs on each thrust bearing, loaded side
- Provisions for mounting two probes (90° apart) on each shaft (input/output side)
- Provisions on casing for mounting two accelerometers

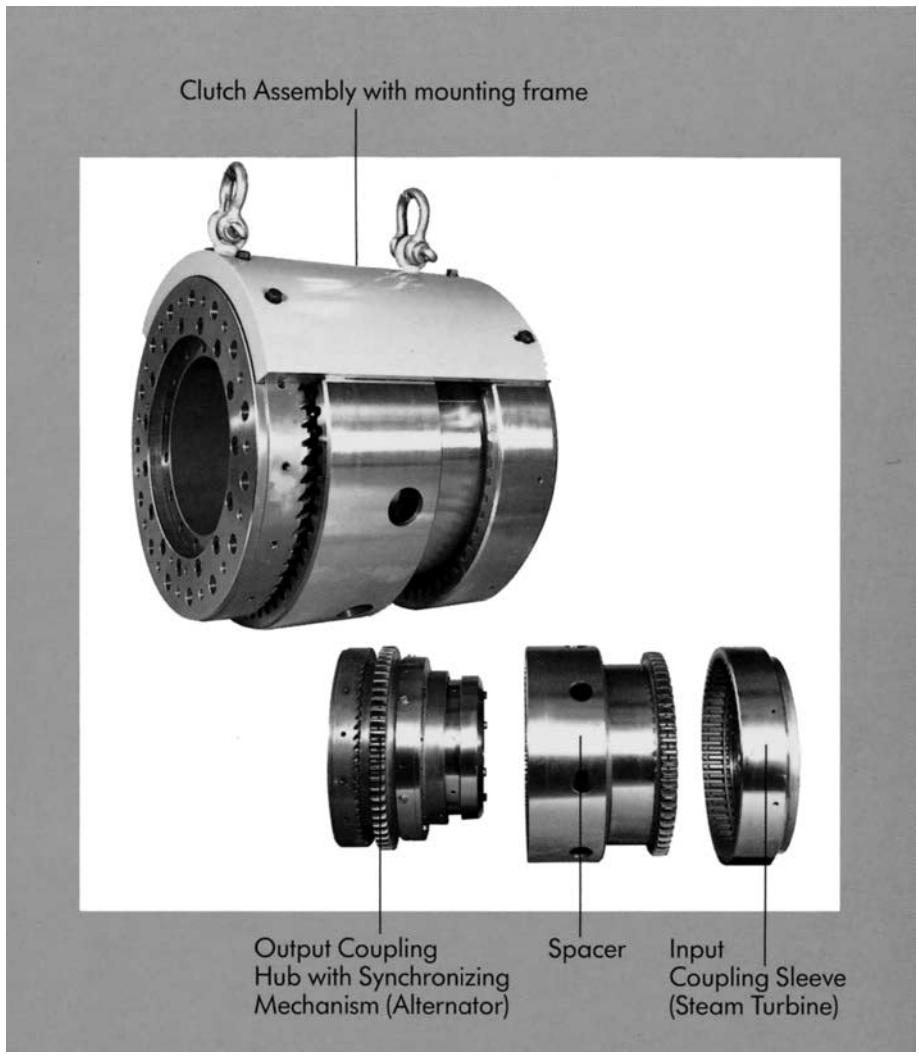


FIG. P-66 Clutch assembly and components. (Source: MAAG Gear Company.)

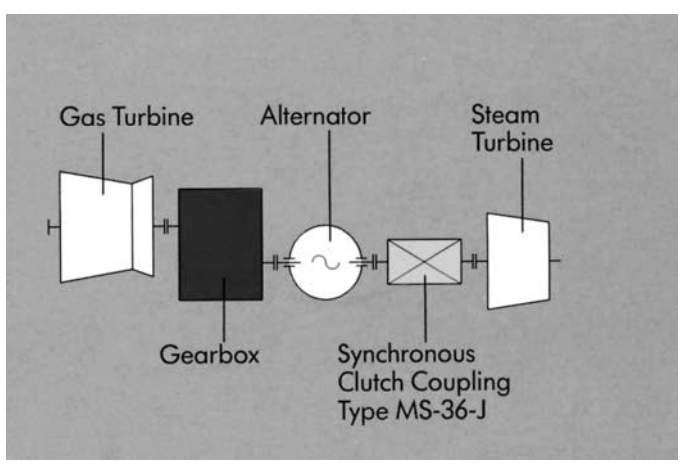


FIG. P-67 Clutch coupling MS-36-J in a working assembly. (Source: MAAG Gear Company.)

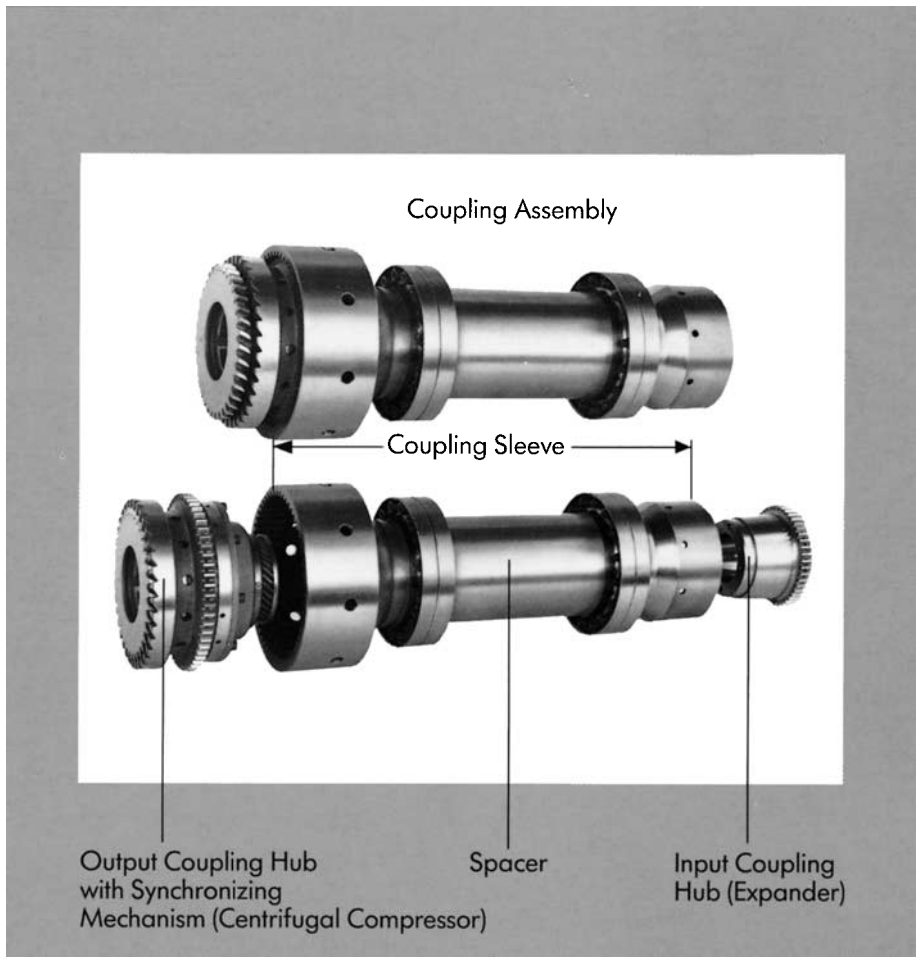


FIG. P-68 MS-14 clutch coupling assembly. (Source: MAAG Gear Company.)

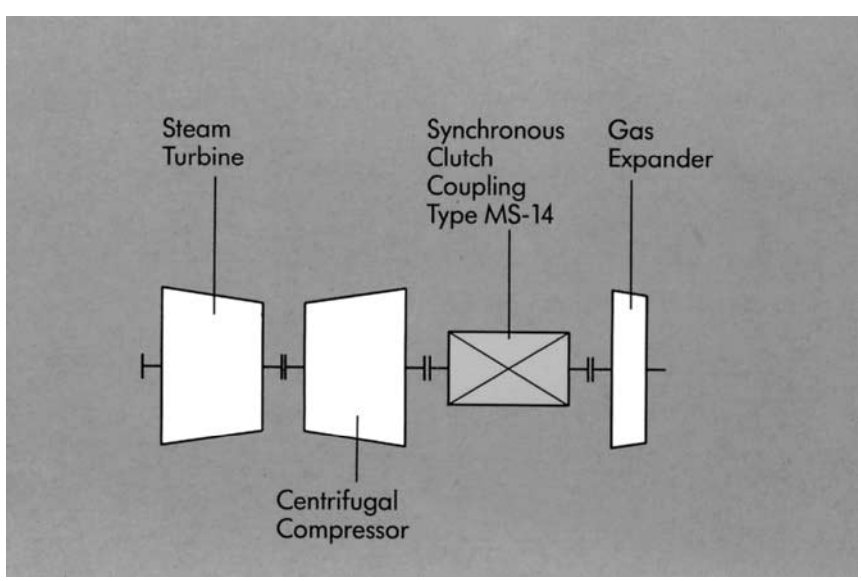


FIG. P-69 Schematic of MS-14 clutch coupling in a working assembly. (Source: MAAG Gear Company.)

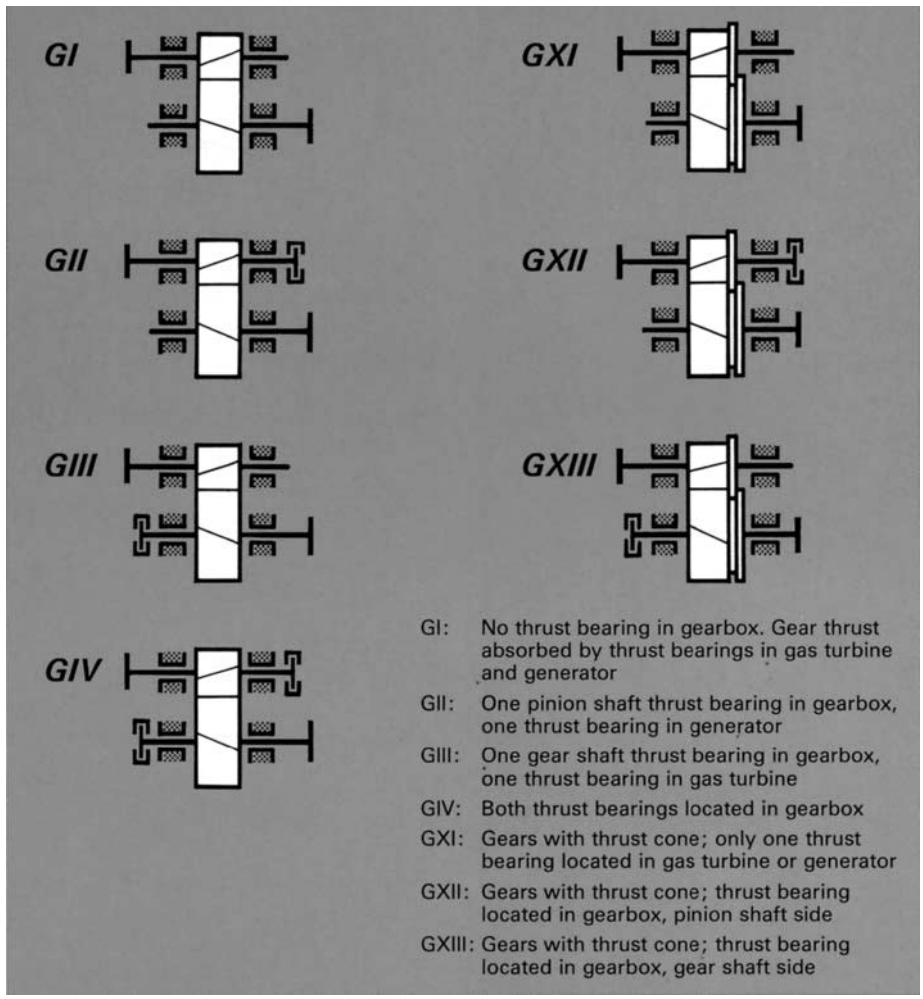


FIG. P-70 Basic gear designs. (Source: MAAG Gear Company.)

Hydrodynamic Power Transmission*

Types of power transmission

1. Mechanical transmission (power-grip toothed-belt drive) (see Fig. P-72)
2. Hydrostatic power transmission (displacement-type transmission) (see Fig. P-73)
3. Hydrodynamic power transmission (converter) (see Fig. P-74)

The circular/elliptical shapes in Figs. P-73 and P-74 symbolize some fluid particles. Their shape is meant to illustrate:

- Utilization of the pressure in hydrostatic power transmissions.
- Utilization of the mass forces in hydrodynamic power transmissions.

* Source: J.M. Voith GmbH, Germany.

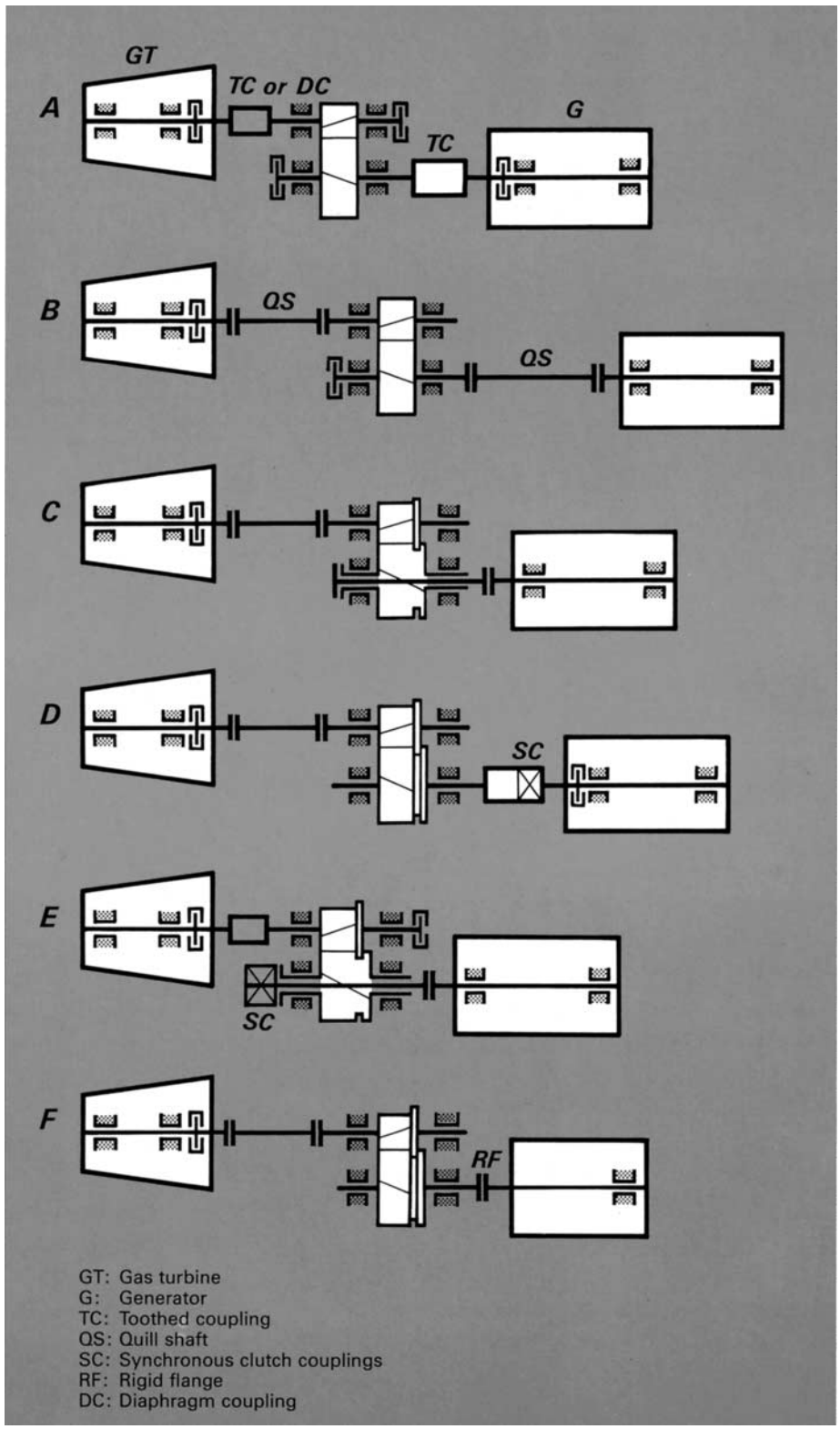


FIG. P-71 Examples of geared systems. (Source: MAAG Gear Company.)

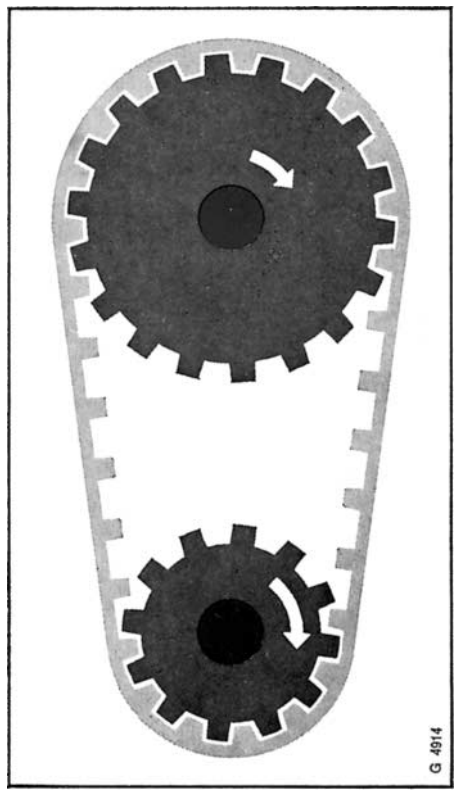


FIG. P-72 Mechanical transmission (power-grip toothed-belt drive). (Source: J. M. Voith GmbH.)

Hydrodynamic power transmissions—also called turbotransmissions or hydrokinetic drives—are hydraulic converters. These converters change the speed and torque between input and output shafts steplessly and automatically. The energy is transmitted by a fluid as medium power transmissions fundamentally differ from all other power transmissions. This applies in particular to all mechanical power transmissions.

Fluids readily fill any available space, move easily, and can transmit pressure in all directions. These peculiarities have, for a long time already, made fluids the most valuable agents to transmit and transform energy for technical applications. While it is typical of hydrostatic power transmissions to transmit pressure (displacement-type transmission), it is a main characteristic of the hydrodynamic power transmissions that they utilize the mass forces of circulating operating fluids.

Figures P-75 and P-76 are a simplified schematic arrangement of the blading of a hydrodynamic power transmission (torque converter).

Fig. P-75: pump impeller (inner varied annulus) and turbine wheel (outer bladed annulus)

Fig. P-76: guide blades (reaction member) (aerofoil shapes illustrated)

The guide blades of this converter are rigidly connected to the converter shell (casing). The casing is filled with the operating fluid. Pump impeller and turbine wheel are rigidly attached on the shafts.

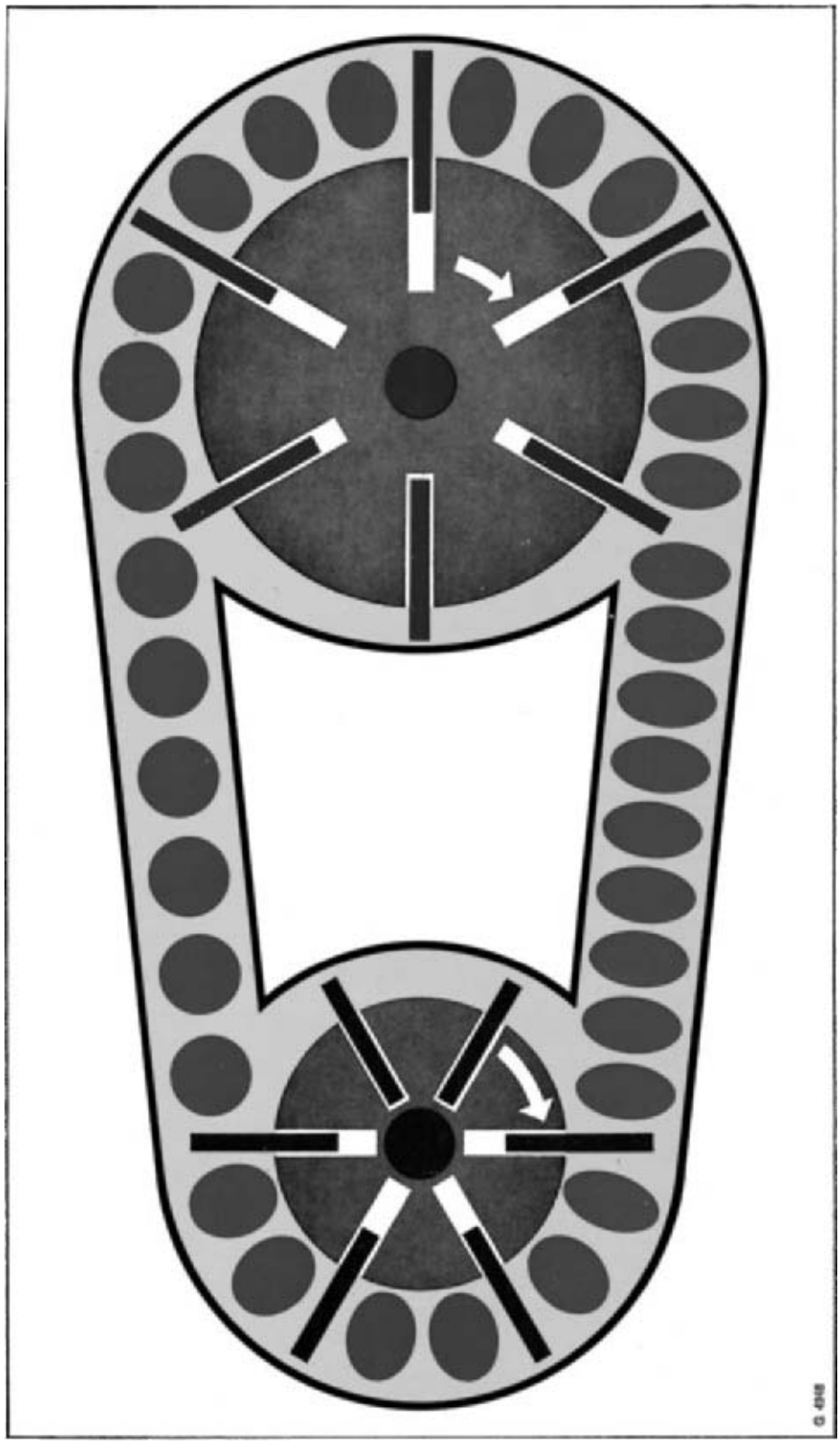


FIG. P-73 Hydrostatic power transmission (displacement-type transmission). (Source: J. M. Voith GmbH.)



FIG. P-74 Hydrodynamic power transmission (converter). (Source: J. M. Voith GmbH.)

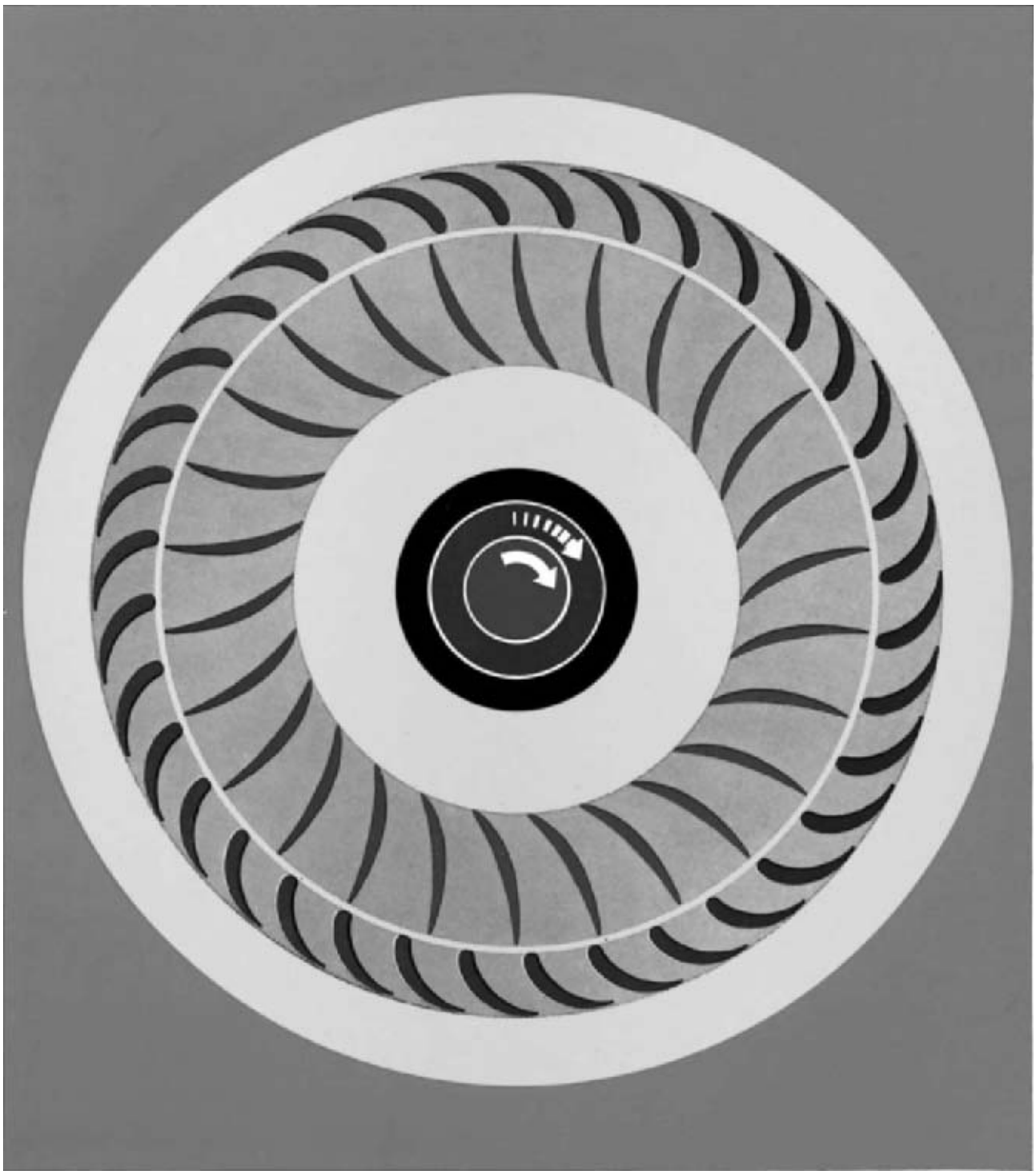


FIG. P-75 Simplified schematic of the blading of a torque converter: pump impeller, inner circle; turbine wheel, outer circle. (Source: J. M. Voith GmbH.)

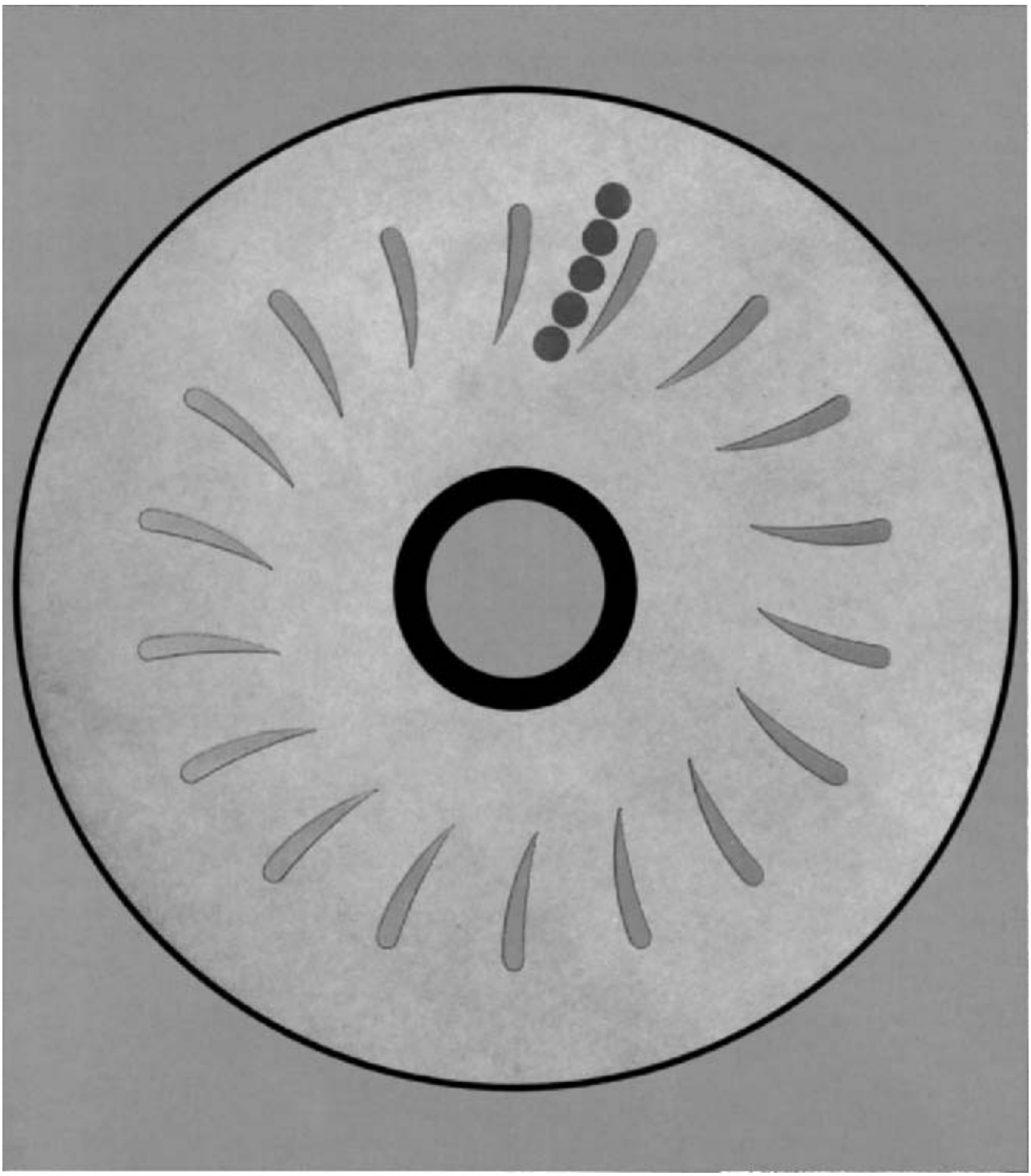


FIG. P-76 Simplified schematic of the blading of a torque converter: guide blades (reaction member). (Source: J. M. Voith GmbH.)

General arrangement of hydrodynamic power transmissions and their principle of operation (Fig. P-77)

The heart of a Föttinger™ converter is the hydraulic circuit that contains pump, turbine, and reaction member, all consolidated in a single casing and forming a closed fluid circuit.

The pump is connected to the input shaft, and the turbine to the output shaft. The fluid flow initiated by the pump drives the turbine. Power is transmitted by the circulation of the fluid between these two members of the converter, utilizing the mass forces of the circulating fluid.

Also with hydrodynamic power transmissions the sum of all torques must be zero. The reaction member absorbs the differential torque between input and output torques. Depending upon the torque acting on the guide blades, the turbine torque (output torque) may be larger or smaller than the pump torque or may be of the same magnitude (input torque). Under different operating conditions, the turbine speed may widely differ from the pump speed.

There is no mechanical connection between input and output ends. (See Fig. P-78.)

In gear units, the gears are correctly meshed and establish a force-locked connection between input and output ends (see Fig. P-79).

In hydrodynamic power transmissions, the circulating fluid connects input and output ends. No form-fit design, but a force-locked connection is used (see Fig. P-80).

Special features of hydrodynamic power transmissions

- Stepless transmission ratio (not constant)
- Flexible connection (no form-fit design)
- Load-controlled operation (the output speed matches the load on the output shaft)
- Transmission is free from wear and tear (no abrasion)
- Vibrational isolation (no mechanical connection between input and output ends)
- No reaction of output load on input end [by using suitable converter blading, free choice of driving motor (engine) with the required overload capacity; no stalling of engine or motor]

Figure P-81 shows a section of the tractive effort curve of a converter. The output speed is always adapted automatically to the prevailing load conditions. Figure P-82 shows the converter running driven equipment steplessly up to speed. Figure P-83 is meant to demonstrate that (since oil has no teeth) converters provide vibrational isolation.

Hydrodynamic power transmission operation

The ratio of input to output speed is not constant (as in the case of gear units) but adapts itself to the output load automatically and steplessly.

The absorbed power is determined by the characteristics of the torque converter.

The torques are not inversely proportional to the speeds as they are with mechanical transmissions.

Reversing the direction of rotation of pump and power flow provides a different behavior of the power transmission.

The converter types differ by the shape of their power absorption curves (absorbed power as a function of the ratio output speed/input speed).

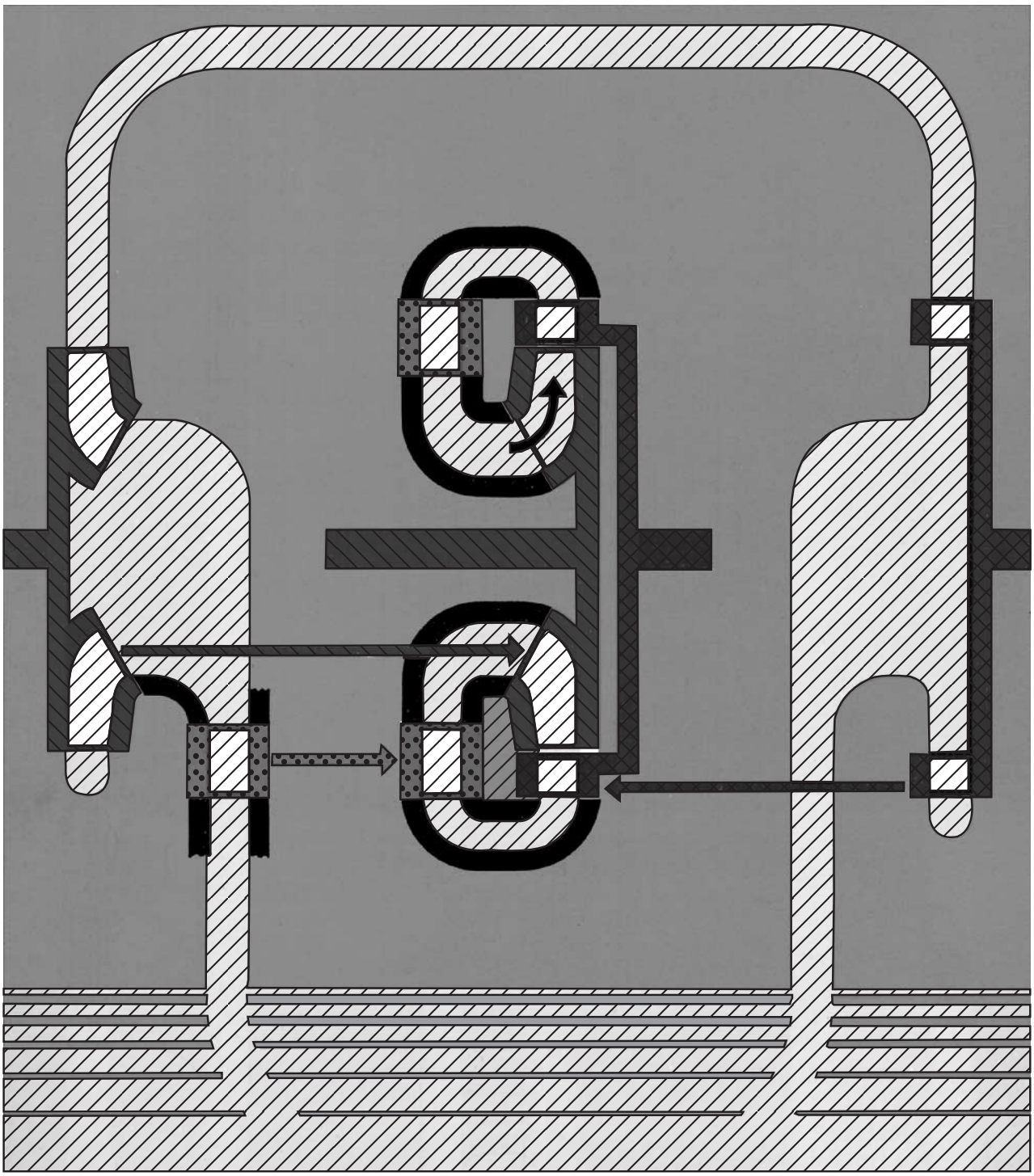


FIG. P-77 Hydrodynamic power transmission: operating principle schematic. (Source: J. M. Voith GmbH.)

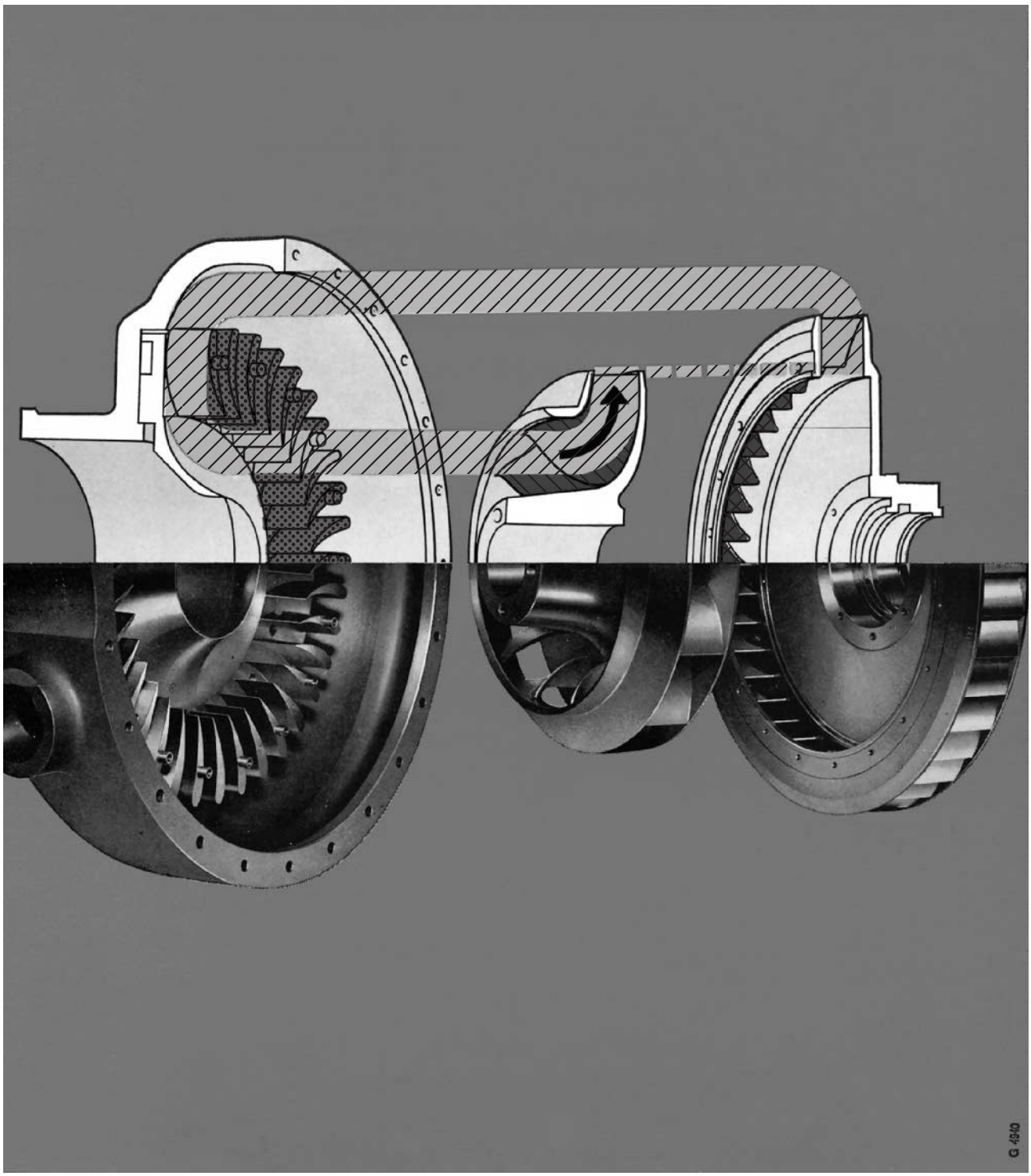


FIG. P-78 Hydrodynamic power transmission: operating principle cutaway. (Source: J. M. Voith GmbH.)

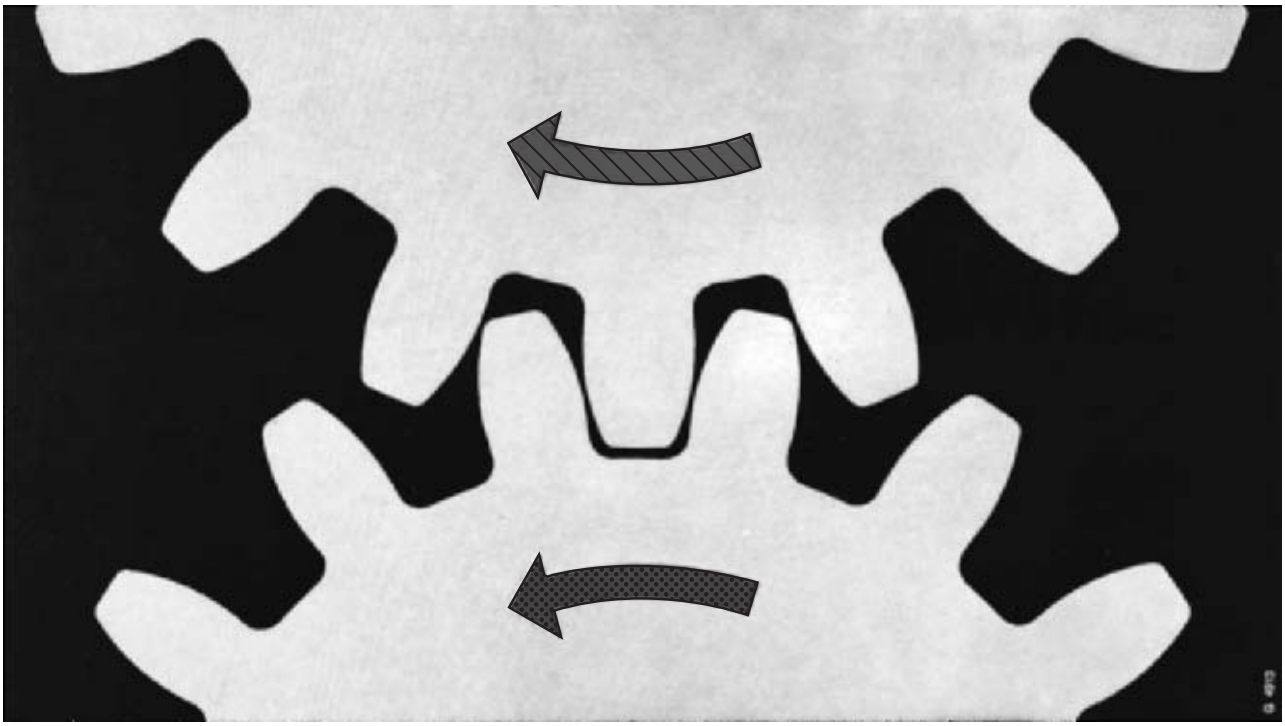


FIG. P-79 Gears: force-locked connection between gears. (Source: J. M. Voith GmbH.)

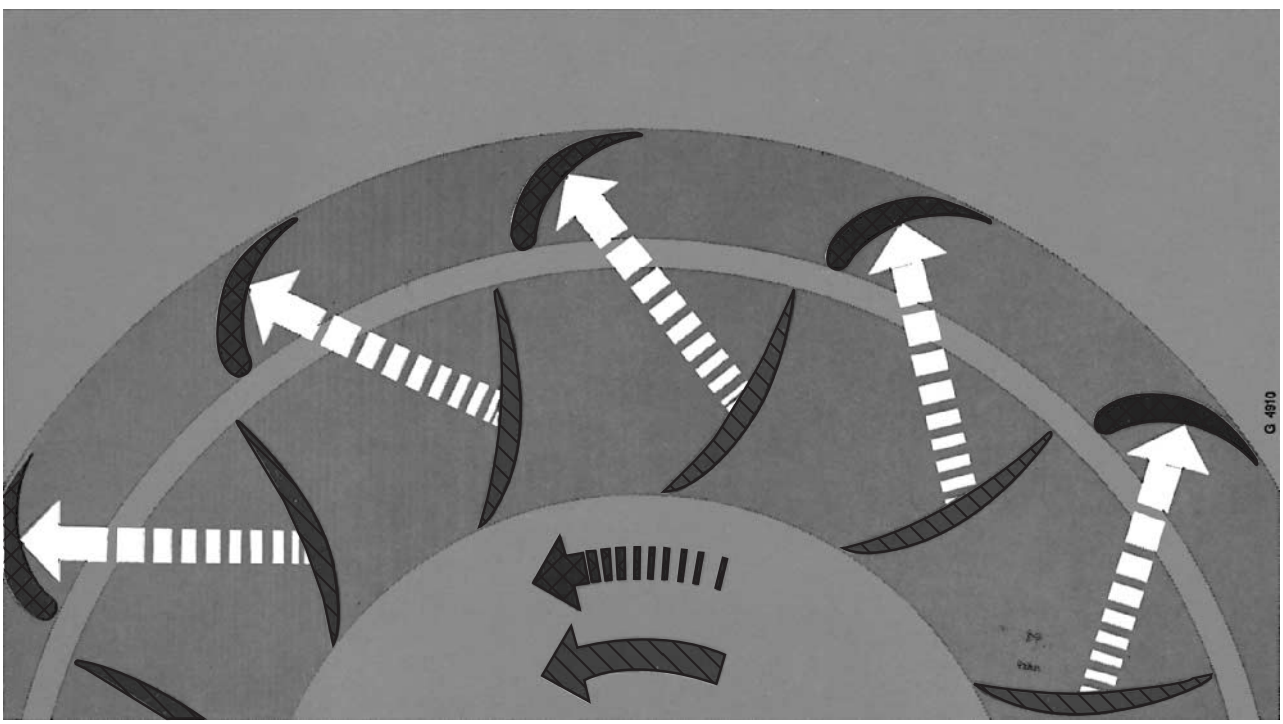
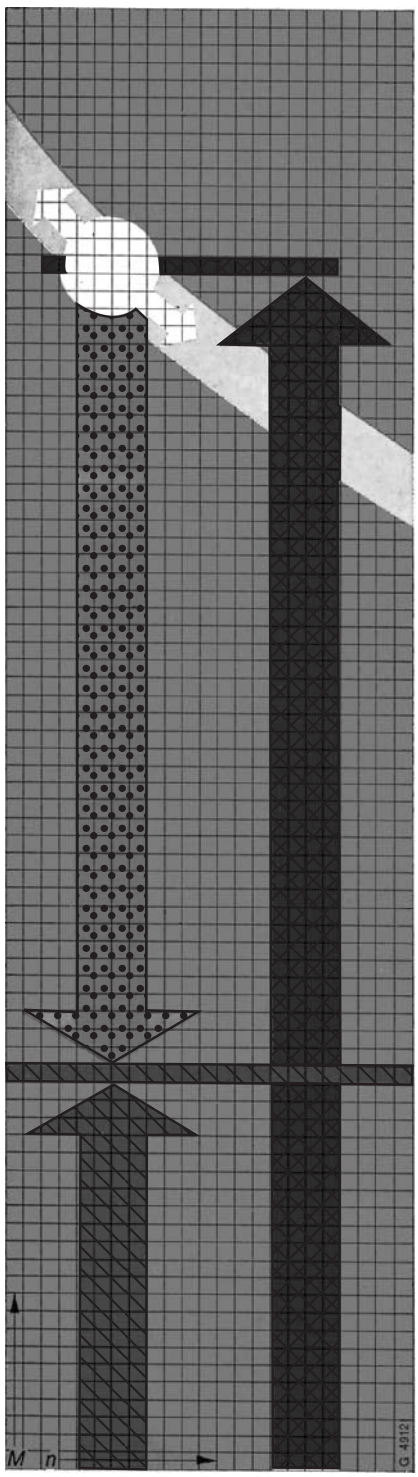
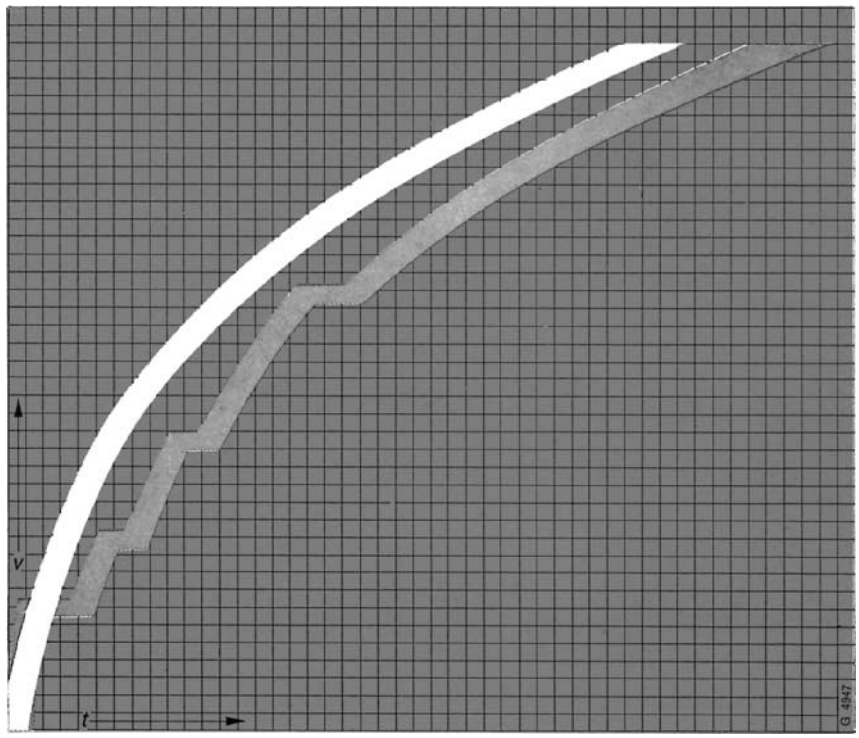


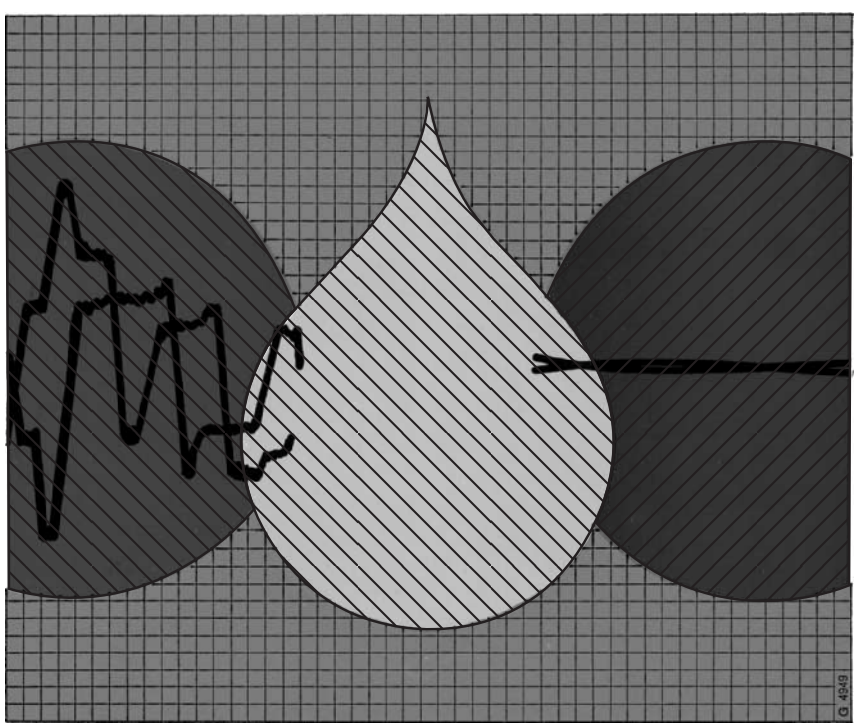
FIG. P-80 Hydrodynamic power transmission: circulating fluid provides connection between input and output ends. (Source: J. M. Voith GmbH.)



81



82

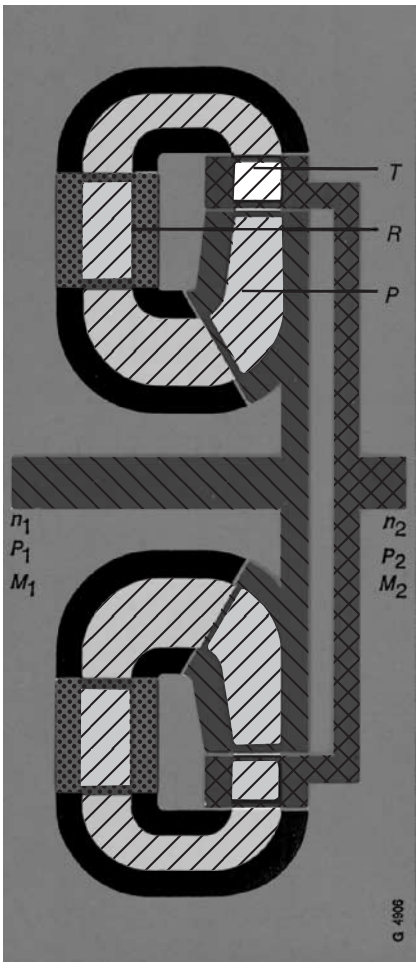


83

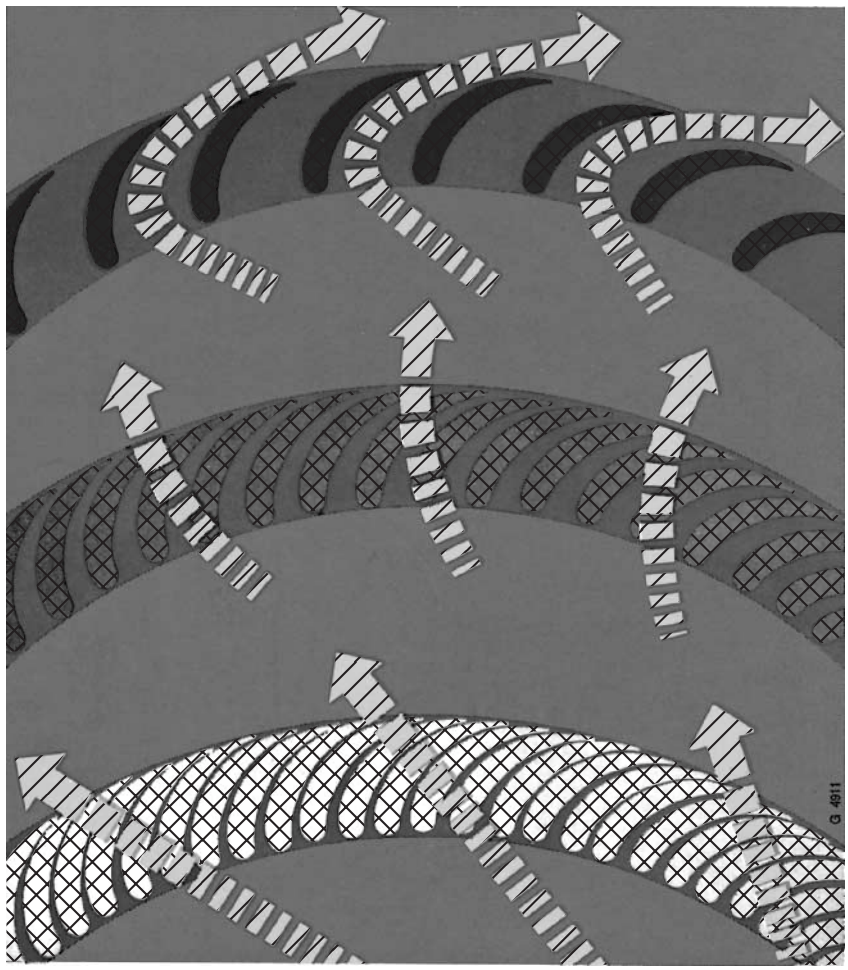
FIG. P-81 Part of the tractive effort curve of a converter. (Source: J. M. Voith GmbH.)

FIG. P-82 Hydrodynamic power can run driven equipment steplessly up to speed. (Source: J. M. Voith GmbH.)

FIG. P-83 As oil has no teeth, hydrodynamic converters provide vibrational isolation. (Source: J. M. Voith GmbH.)



84



85

FIG. P-84 Schematic component of torque converter. (Source: J. M. Voith GmbH.)

FIG. P-85 Flow of operating fluid through turbine wheel under operating conditions. (Source: J. M. Voith GmbH.)

By acceleration (see Figs. P-84 and P-85) of a fluid mass inside the pump, a torque M_1 is created at the input shaft of the torque converter. The fluid mass is decelerated again in the turbine, thus developing a torque M_2 that is transmitted to the output shaft. Figure P-84 shows the schematic arrangement of the torque converter. Figure P-85 shows the flow of operating fluid through the turbine wheel under various operating conditions. Figure P-86 shows the converter's torque and efficiency curves (characteristics).

The stationary reaction member (guide blades) takes up the difference between input and output torque, thus providing torque multiplication. With the torque converter shown in the illustration, the absorbed torque M_1 is roughly constant with constant input speed n_1, even if the output speed n_2 fluctuates heavily. With increasing output speed, the torque M_2 at the output shaft steadily drops automatically and steplessly from a high startup torque. Any change in the

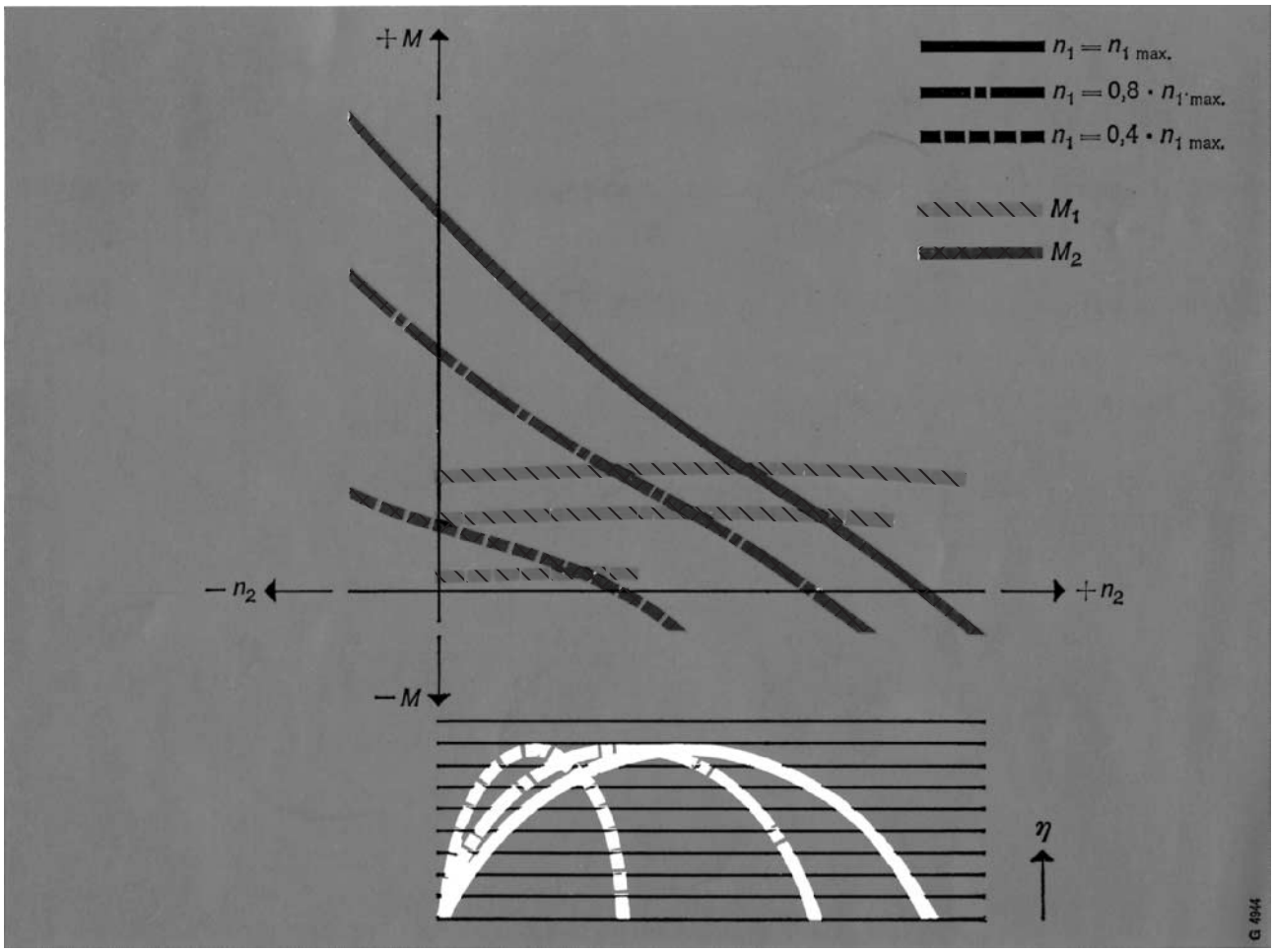


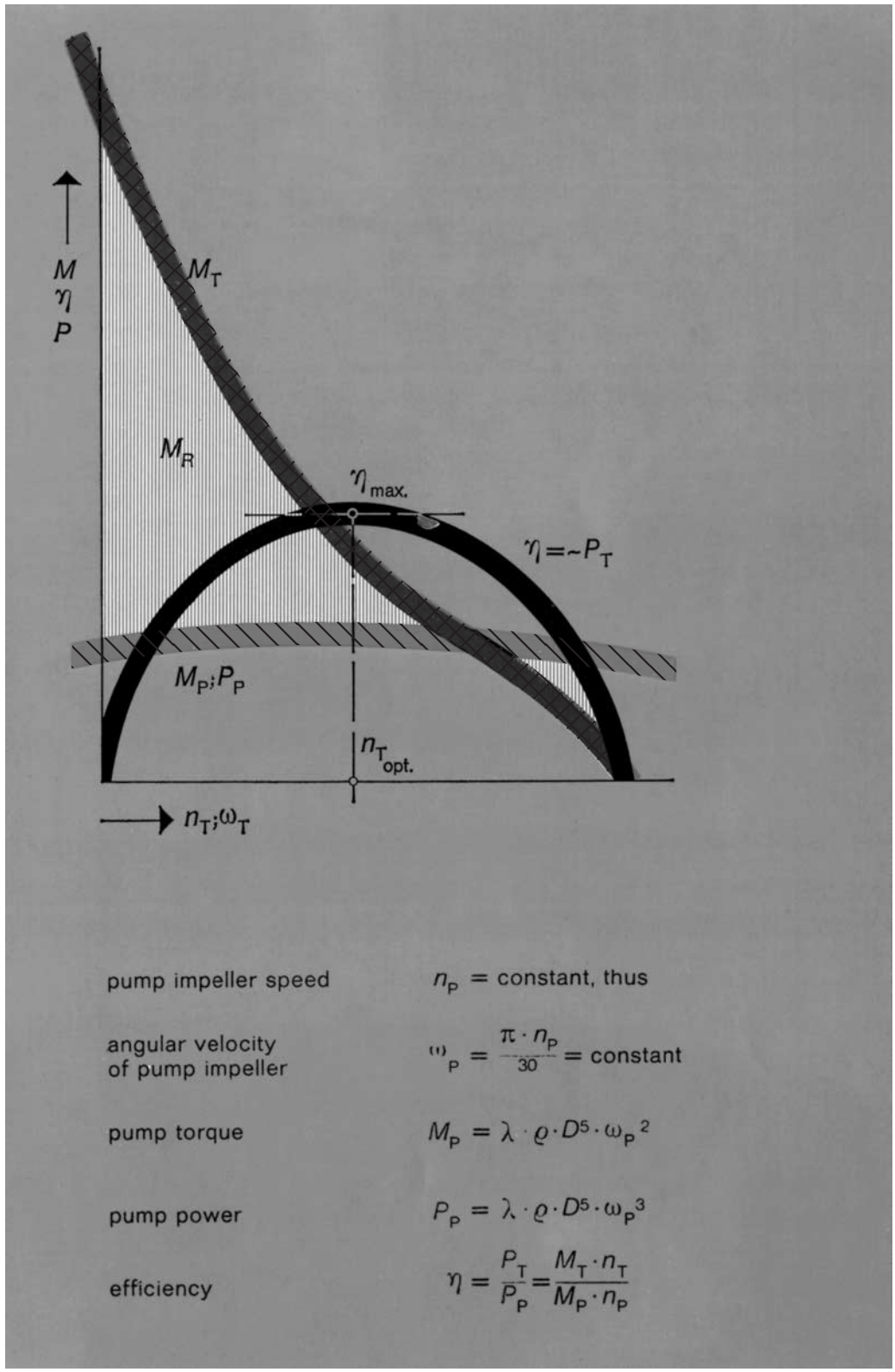
FIG. P-86 Torque and efficiency curve characteristics. (Source: J. M. Voith GmbH.)

deceleration of the fluid mass—due to a different turbine speed—also causes the transmitted torque to change. The circulating fluid is redirected by the turbine wheel, which causes the fluid to decelerate, and is shown for different operating conditions, viz. startup ($n_2 = 0$), rated speed ($n_2 = n_{\text{optimum}}$), and runaway speed ($n_2 = n_{\text{maximum}}$).

A change in output torque and output speed does not affect the motor (engine), even if the output speed should rise to such an extent that the output torque becomes zero or even negative. When the output speed is above the runaway speed, the torque converter produces a braking effect with no reaction on the motor (engine).

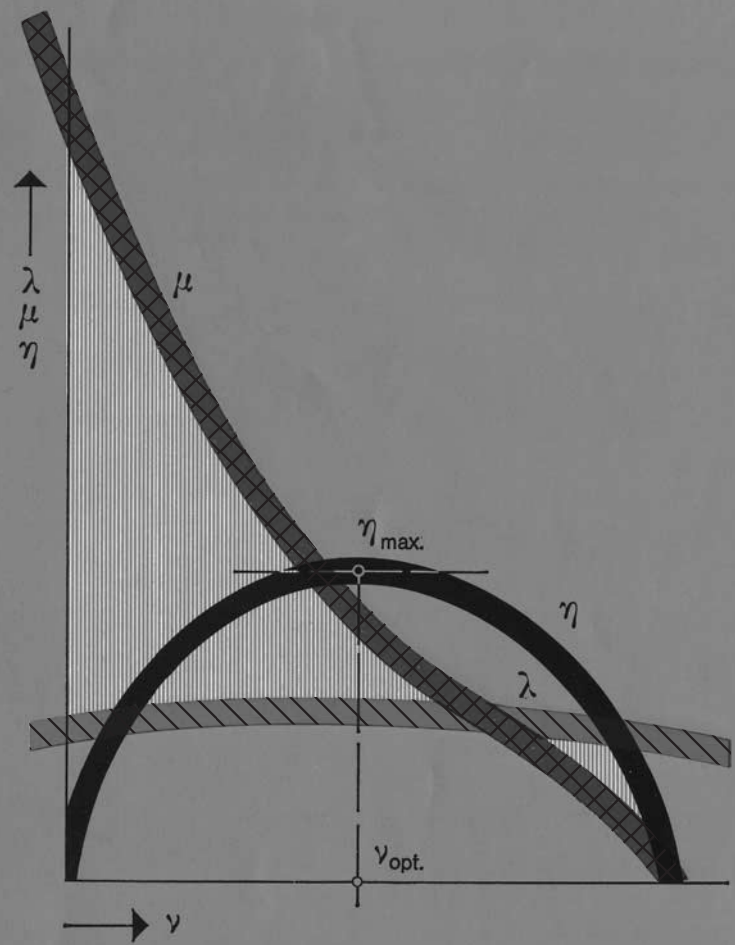
The characteristics of hydrodynamic power transmissions

See Figs. P-87 and P-88.



pump impeller speed	n_P = constant, thus
angular velocity of pump impeller	$\omega_P = \frac{\pi \cdot n_P}{30}$ = constant
pump torque	$M_P = \lambda \cdot \varrho \cdot D^5 \cdot \omega_P{}^2$
pump power	$P_P = \lambda \cdot \varrho \cdot D^5 \cdot \omega_P{}^3$
efficiency	$\eta = \frac{P_T}{P_P} = \frac{M_T \cdot n_T}{M_P \cdot n_P}$

FIG. P-87 Characteristics of a torque converter. (Source: J. M. Voith GmbH.)



speed ratio $\nu = \frac{n_T}{n_P} = \frac{\omega_T}{\omega_P}$

torque conversion $\mu = \frac{M_T}{M_P}$

efficiency $\eta = \frac{P_T}{P_P} = \frac{M_T}{M_P} \cdot \frac{\omega_T}{\omega_P} = \mu \cdot \nu$

power coefficient $\lambda = \frac{M_P}{\varrho \cdot D^5 \cdot \omega_P{}^2}$

specific gravity of operating fluid $\varrho = \frac{\gamma}{g}$

FIG. P-88 Dimensionless characteristics of a torque converter. (Source: J. M. Voith GmbH.)

Basic blading arrangements and associated converter characteristics

Power absorbed by converters is virtually constant. High torque multiplication possible. Suitable for motors (engines) that are sensitive to lugging down of their speed. See Figs. P-89, P-93, and P-97.

Main fields of application. Diesel locomotives and diesel railcars. Stationary drives with electric motors. Vehicles and construction machinery.

Power absorbed by converters drops at certain speeds. Clear limitation of maximum output speed. No overload protection required. See Figs. P-90, P-94, and P-98.

Main field of application. Road vehicles.

Power absorbed by converters drops. With increasing turbine speed, the power absorbed by the pump drops. The load on the driving motor (engine) increases with decreasing output speeds; the engine speed is lugged down. This results in fuel savings. See Figs. P-91, P-95, and P-99.

Main fields of application. Construction machinery. Shunting locomotives.

Power absorbed by converters increases. With increasing turbine speed, the absorbed power increases. Such characteristics are favorable for differential converters. See Figs. P-92, P-96, and P-100.

Main fields of application. Vehicles, in particular floor-level conveying equipment such as fork lift trucks, etc.

Operating costs comparison

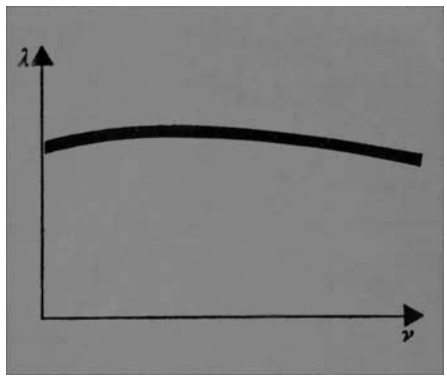
Geared variable-speed turbocouplings reduce costs in conversions and new installations.

The generic advantages of a geared variable-speed turbocoupling are:

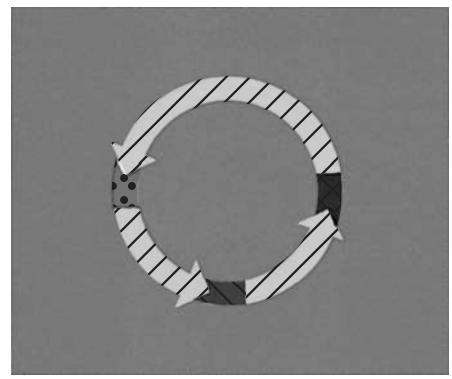
- A compact unit with integrated gear stage, designed and built to API613, SF1.4
- Motor starting under no load—stopping of the turbocompressor while the motor continues to run (rapid emptying)
- Controlled starting and run-up through critical speeds and process fields up to maximum compressor speed
- A wide infinitely variable-speed control range. Constant compressor output pressure in spite of varying molecular weight of the gas to be pumped
- Separate control of the starting and operating fields, each with control signals of 4–20 mA or 0.2–1 bar
- Energy saving compared with throttling on the suction side
- Damping of shock loads through hydrodynamic power transmission
- A simple unit requiring a minimum of maintenance and providing almost 100 percent availability
- Explosion-proof regulations can be inexpensively complied with
- The possibility of using standard squirrel cage motors

If optimal use is to be made of the advantages of hydrodynamic variable-speed couplings within an overall plant, then close cooperation is necessary between

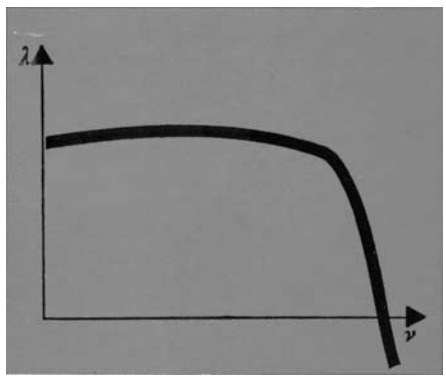
89



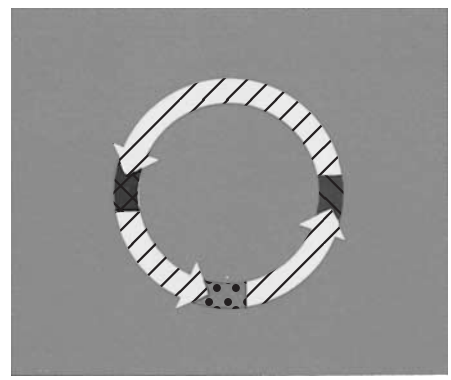
93



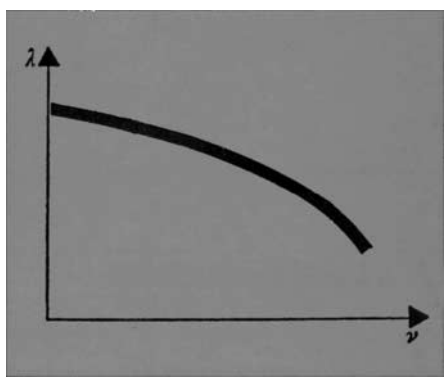
90



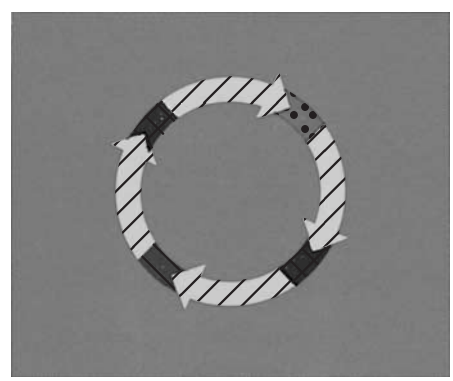
94



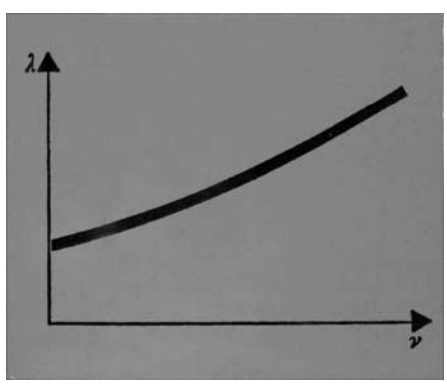
91



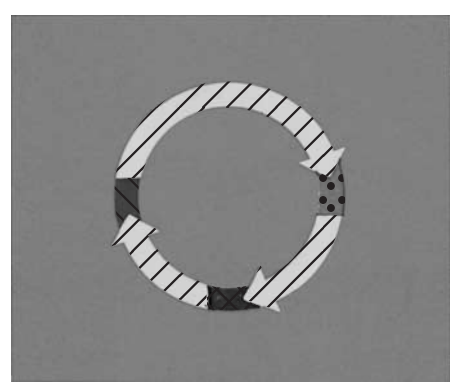
95



92



96



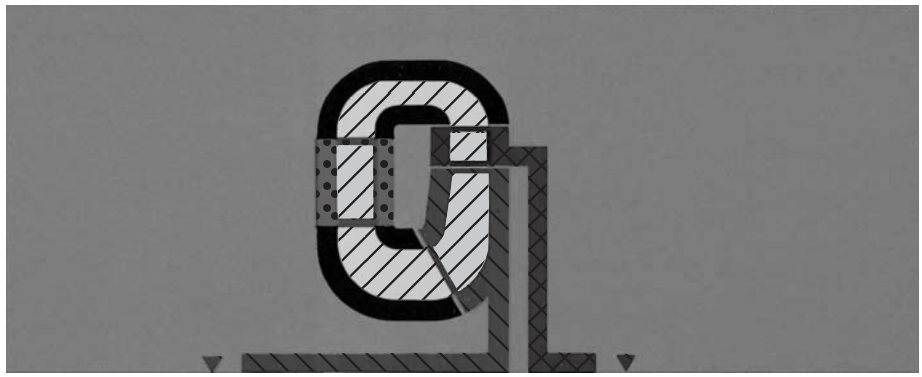
FIGS. P-89, P-93, P-97 Power absorbed by converters is constant. (Source: J. M. Voith GmbH.)

FIGS. P-90, P-94, P-98 Power absorbed by converters drops at certain speeds. (Source: J. M. Voith GmbH.)

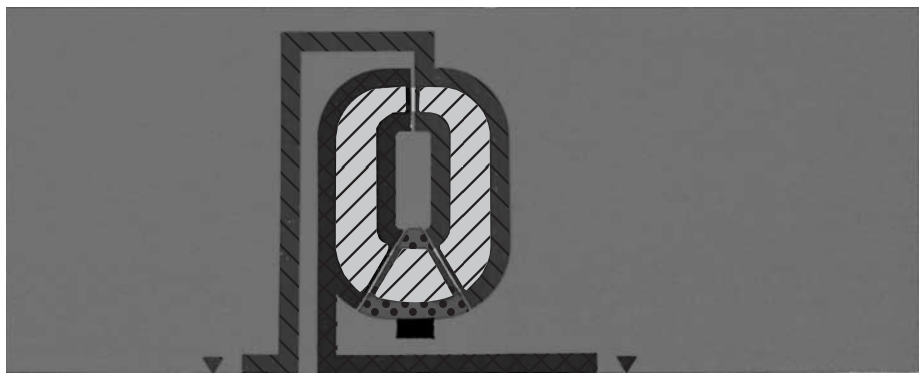
FIGS. P-91, P-95, P-99 Power absorbed by converters drops. (Source: J. M. Voith GmbH.)

FIGS. P-92, P-96, P-100 Power absorbed by converters increases. (Source: J. M. Voith GmbH.)

97



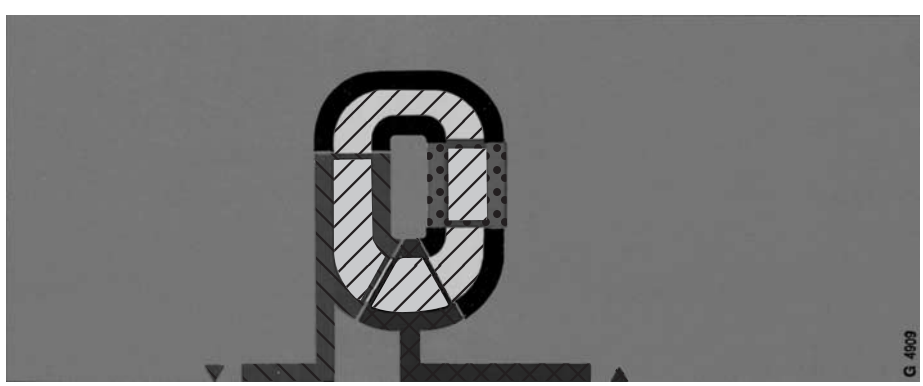
98



99



100



the plant designer and the coupling manufacturer at the beginning of the project.

See also Figs. P-101 through P-103.

Scope for varying the characteristics and the connection of hydrodynamic power transmissions

The simplest version of a hydrodynamic power transmission has a constant operating-fluid filling. The converter characteristics are rigid and cannot be changed.

Influencing the converter circuit. (See Figs. P-104 through P-107.) Changing the characteristics of the converter requires suitable means in the hydraulic circuit. A slide valve may be used to throttle the flow of the circulating fluid. (See Fig. P-105.) With the slide valve closed, the startup of the motor or engine is facilitated. When the power flow is cut off, engagement of mechanical couplings will be eased.

Another possibility to change the characteristics is provided by adjustment of the guide blades, i.e., change of the position of the guide blades (reaction member). (See Fig. P-106.) Changing the guide-blade position varies the ratios of output/input speed and output/input torque.

Engaging and disengaging of converters can also be achieved by the filling and emptying principle. (See Fig. P-107.) Furthermore, by variation of the oil filling the transmitted power can be adjusted steplessly.

By applying this principle several converters can be used for opposite directions of running (hydrodynamic forward-reverse transmissions) and/or several speed ranges (hydrodynamic multicircuit transmission).

Influencing the circuit by using mechanical elements. (See Figs. P-108 through P-111.) The so-called "Trilok" converters represent a special type of converter: a freewheeling arrangement, which becomes effective at a well-defined speed ratio to prevent the guide blades from taking up a reaction torque, which converts the torque converter into a hydraulic coupling.

In some cases, it may be desirable to bridge the converter after the equipment has been run up to speed by the hydrodynamic converter. This can be accomplished by providing a direct mechanical drive.

If, for the prevailing operating conditions, the torque multiplication provided by the converter should not meet requirements, the operating range can be extended by a mechanical gearbox fitted behind the converter.

In the 2-channel arrangement of differential converters, the portion of the input power that is transmitted hydrodynamically decreases with increasing running speed while that transmitted mechanically increases until eventually the whole power is transmitted entirely mechanically. Differential converters are often followed by planetary gears.

Case studies*

Case study 1: Hydrodynamic variable-speed couplings in the petrochemical industry. In the course of recent years, use of hydrodynamic variable-speed couplings have proven to be an excellent means of controlling the speed of crude-oil and liquid-gas pumps in many production areas (including the Middle East and the North Sea). More than 150 geared variable-speed couplings and variable-speed couplings made by this

* Source: J.M. Voith GmbH, Germany.

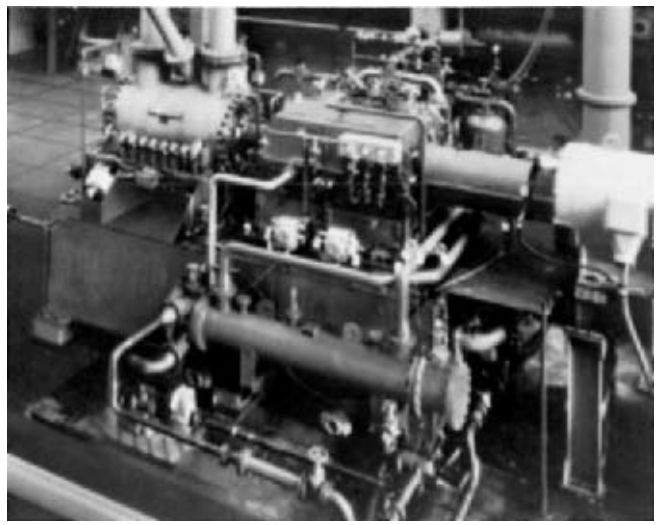


FIG. P-101 Compressor drive in a coal-pilot gasification plant. Type R18KGS—API, n_e = 1490 rpm, P = 7.3 MW, n_a = 10,548 rpm. (Source: J. M. Voith GmbH.)

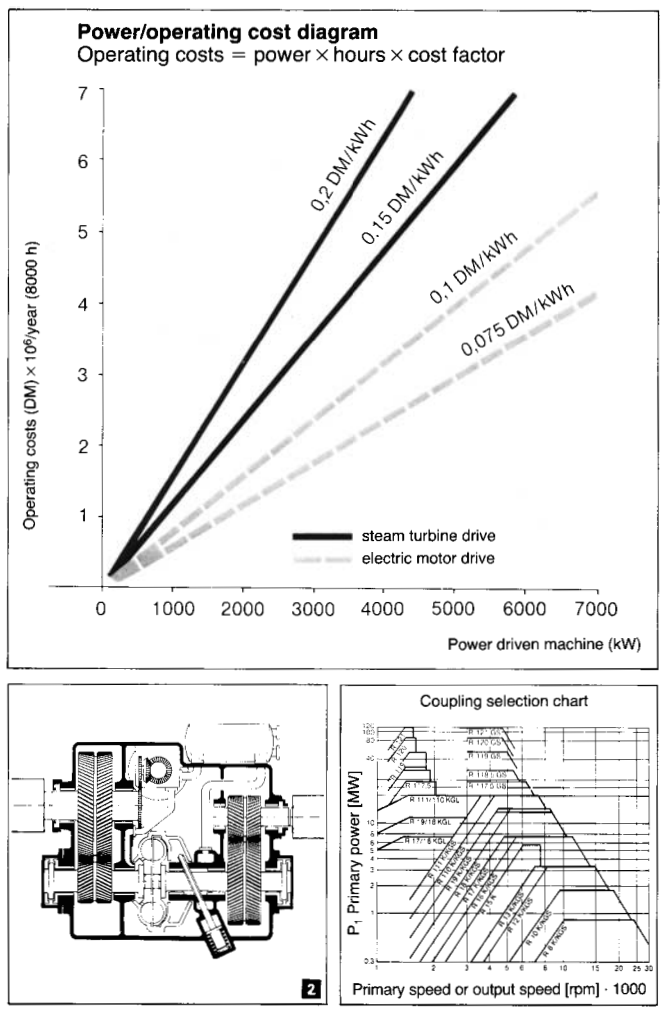


FIG. P-102 Design of a Voith hydrodynamic geared variable-speed coupling R1KGS (API) and coupling selection chart. (Source: J. M. Voith GmbH.)

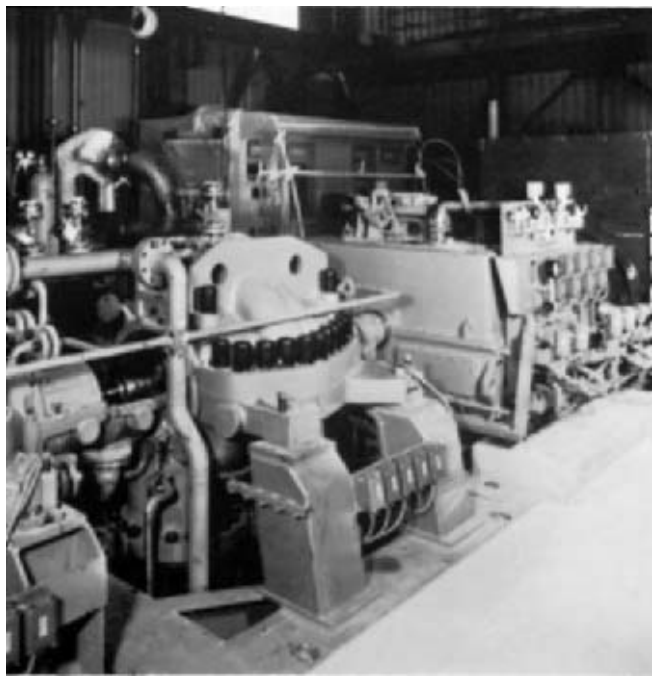


FIG. P-103 Compressor drive in an ethylene process plant. Type R16KGS—API, n_e = 1490 rpm, P = 2.9 MW, n_a = 12,604 rpm. (Source: J. M. Voith GmbH.)

information source are in use in crude-oil loading stations and pipeline pumping stations—often working under extremely severe environmental and operating conditions.

Operating periods of 15 years with as many as 50,000 hours per unit are not unusual. Transmission powers vary between 15 and 11,000 kW with pump speeds varying between 900 and 8000 rpm.

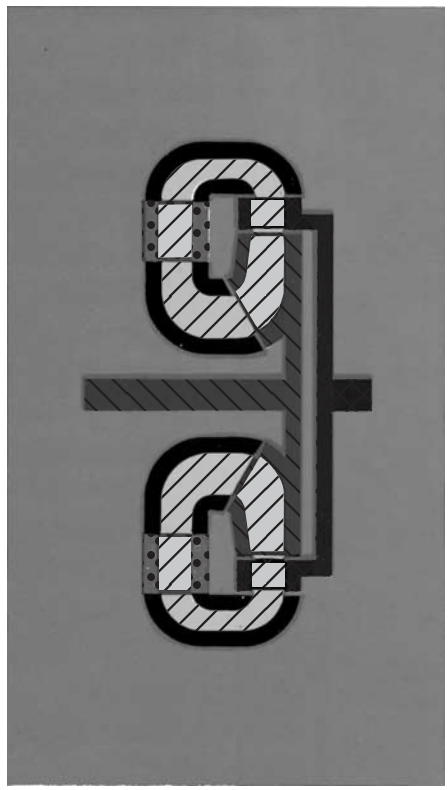
As previously noted, the variable-speed coupling made by this OEM is a hydrodynamic coupling (turbocoupling) that works on the Föettinger principle. The mechanical energy delivered by the drive motor is converted to kinetic energy by a pump impeller (primary wheel) and back to mechanical energy by a turbine wheel (secondary wheel) that is connected with the driven machine. Since there is no mechanical contact between the primary wheel and the secondary wheel, power is transmitted without wear. Mineral oils with suitable additives are used as operating fluid. See Fig. P-112.

The quantity of fluid in the variable-speed coupling can be varied at will during operation by means of a scoop tube and thus it is possible to adjust the power output level to meet the requirement at any time. The speed of the driven machine can be infinitely varied.

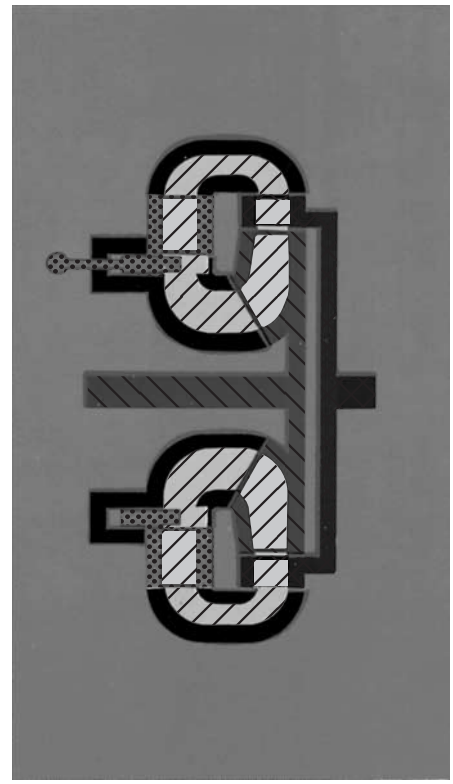
Various arrangements—with or without a mechanical gear stage—provide for adapting the coupling to the required speed.

Type SVNL. Type SVNL variable-speed couplings can be supplied either with plain or roller bearings. A particular design feature is the division of the coupling housing by a horizontal flange. This is necessary where the internal parts are large in size and it is often called for on smaller couplings in order to accommodate special installation conditions at the location concerned.

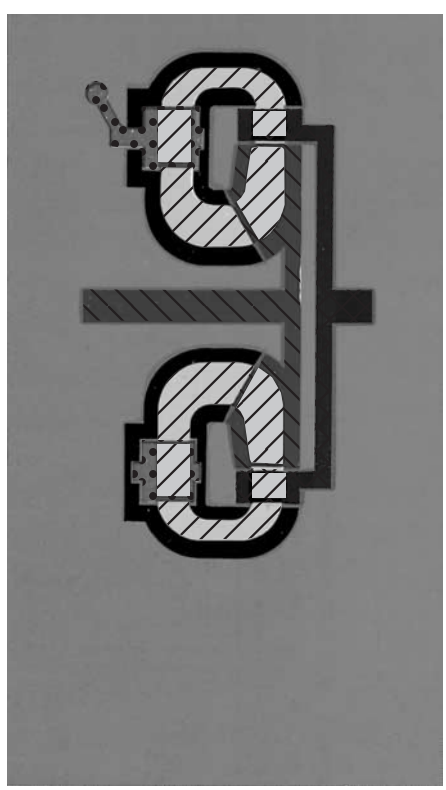
Up to size 487II (see Table P-13), these couplings are supplied as standard with an electrically driven oil pump. From size 487III upward, this pump is driven



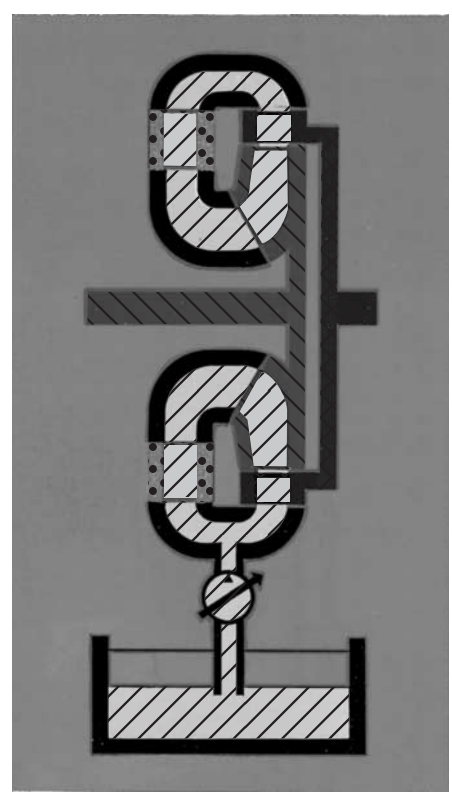
104



105



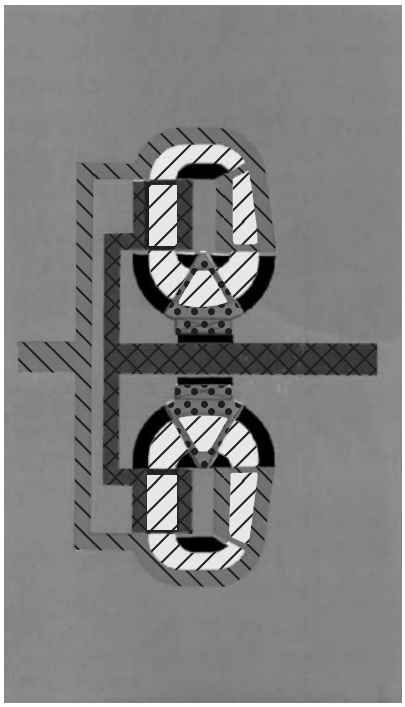
106



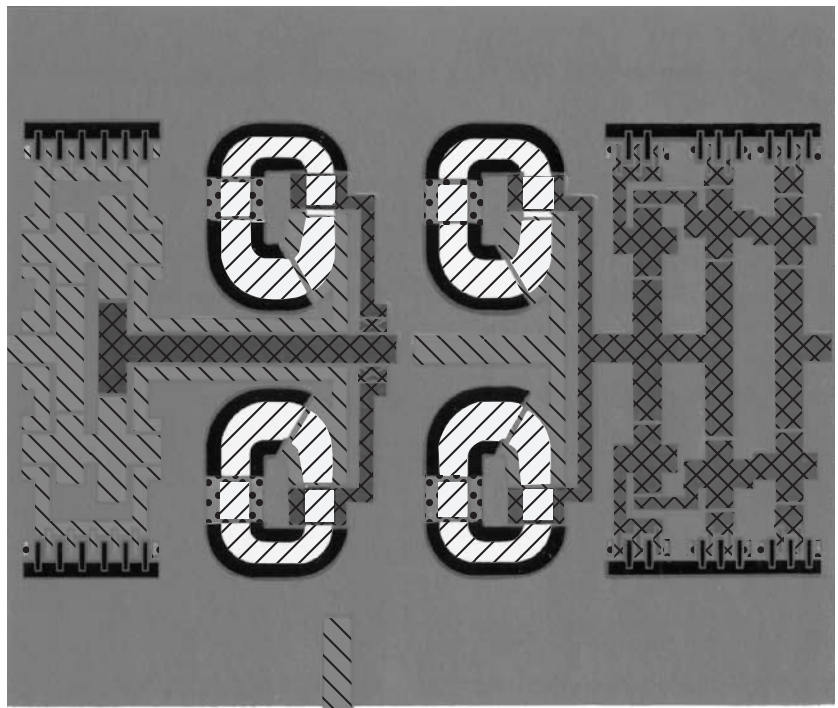
107

FIG. P-104 The converter circuit. (Source: J. M. Voith GmbH.)

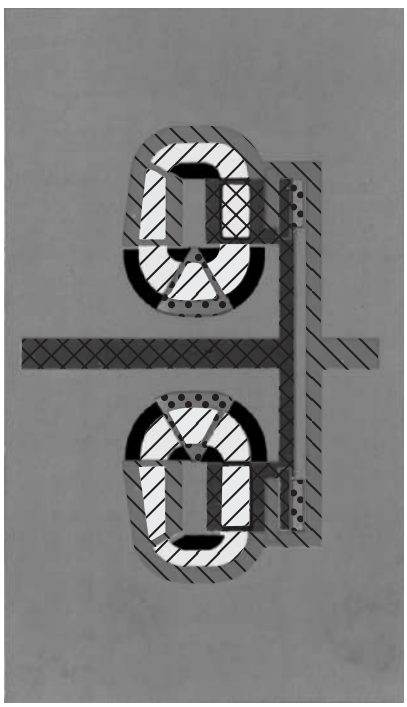
FIGS. P-105, P-106, P-107 Three methods of influencing the converter circuit. (Source: J. M. Voith GmbH.)



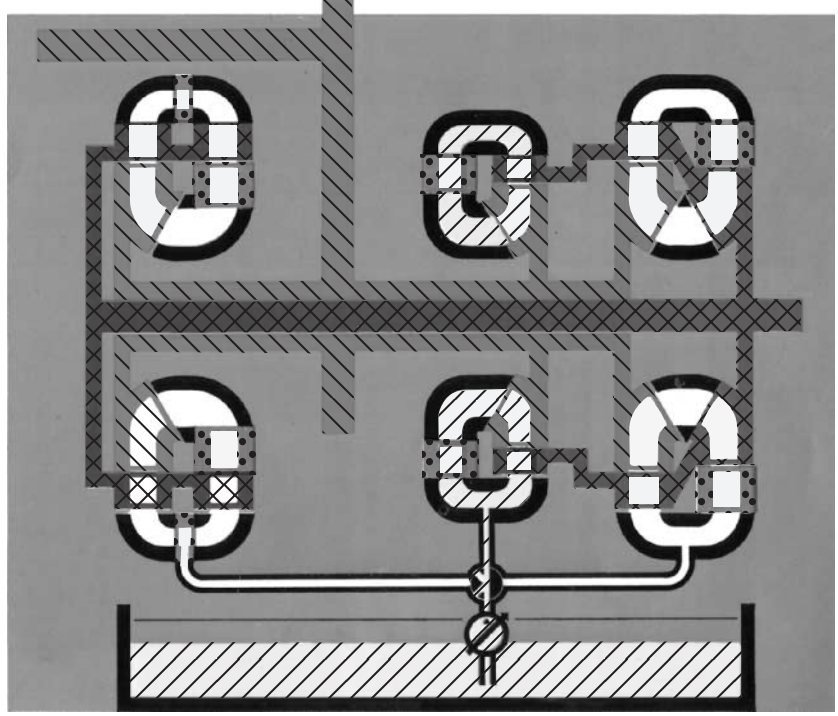
108



109



110



111

FIGS. P-108, P-109, P-110, P-111 Influencing the converter circuit using mechanical elements. (Source: J. M. Voith GmbH.)

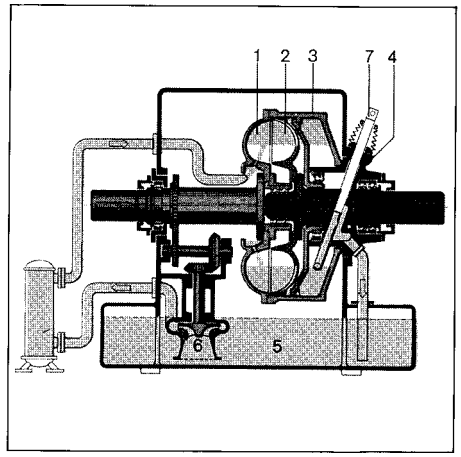


FIG. P-112 A simplified cross-section of a type SVNL hydrodynamic variable-speed coupling (self-supported, with the working pump gear driven from the primary shaft). 1, primary wheel; 2, secondary wheel; 3, shell; 4, scoop tube housing; 5, oil sump; 6, oil circulation pump; 7, scoop tube. (Source: J. M. Voith GmbH.)

TABLE P-13 Coupling Types in Service

Quantity	Type	Power of Each Pump	Pump Speed
10	562SVLIII	2700 kW	3495 rpm
3	R17K	3134 kW	4102 rpm
3	R15K550	4000 kW	6000 rpm
10	866SVNL	1460 kW	1445 rpm
2	R10K-API	802 kW	8950 rpm
4	R12K315-API	512 kW	5295 rpm
2	R12K315-API	683 kW	5026 rpm
3	R15K550-API	2980 kW	6200 rpm
2	464SVL-API	1193 kW	3585 rpm
3	464SVL-API	1388 kW	3564 rpm
1	R12K315-API	683 kW	5026 rpm

directly from the primary shaft by means of a mechanical gear. Deviations from this standard arrangement are sometimes necessary and can be provided in order to meet particular plant conditions.

Connections to the motor and to the driven machine are affected by flexible connecting couplings that allow for angular displacement. Figure P-113 illustrates transmission capacities in relation to speed.

Type SVL. Type SVL variable-speed couplings are suitable for driving fast-running machines of high power, such as boiler feed pumps, centrifugal compressors, and pipeline pumps (see Figs. P-114 and P-115).

To meet such requirements, the design features of this type of coupling include a strong cast housing designed to minimize vibration, plain bearings, use of steel castings for the coupling wheels, controlled circulation of the working fluid, and a lubrication system with a separate lubricating oil pump.

The working circuit includes the following major components: a scoop tube, scoop tube housing, heat exchanger, control valve, and return flow to the coupling. It follows that circulation is effected entirely by utilizing the dynamic pressure

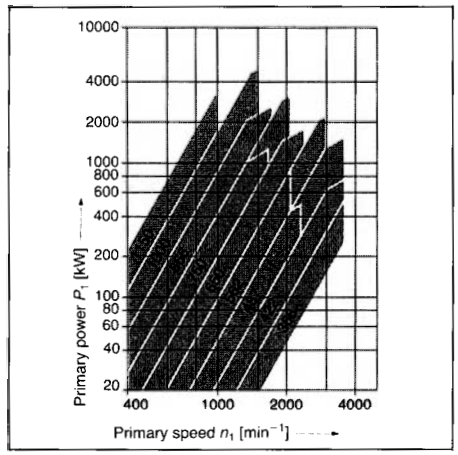


FIG. P-113 Power diagram for type SVNL variable-speed couplings. (Source: J. M. Voith GmbH.)

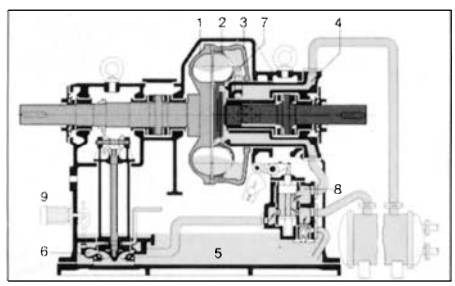


FIG. P-114 Cross-section of a type SVL hydrodynamic variable-speed coupling. 1, primary wheel; 2, secondary wheel; 3, shell; 4, coupling housing; 5, oil sump; 6, oil circulation pump; 7, scoop tube; 8, circulation control valve. (Source: J. M. Voith GmbH.)

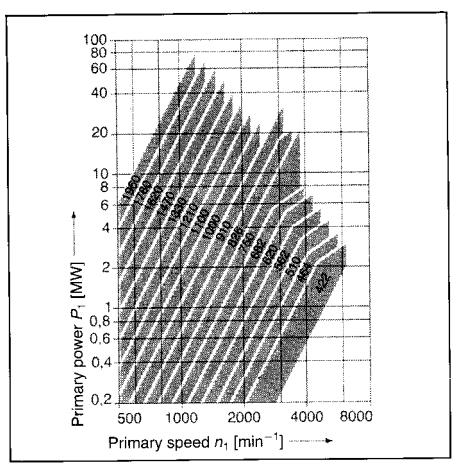


FIG. P-115 Power diagram for type SVL variable-speed couplings. (Source: J. M. Voith GmbH.)

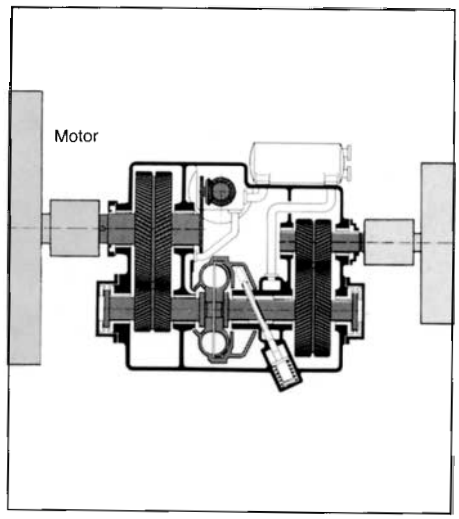


FIG. P-116 Cross-section of a type R KGS geared variable-speed coupling. (Source: J. M. Voith GmbH.)

generated at the mouth of the scoop tube. It is only changes in the quantity of oil in the coupling that are affected by the working pump and the control valve. This valve is actuated by a cam disk that is shaped to match the load characteristic. It provides optimal adjustment of the output of the oil circulation pump.

After matching the oil type, volume, and pressure, the lubricating oil supply to the motor, driven machine, and connecting couplings can be provided by the lubrication system of the variable-speed coupling.

Where particularly high output speeds are required, the variable-speed turbocoupling is equipped with two single-stage gears, one on the input side and one on the output side of the coupling. The result is the geared variable-speed coupling type R KGS (see Figs. P-116 and P-117).

Type RK. Geared variable-speed couplings combine a mechanical gear with a hydrodynamic variable-speed coupling in a common enclosure whose bottom half serves as an oil sump.

Starting from the basic concept R1.K (Figs. P-118 and P-119), comprising a speed increasing gear on the input side of a variable-speed coupling, various types have been developed to match such applications as infinitely variable-speed control of fast-running, high-powered driven machines such as boiler feed pumps or, with a speed-reducing gear on the output side, for the speed control of low-speed machines such as coal pulverizers, induced draught fans, and crude-oil pumps.

Geared variable-speed couplings are constructed for powers of up to 60,000 kW and more.

Case study 2: Pipeline pump stations and loading stations in Saudi Arabia. In the pumping stations 1 to 8 (Fig. P-120), a total of 52 geared variable-speed couplings are in service—often under extreme environmental and operating conditions. See also Figs. P-121 and P-122.

The advantages offered by the use of geared variable-speed couplings are:

- Energy savings compared with the throttling of centrifugal pumps when operating under partial load
- Design appropriate to the relevant explosion-proof regulations

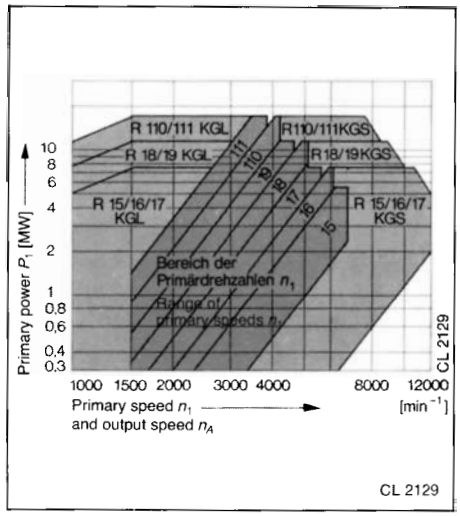


FIG. P-117 Power diagram for types R KGS and R KGL geared variable-speed couplings. (Source: J. M. Voith GmbH.)

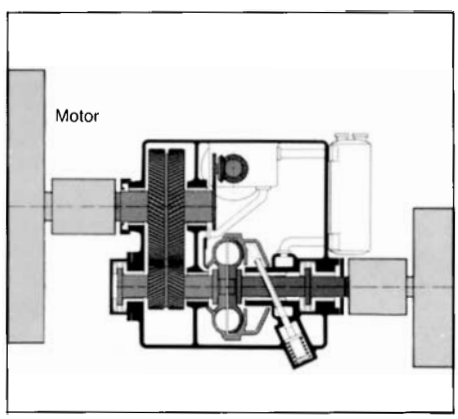


FIG. P-118 Cross-section of a type R1.K geared variable-speed coupling with speed increasing gear on the input side of a variable-speed turbocoupling. (Source: J. M. Voith GmbH.)

- Increased service life of the driven machine and other plant elements as a result of infinitely variable-speed control resulting in reduced wear
- Motor starting under no load followed by controlled load application to requirements
- Gentle acceleration of the driven machine by adjustment of the coupling filling
- Vibration and shock damping
- Wear-free power transmission by way of an operating fluid
- Disconnection of power transmission without stopping the prime mover
- Lubricating oil supplied to the motor and driven machine from the oil system of the geared variable-speed coupling

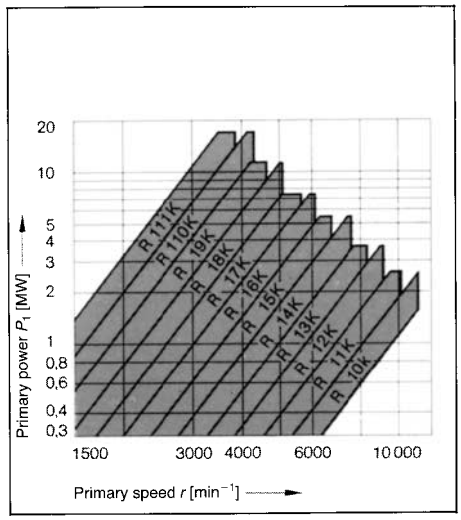


FIG. P-119 Power diagram for type R1.K geared variable-speed couplings. (Source: J. M. Voith GmbH.)

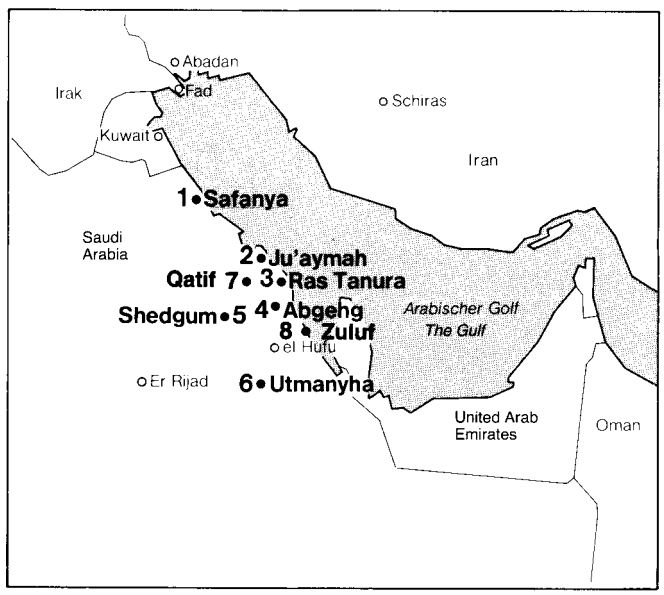


FIG. P-120 Pumps station sites in Saudi Arabia using geared variable-speed couplings. (Source: J. M. Voith GmbH.)

- Compact design with the gear units and the variable-speed coupling being contained in a single housing
- Low space requirement
- Economy in operation due to energy savings and also as a result of the motor and driven machine being subjected to less stress

The following features are of particular interest for the drive of crude-oil and liquid-gas pumps:

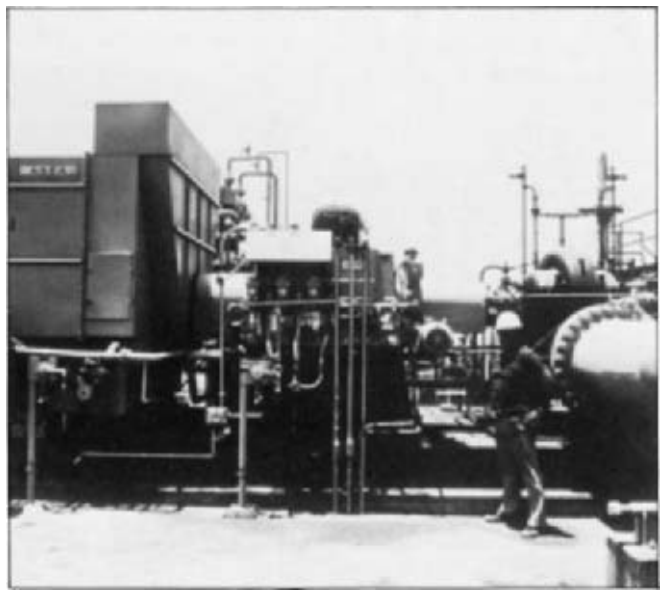


FIG. P-121 A bulk oil cargo terminal in the desert with a Voith type R17KGL geared variable-speed coupling. P_a = 7,000 kW, n_a = 1372 rpm. (Source: J. M. Voith GmbH.)



FIG. P-122 A bulk cargo oil loading station in Juaymah with a Voith R19KGL geared variable-speed coupling designed for the following data: P_a = 11,000 kW, n_a = 1800 rpm. In this station there are 11 units of the same size that are connected in series. They serve to pump crude oil to an offshore oil terminal. (Source: J. M. Voith GmbH.)

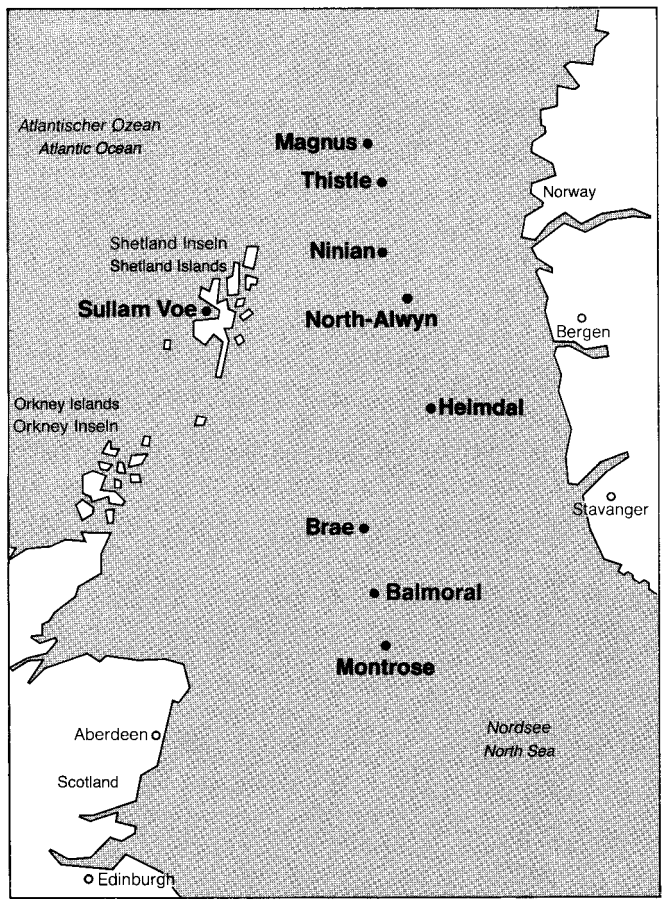


FIG. P-123 Some offshore platforms in the North Sea that use variable-speed turbocouplings. (Source: J. M. Voith GmbH.)

1. The possibility of adjusting the pump delivery by speed control, using a relatively simple control loop.
2. The possibility of keeping a motor running continuously and stopping the pump by emptying the coupling.

As is known, in bulk cargo oil terminals, the pump delivery must be kept constant irrespective of any variation in suction or delivery heads, and it is also necessary to keep the motors running during brief interruptions in delivery (such as on changing from one tanker to another) in order to avoid high peak starting currents. This is possible simply by draining the coupling. The secondary wheel is stopped by means of a brake.

Case study 3: Offshore pumping stations in the North Sea. A total of 16 variable-speed turbocouplings and geared variable-speed turbocouplings from this OEM are installed on a number of offshore platforms in the North Sea (Fig. P-123). Here the crude oil is pumped from a central platform through pipelines to the oil tanks on the Shetland Islands.

Speed control is necessary in order to maintain the flow against pipe friction that varies in relation to the quantity of oil delivered. Viscosity variations can also be compensated for by varying the pump speed.

The coupling types in service are depicted in Table P-14.

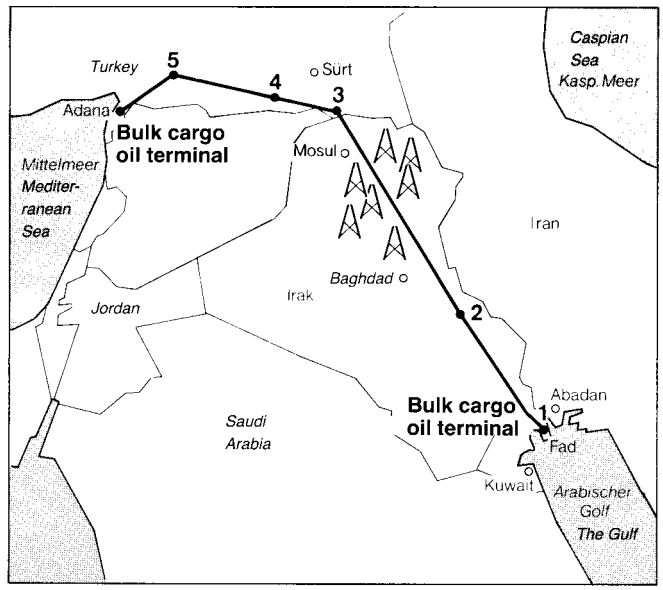


FIG. P-124 The course of the Iraq–Turkey pipeline. (Source: J. M. Voith GmbH.)

TABLE P-14 Information Source's New Installations Abroad

Application	Qty.	Type	Power	Speed
Oil pipeline in Russia	12	487 SVTL	1169 kW	2879 rpm
Water injection pumps in Holland	3	487 SVTL	820 kW	2908 rpm
Water injection pumps in Saudi Arabia	3	R19K600	6893 kW	4311 rpm
Pipeline Naftovod	6	R17K	2500 kW	4500 rpm
in Yugoslavia	4	562 SVNLG	1524 kW	2915 rpm
	4	487 SVNLG	800 kW	2917 rpm

Case study 4: Iraq–Turkey pipeline. To overcome pipe friction and changes in altitude along the path of a pipeline, it is generally necessary to install pressure-boosting stations.

In the case, for example, of the Iraq–Turkey pipeline with a length of around 1000 km, there are five pumping stations that are each equipped with four variable-speed couplings (Voith type 620SVLIII), each transmitting a power of 3400 kW at a speed of 2885 rpm. (See Figs. P-124 and P-125.)

At the bulk cargo oil terminal No. 1 in Fad, an additional two variable-speed couplings type 1,000SVNLIIV are in service, each transmitting 3000 kW at a speed of 1455 rpm.

General cases of the installations are listed in Tables P-14 and P-15 and shown in Figs. P-126 through P-128.

Case study 5: Energy savings example—boiler feed pumps. Energy saving on boiler feed pumps. Speed adaptation of boiler feed pumps by replacing the gear stages in the geared variable-speed turbocoupling. (See Figs. P-129 and P-130.)



FIG. P-125 A Turkish pumping station on the Iraq–Turkey pipeline with a Voith type 620SVL variable-speed turbocoupling with an oil/air oil cooler. (Source: J. M. Voith GmbH.)

Design

Motor power required: 8040 kW

Pump power: 7600 kW

Pump speed: 6500 rpm

This design was made in the planning stage considering most various construction variables with regard to piping, etc.

Maximum operating point

Motor power: 6160 kW

Pump power: 5300 kW

Pump speed: 5700 rpm

After some time it turned out that this operating point is never exceeded.

Optimized condition

Motor power: 5460 kW

Pump power: 5300 kW

Pump speed: 5700 rpm

Difference of motor powers: 6160 kW – 5460 kW = 700 kW

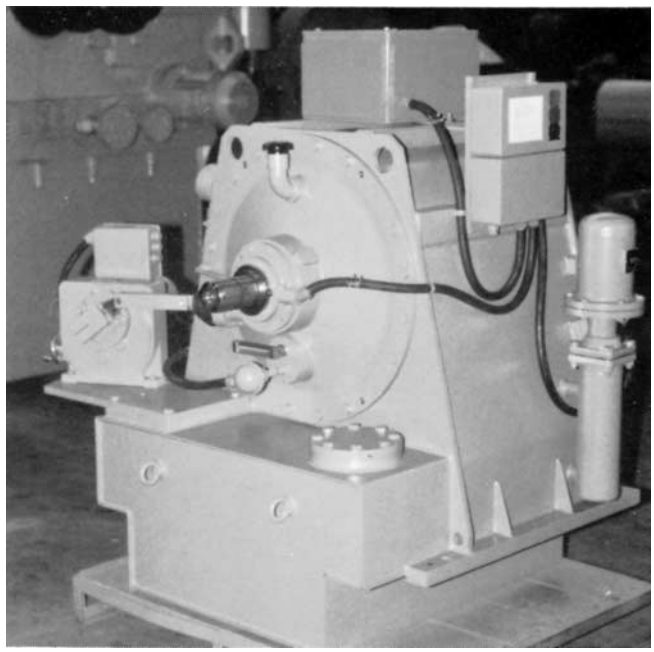


FIG. P-126 A Voith type 487SVTL variable-speed turbocoupling for a pipeline in Russia. (Source: J. M. Voith GmbH.)



FIG. P-127 A model of the water injection station at Qatif in Saudi Arabia. (Source: J. M. Voith GmbH.)



FIG. P-128 2 × R16K with gear units E45 arranged on the output side with air-oil cooler for driving a crude oil loading pump in Ras Tanuara, Saudi Arabia. Power consumption of pump 4000 kW, pump speed 1200 rpm. (Source: J. M. Voith GmbH.)

TABLE P-15 Information Source's Installations in West Germany

Refinery	Coupling Type	Rated Power	Speed
Vohburg	1 × 274SVN	30 kW	2980 rpm
Vohburg	2 × 316SVN	40 kW	2980 rpm
Vohburg	4 × 366SVN	15 kW	1450 rpm
Vohburg	2 × 422SVN	45 kW	1450 rpm
Vohburg	6 × 464SVL	1016 kW	4180 rpm
Vohburg	2 × 487SVNL	376 kW	2890 rpm
Vohburg	2 × 650 SVN	50 kW	740 rpm
Godorf	3 × 510SVL	540 kW	2980 rpm
Speyer	4 × 422SVN	258 kW	2980 rpm
Speyer	3 × 366SVN	120 kW	2980 rpm
Uerdingen	2 × 487SVN	55 kW	1450 rpm
Uerdingen	2 × 422SVNRI	37 kW	1470 rpm
Bucholtwelmen	3 × 422SVNL	255 kW	2980 rpm

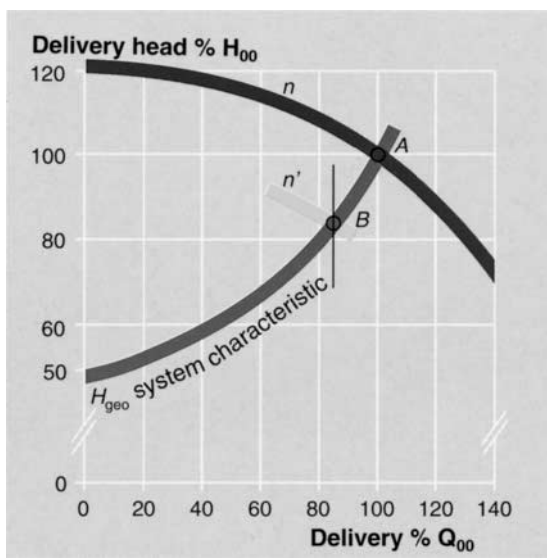


FIG. P-129 Q-H diagram with system characteristic and pump characteristic—design point A and speed-regulated condition B. (Source: J. M. Voith GmbH.)

This represents the continuous energy saving during operation. With an electricity price of 0.12 DM/kW hour and just 6000 operating hours per year, there is an annual saving of 500,000 DM.

Case study 6: Speed regulation using hydrodynamic variable-speed couplings in place of throttle regulation. This case is a summary and discussion of the various economic factors that influence the choice between speed and throttle regulation. A power station boiler feed pump is used as an example but the argumentation equally applies to all other drives with a parabolic or near-parabolic relationship between torque and speed, e.g., all types of circulating pumps and fan drives.

Principle of throttle and speed regulation of boiler feed pumps in power stations. As Fig. P-131 shows, the operating point is the point where the boiler resistance curve and the characteristic pump curve (n = const.) intersect. To operate at a different point, either the boiler resistance curve or the pump curve must be

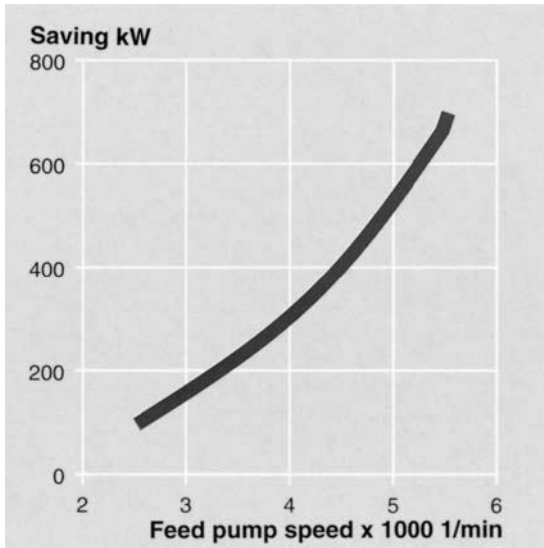


FIG. P-130 Energy-saving curve with optimized ratio. Example: boiler feed pump drive. (Source: J. M. Voith GmbH.)

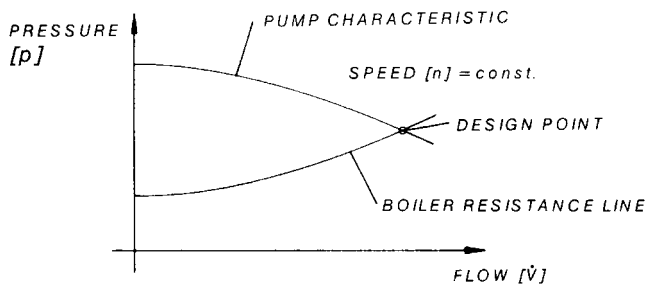


FIG. P-131 Pump characteristics and boiler resistance curve. (Source: J. M. Voith GmbH.)

changed. This has led to the development of two regulating methods that are now in general use:

a. By installing a throttle valve, the boiler resistance curve is brought up to the required higher pressure level (throttle regulation).

b. By varying the pump speed, the pump curve is changed (speed regulation).

Design of the boiler feed pump. The maximum continuous load of boiler must be the basis for determination of the design point of the feed pump (see Fig. P-132).

1 = Boiler resistance curve in constant pressure operation
2 = Boiler resistance curve in varying pressure operation
3 = Pump curve of the half-load pump
4 = Pump curve of the full-load pump
5 = Maximum continuous rating of the boiler
6 = Design point
Pk = Maximum permissible boiler pressure

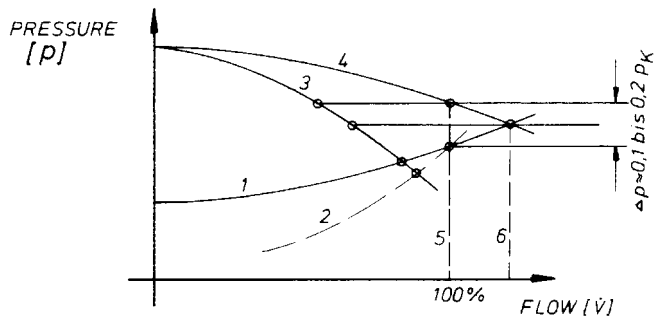


FIG. P-132 Curves required for boiler feed pump design. (Source: J. M. Voith GmbH.)

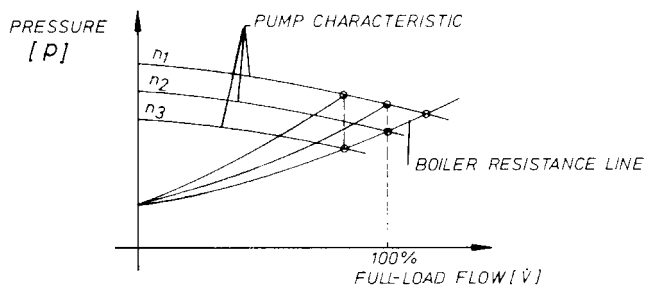


FIG. P-133 Optimizing pump performance. (Source: J. M. Voith GmbH.)

The pressure head of the feed pump must be so sized that the geodesic head can be overcome and the required pressure increase, flow rate, and pressure losses plus a 10 percent extra (related to the maximum boiler operating pressure) can be attained.

For completeness, Fig. P-132 also has the curves for constant and varying pressure operation. The fact that the position of these curves differs has no real effect on the subject under discussion. Furthermore this introductory section only deals with the basic outline of the considerations that affect the choice of a feed pump.

Comparison between speed regulation and throttle regulation

OPTIMUM USE OF PUMP. Figure P-133 shows that with throttle regulation, the throttle valve has to raise the boiler resistance curve even during full-load operation. Each flow rate reduction required can only be achieved by artificially further raising the boiler resistance curve. The delivery throttling therefore decreases the unit's efficiency since the power throttled off cannot be recovered.

With speed regulation of the pump only a relatively small speed change is required to cover the whole boiler load, and the pump efficiency does not change very much. Furthermore when the delivery rate drops the speed and pump loading also drops, this means extended pump life due to reduced wear.

INCREASED PLANT LIFE. It is a general rule for all pump units (not only in power stations) that speed regulation has a beneficial effect on performance and service life, because the operating conditions in the pump are more favorable. If high-pressure pumps in particular are run for a long period with the flow heavily throttled back, overheating can occur leading to wear on the races, shaft bushes, seals, etc. The other units around the pump are also loaded in that energy is put

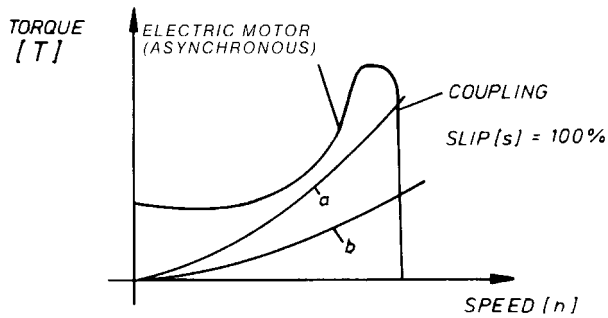


FIG. P-134 Torque versus speed for an asynchronous motor. (Source: J. M. Voith GmbH.)

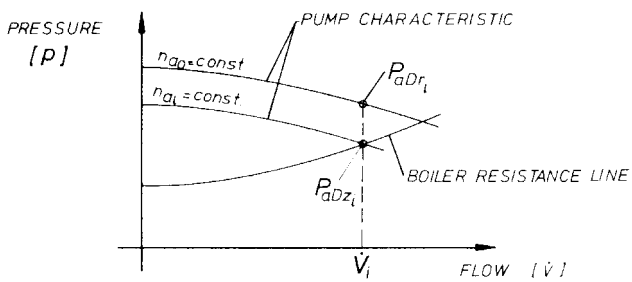


FIG. P-135 Flow versus pressure pump characteristic. (Source: J. M. Voith GmbH.)

into the flow and then destroyed by the throttle valve or transformed into heat; of course the valve is itself very heavily stressed.

ENERGY SAVINGS. As already mentioned, speed regulation saves energy that is dependent on the load schedule and the pump and plant curves. A detailed, quantitative analysis of these savings follows under "Efficient Energy Utilization with Speed Regulation."

EASIER MOTOR STARTUP. When an electric motor starts up, problems occur that come under the heading breakaway torque, startup current, and motor design in accordance with the startup conditions. The hydrodynamic coupling represents a proven means of overcoming these problems.

Even the most unfavorable case, namely a blocked driven machine, would only subject the motor to a load that rises to the square of the motor speed (see Fig. P-134). This assumes the coupling is full (curve *a* in Fig. P-134) whereas in practice the coupling filling is set to minimum during startup (curve *b*). In the case of extreme requirements, the favorable behavior of the coupling is backed up by an integrated hydraulic device, usually called a rapid start device. This means that the motor can have a large acceleration capability at its disposal and really only needs to overcome the inertias of the motor armature and coupling wheels. Controlled loading by automatic filling of the coupling does not start until the most favorable rated torque on the motor curve has been reached.

Since the hydrodynamic coupling is the connecting member between the motor and the driven machine, it influences not only the motor-side startup conditions but also those of the driven machine. By connecting the coupling's scoop tube to suitable control instruments or closed loops, it is possible to carry out complicated operations, such as torque limitation, acceleration limitation, and startup time

reduction. In the case of boiler feed pumps the ability of the standby unit to start up very rapidly plays a major role in protecting the whole plant. This is one of the reasons why the hydrodynamic variable-speed coupling has become such an important element in power stations, since this condition is met simply by connecting up the scoop tubes of two couplings in parallel and adapting the hydraulic or electric control circuit accordingly.

The dynamic behavior of coupling together with the units they are connected to is by now very well known, so the designer can be supplied with suitable data.

SEPARATION OF DRIVING AND DRIVEN MACHINES. Separation of the driving and driven machines can mean vibration and shock separation or absorption; it can also include the possibility of power separation by draining the coupling. At all events the hydrodynamic variable-speed coupling can optimally satisfy both requirements that are of major importance as far as operating performance is concerned. Because of the present-day speed and performance demands made on machines, shock and vibration absorption is an important factor, not only when reciprocating with rotating machine elements work together. The drainability of the coupling is mostly used to protect the prime mover, i.e., to relieve the load on the motor, to avoid long-duration startup current peaks, and to reduce the load on the mains. It is often the case that large motor units are not shut down when the driven machine is to be stopped and that frequent motor startups, and thus mains loading, are avoided simply by draining and filling of the coupling. This means the plants are more economical because of increased protection of the electrical and mechanical drive elements even with very high startup frequency.

This subsection was a summary of the various factors that illustrate the advantages of speed regulation using a hydrodynamic variable-speed coupling in feed pump drives and that can be applied to these drives. Some of the factors mentioned affect economy of operation indirectly rather than directly, e.g., reduced breakdown rate, different design data for the whole plant, operating safety, and flexibility.

Efficient energy utilization with speed regulation. To calculate the energy savings achieved by speed regulation using a hydrodynamic variable-speed coupling, the unit resistance curve, the characteristic pump curve for various speeds, and the load schedule must be known. The unit and pump curves are specified parameters and thus known within certain tolerances. This is not the case with the load cycle, which depends on the interrelationship with other power stations, the power increase or requirements over a large number of years, the block size, etc. It is very difficult to make correct assumptions because the interdependent factors are highly complex and the developments have to be assessed for a long time ahead because of the normal amortization periods for large plants.

However the following calculation shows that speed regulation pays for itself very quickly simply due to the throttle energy saved. It is therefore the author's opinion that a decision in favor of speed regulation is not really as difficult as it first appears.

The power consumption of the pump at the individual operating points i corresponds to the following formula; the separate values for V_i and p_i out of the unit resistance curve and the pump curve can be deduced.

$$P_a = \dot{V}_i \cdot \frac{p_i}{\eta_i} \tag{P-5}$$

P_a = pump power consumption with throttle regulation
$\dot{V}_i$ = oil flow
P_i = pressure
η_i = pump efficiency

The energy difference per pump unit is as follows:

$$\Delta E_{total} = \sum_{i=1}^{n} (P_{Dri} - P_{Dzi})\, i \cdot t_i \qquad \text{(P-6)}$$

P_{Dri} = power requirements with throttle regulation
P_{Dzi} = power requirements with speed regulation
t_i = startup frequency at flow V_i (h/year)

where the following applies:

$P_{Dri} = P_{aDri}$ (pump power consumption with throttle regulation; see Fig. P-135).

Taking into consideration the efficiency and slip of the fluid coupling, the resultant power requirements with speed regulation are:

$$P_{Dzi} = P_{aDzi} \cdot \frac{n_{ao}}{n_{ai}} \cdot \frac{100}{100 - s_{min}} (1 + a) \qquad \text{(P-7)}$$

P_{aDzi} = pump power consumption with speed regulation (see Fig. P-136)
n_{ao} = (maximum) pump speed with throttle regulation
n_{ai} = pump speed with speed regulation
s_{min} = minimum slip of fluid coupling in percent
a = symbol for mechanical efficiency of coupling (0.02 – 0.03)

Since Eq. (P-7) is an approximate formula for what is in reality a difficult physical process, Eq. (P-7) should only be used as a rough calculation. Values a and s_{min} in Eq. (P-7) are average values reflecting the fluid coupling's efficiency, which is in fact dependent on the coupling size and type and the load. However the formulae are accurate enough for our purposes and, being simplified, make it easier to understand the basic interrelationships.

NUMERICAL EXAMPLE. The previous description of the economical factors is here further clarified using a numerical example. Figure P-136 illustrates the following calculations.

The example is based on a real medium size power station; some values that are valid for this particular power station have been modified to make them of more general application. The author believes that the data were selected to the advantage of throttle regulation; this is dealt with in more detail in the following section where the design values are discussed. This example is mainly meant to be an example that can be quickly amended by introducing further data.

The throttle curve in Fig. P-136 is, because of its flat rise, more or less an ideal throttle regulation curve.

Such a curve cannot always be achieved because in practice too flat throttle curves mean instable pump behavior. Cases where the throttle curve does rise more acutely mean higher energy losses for throttle-regulated plants.

The pump is normally designed so that maximum efficiency is achieved at the point where the pump most often operates; this we have called normal point B_N. This applies both to throttle and speed regulation. However, with speed regulation, the design efficiency can be maintained with very little deviation over a wide regulating range whereas with throttle regulation the efficiency drops sharply as the throttle regulation increases.

Three operating points are given in Table P-16.

The following are assumed: feed water temperature of 165°C, specific gravity of 900 kg/m^3, and motor speed of 1480 1/min.

The appropriate coupling for these values is the hydrodynamic variable-speed geared coupling type R 16 K-550. For clarity arrangement sketches have been added

TABLE P-16 Operating Points

Operating Point	m (kg/s)	p (bar)	n_a (1/min)	P_a (kW)	t (h/year)
Max. point B_M	172.5	209	5980	5348	50
Norm. point B_N	155	199.7	5870	4417	5000
Min. point B_S	122	182.2	5500	3236	2800

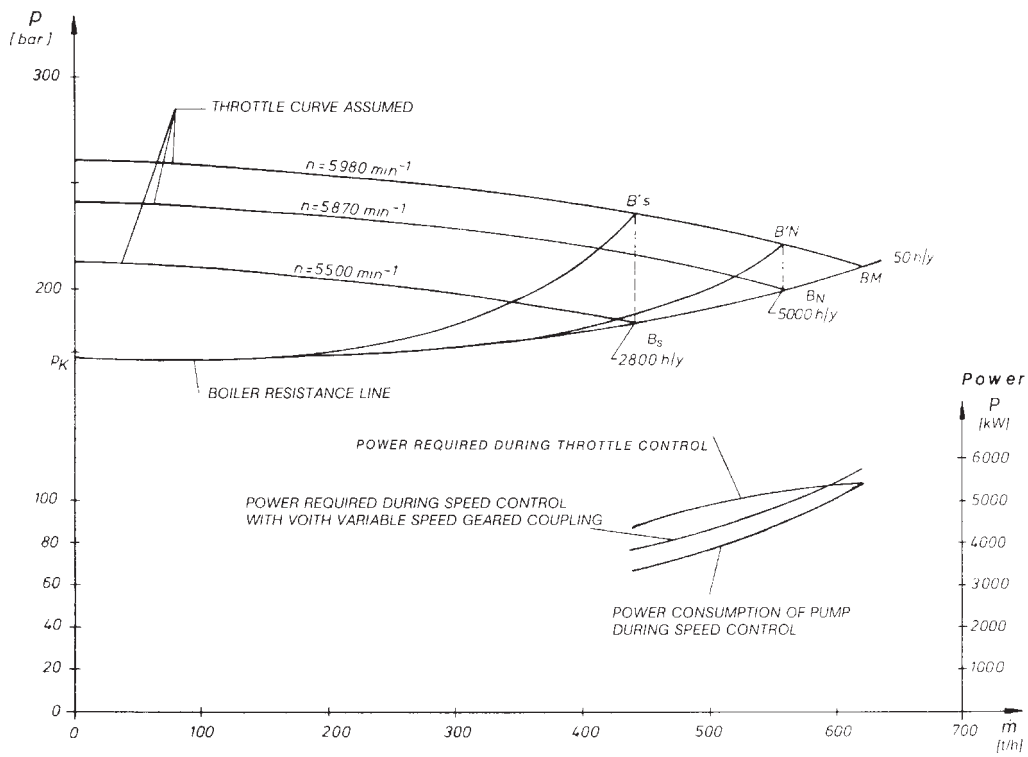


FIG. P-136 Comparison of throttle control and geared coupling control. (Source: J. M. Voith GmbH.)

(Figs. P-137 and P-138) showing the motor, feed pumps, and the coupling that combines a mechanical gear and a fluid coupling.

The theoretical energy saving is opposed by the efficiency losses of the coupling, detailed in Table P-17 as slip, and the resultant hydraulic and mechanical losses (bearing friction, etc). The power losses were calculated by means of a table computer program (see Table P-17).

We have assumed the pump efficiency for throttle regulation operation at the corresponding points to be as follows:

Maximum performance $B_M\eta = 0.76$
Normal performance $B_N\eta = 0.74$
Minimum performance $B_S\eta = 0.66$

$$P = \frac{V \cdot p}{\eta} = \frac{m \cdot p}{\rho \cdot \eta}$$

TABLE P-17

Operation Point	s (%)	P_{vs} (kW)	P_{vm} (kW)
Max. point B_M	2.7	148	149
Norm. point B_N	4.5	208	155
Min. point B_S	10.5	382	172

P_{vs} = hydraulic losses; P_{vm} = mechanical losses in the coupling; s = slip.

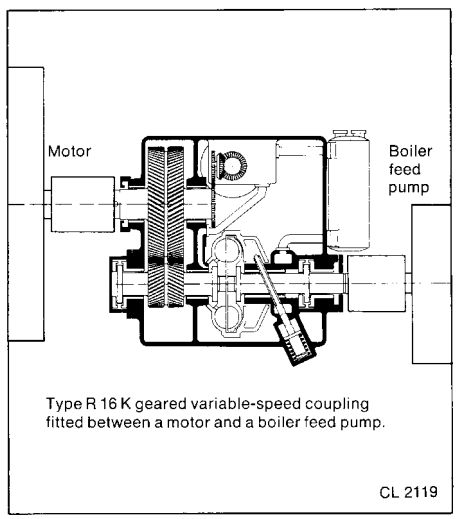


FIG. P-137 Arrangement sketch: geared variable-speed coupling. (Source: J. M. Voith GmbH.)

Table P-18 shows the result for the three load points: column one gives the performance with throttle regulation using the above formula, column two the performance with speed regulation taking account of the losses in the fluid coupling, and column three the difference between the two.

It is immediately obvious from these results that throttle regulation is only favorable for maximum power operation. However, this operating point is only used under certain conditions and, even then, very rarely; this is why the second point is called the normal point. When one considers the normal point and the minimum power operating point, the advantage of speed regulation using a hydrodynamic variable-speed coupling is unmistakable.

To be in a position to assess the difference between the two types of regulation from the financial point of view, we must first establish an annual load schedule. The schedule assumes that the power station block runs for 7850 hours per year with 5000 hours at normal power. This assumption again favors throttle regulation since power station blocks of this rating are used more for peak or average loads and not, as assumed here, for base loads. The results in MWh are given in Table P-19.

The 3757.15 MWh savings achieved by speed regulation using a hydrodynamic variable-speed geared coupling means that the fluid coupling pays for itself in less than a year, assuming a price per kWh of 0.03 DM.

The energy price is highly dependent on fuel costs, plant size, commercial valuations, etc., but the assumed value should represent a reasonable mean value.

TABLE P-18 Results for Three Load Points

Operating Point	Throttle Regulation (kW)	Speed Regulation (kW)	Throttle Coupling (kW)
Max. point B_M	5348	5645	–297
Norm. point B_N	5120	4780	340
Min. point B_S	4530	3790	740

TABLE P-19

Annual Op. Hours	Throttle Regulation (MWh)	Speed Regulation (MWh)	Throttle Coupling (MWh)
B_M = 50	267.40	282.25	–14.85
B_N = 5,000	25,600.00	23,900.00	1,700.00
B_S = 2,800	12,684.00	10,612.00	2,072.00
Total 7,850	38,551.40	34,794.25	3,757.15

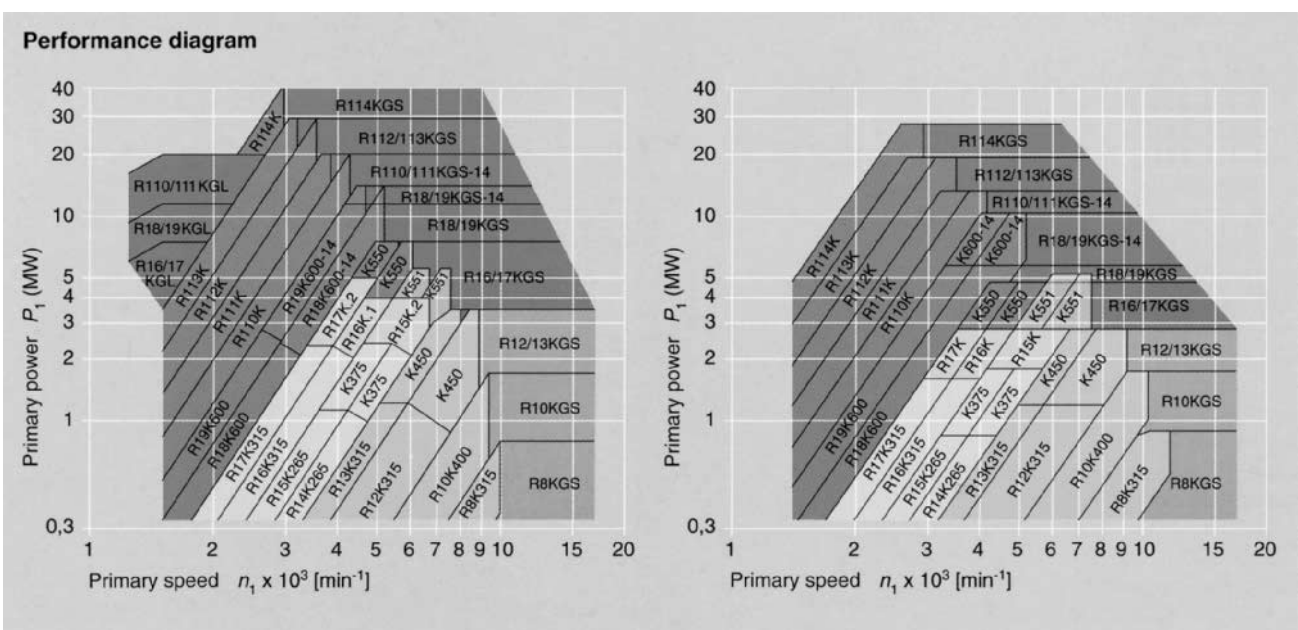


FIG. P-138 Performance diagram: geared variable-speed coupling. (Source: J. M. Voith GmbH.)

The material in this case study resulted in a recommendation in favor of the speed regulation of power station feed water pumps by hydrodynamic variable-speed couplings. The extra unit pays for itself within approximately one year. A decision in favor of speed regulation is indicated. Factors such as power station blocks designed for based loads (i.e., high-performance blocks) being used for peak and medium loads 5–10 years later support the decision.

TABLE P-20 Equipment and Design Characteristics

	Gear	Pump	Scoop Tube	Oil Flow
R8-13K	Single-stage, single helical	Gear pump	Hydraulically operated	Constant
R8-13KGS	Double-stage, double helical	Gear pump	Hydraulically operated	Constant
R . . . K	Single-stage, double helical	Pump combination	Direct mechanically operated	Variable
R . . . K551	Single-stage, single helical	Pump combination	Hydraulically operated	Variable
R . . . K265 R . . . K375	Single-stage, single helical	Gear pump	Hydraulically operated	Variable temperature controlled
R . . . K550/ 600/630	Single-stage, double helical	Pump combination	Hydraulically operated	Variable
R . . . KGS R . . . KGL	Double-stage, double helical	Pump combination	Hydraulically operated	Variable

Slip losses are small; they do not affect the efficiency of the total power station much. However, consideration was paid to the possibility of bringing the slip losses back to the working circuit, e.g., the power station circuit or the central heating circuit by means of a heat exchanger. This means that speed regulation with the hydrodynamic coupling proved itself superior to throttle regulation, and that speed regulation has been improved.

Coupling selection

The design of a geared variable-speed coupling is mainly determined by the input power and input speed of the driven machine. The performance diagram shows the power transmitted by various coupling types and sizes, depending on the input and output speeds.

Table P-20 shows varying equipment and design characteristics of the individual types of geared variable-speed couplings. This information source has couplings for individual applications. Matching coupling combinations are created by arranging the gear stage and the coupling in relation to the required power, motor speed, and speed of the driven machine. Coupling types are generally as follows (see Fig. P-139):

1. A step-up gear is located in front of the coupling. This allows adaptation of high-speed machines to the speeds of 2- or 4-pole squirrel cage motors (Type R . . . K).
2. The coupling is driven direct and adapted to the speed required by the driven machine by a step-up gear situated after the coupling. For high powers at very high speeds (Type R . . . GS).
3. A step-up gear is located both at the input and the output side. This is suitable for particularly high output speeds, e.g., with high-speed boiler feed pumps (Type R . . . KGS).
4. A step-up gear at the input side and a reduction gear at the output side make this type into a fast reacting and compact coupling for fast response times (Type R . . . KGL).

TABLE P-20 Equipment and Design Characteristics

	Gear	Pump	Scoop Tube	Oil Flow
R8-13K	Single-stage, single helical	Gear pump	Hydraulically operated	Constant
R8-13KGS	Double-stage, double helical	Gear pump	Hydraulically operated	Constant
R . . . K	Single-stage, double helical	Pump combination	Direct mechanically operated	Variable
R . . . K551	Single-stage, single helical	Pump combination	Hydraulically operated	Variable
R . . . K265 R . . . K375	Single-stage, single helical	Gear pump	Hydraulically operated	Variable temperature controlled
R . . . K550/ 600/630	Single-stage, double helical	Pump combination	Hydraulically operated	Variable
R . . . KGS R . . . KGL	Double-stage, double helical	Pump combination	Hydraulically operated	Variable

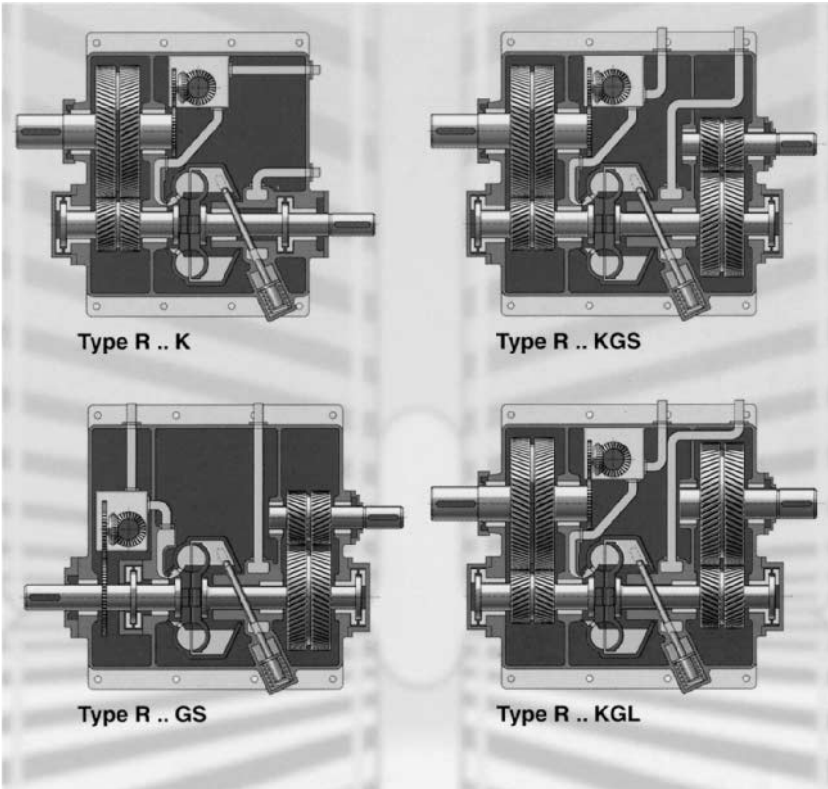
Slip losses are small; they do not affect the efficiency of the total power station much. However, consideration was paid to the possibility of bringing the slip losses back to the working circuit, e.g., the power station circuit or the central heating circuit by means of a heat exchanger. This means that speed regulation with the hydrodynamic coupling proved itself superior to throttle regulation, and that speed regulation has been improved.

Coupling selection

The design of a geared variable-speed coupling is mainly determined by the input power and input speed of the driven machine. The performance diagram shows the power transmitted by various coupling types and sizes, depending on the input and output speeds.

Table P-20 shows varying equipment and design characteristics of the individual types of geared variable-speed couplings. This information source has couplings for individual applications. Matching coupling combinations are created by arranging the gear stage and the coupling in relation to the required power, motor speed, and speed of the driven machine. Coupling types are generally as follows (see Fig. P-139):

1. A step-up gear is located in front of the coupling. This allows adaptation of high-speed machines to the speeds of 2- or 4-pole squirrel cage motors (Type R . . . K).
2. The coupling is driven direct and adapted to the speed required by the driven machine by a step-up gear situated after the coupling. For high powers at very high speeds (Type R . . . GS).
3. A step-up gear is located both at the input and the output side. This is suitable for particularly high output speeds, e.g., with high-speed boiler feed pumps (Type R . . . KGS).
4. A step-up gear at the input side and a reduction gear at the output side make this type into a fast reacting and compact coupling for fast response times (Type R . . . KGL).



Type	L	A	B	H
R 12/13 K 315	1089	315	955	410
R 12/13 K 450	1490	450	1400	450
R 14/15 K 375	1405	375	1310	380
R 14/15 K 551	1510	550	1780	420
R 15/16 K	1425	508	1542	700
R 17 K	1599	512	1555	700
R 16/17 K 550	1855	550	1780	420
R 18/19 K 600	2200	600	1940	420
R 18/19 K 600-14	2532	600	2020	790
R 16/17 KGS	1950	230	1780	420
R 18/19 KGS	2200	220	1940	420
R 18/19 KGS-14	2531	220	2020	790
R 110/111 KGS	2840	170	2110	500
R 16/17 KGL	2160		1780	420
R 18/19 KGL	2640		2090	420
R 110/111 KGL	3025		2225	500

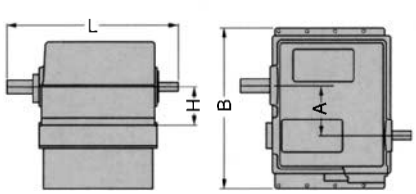


FIG. P-139 Generic hydrodynamic coupling types and dimensions in mm. (Source: J. M. Voith GmbH.)

Geared variable-speed couplings for low-speed machines

Design and function. A reduction gear is located after the coupling for applications in the drives of low-speed machines. The coupling and the gear are contained in a common housing the lower part of which is designed as an oil tank.

By the optional installation of a hydrodynamic retarder with favorable braking characteristics, a high retardation of large masses of the connected driven machine is achieved.

Speed control is carried out by changing the oil-fill of the coupling through the scoop tube that reaches into the working chamber of the coupling. The transmitted power and the speed can be controlled steplessly. (See Fig. P-140.)

Oil supply. Working and lube oil are both supplied by the integrated oil tank, however, from separate circuits. The oil supply for the motor and the driven machine may also originate from this source.

Performance diagram. This serves to identify the appropriate coupling types and sizes in relation to the input power and the required output speed.

Coupling types. Controlling low-speed machines such as coal mills, induced draught fans, and crude oil pumps demands different coupling designs.

Depending on the application and for optimum adaptation to existing space conditions, the gear stage is either designed as helical or bevel gear.

Genetic coupling types are as follows (see Fig. P-141):

1. A helical step-down gear is arranged behind the variable-speed turbocoupling. This type is suitable for applications in coal mills, crushers, low-speed pumps, and fans (Type R . . . A).
2. Geared variable-speed coupling for low-speed driven machines with high moment of inertia, e.g., coal mills. This type features a bevel gear stage behind the turbocoupling (Type R . . . B).

Torque Conversion*

A torque converter can be used where torque and speed have to be transmitted or controlled between a driving and a driven machine in a simple and economical way.

A torque converter is “open” to both sides. It absorbs torque from the driving machine and transmits it according to requirement to the driven machine. This process is governed by the “natural” characteristics of the converter, or can be controlled at will by varying the pitch of the guide blades (adjustable converter).

In both cases, the converter can be incorporated in a process control system. In this prospectus, the attempt is made with examples to concisely yet clearly describe the hydrodynamic torque converter, its characteristics, and its application possibilities.

The most important areas of application are listed in Fig. P-142. Note symbols in Fig. P-143.

When is it expedient to use a simple torque converter and when an adjustable one? This varies from case to case. The most important criteria in this decision are the characteristics of the prime mover and the load curve governed by the job to be handled.

What is regulated, what is controlled?

See Figs. P-144 through P-146.

---

*Source: J.M. Voith GmbH, Germany.

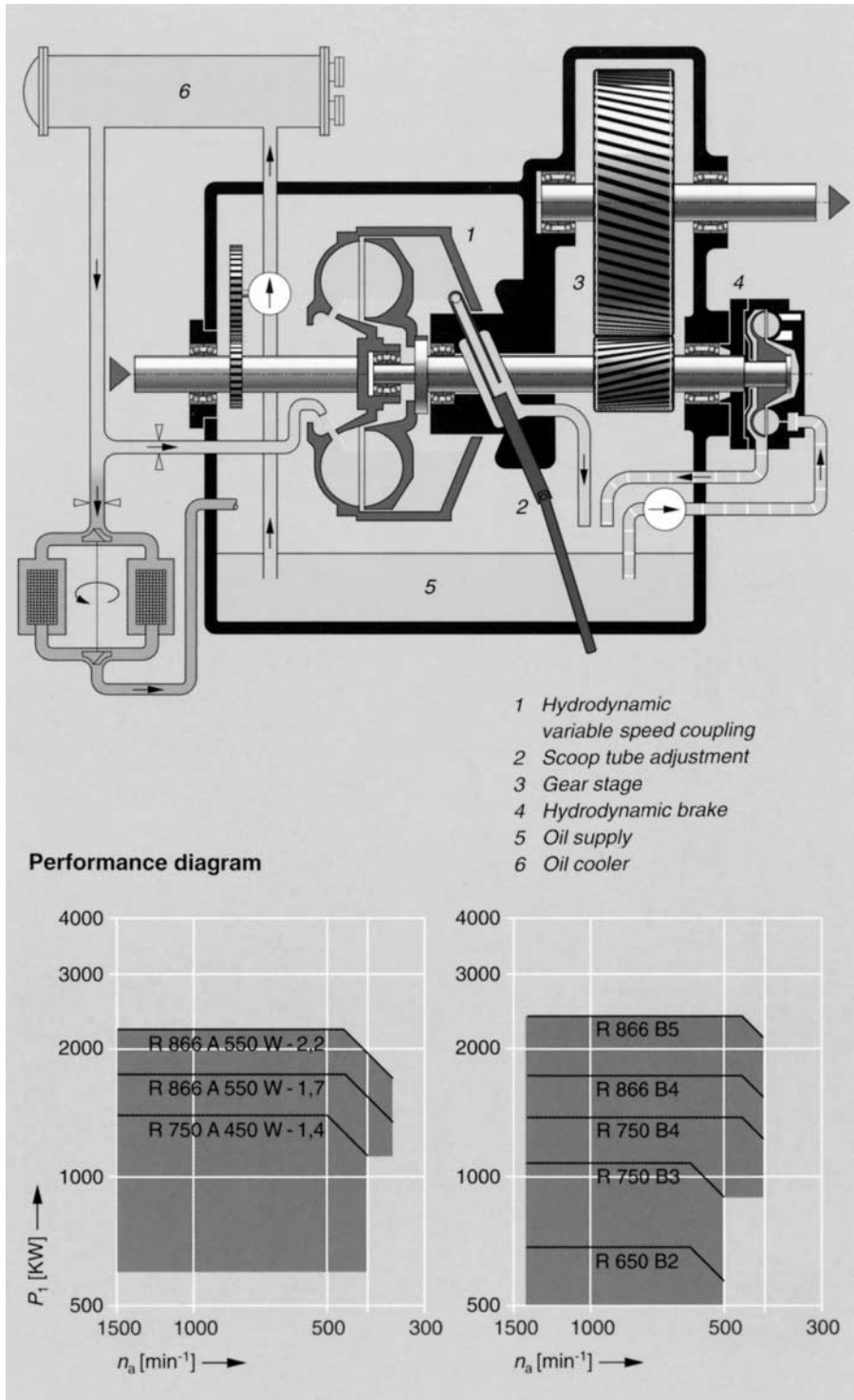


FIG. P-140 Geared variable-speed coupling for low-speed machines. (Source: J. M. Voith GmbH.)

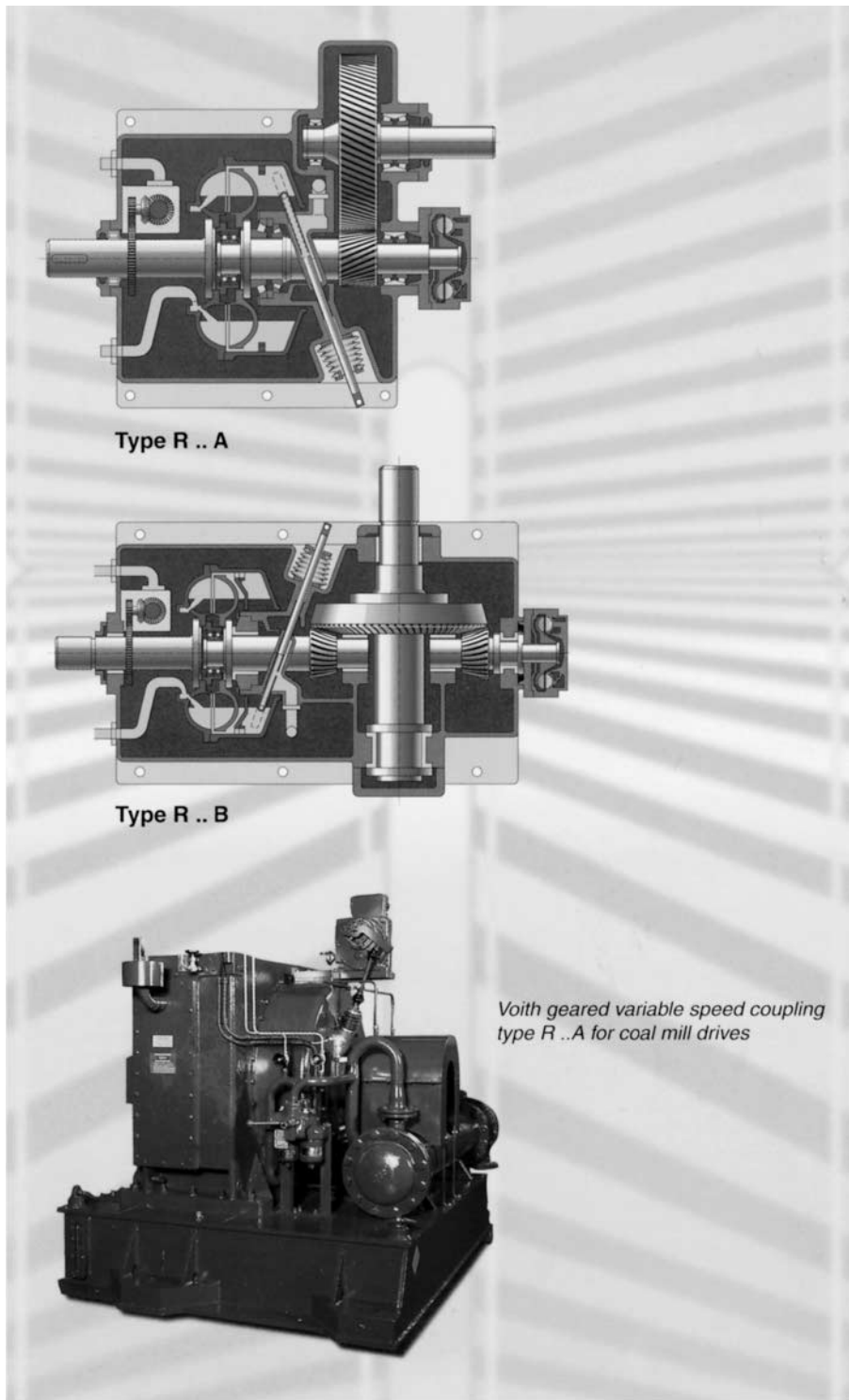


FIG. P-141 Generic model of geared variable-speed couplings for low-speed machines. (Source: J. M. Voith GmbH.)

Iron and steel industry			**Entertainment industry**		
Rolling mill drives, shape rolling mills		●	Wave generators		●
Reeling machines		●			
Wire coiling machines, roll-over machines		●	**Mining**		
Wire drawing machines		●	Coal planers	●	●
			Hauling winches		●
Wood-working industry			Rotary kilns	●	●
Veneer cutting machines		●	**Drilling**		
			Rotary drives	●	
			Lifting gear	●	
Chemical industry			Scavenging pumps	●	
Centrifuges, peeling centrifuges		●			
Extruders		●	**Construction**		
Fans, compressors		●	Scrapers, shovels	●	●
Mixers, agitators		●	Bucket wheel,	●	●
Mills, ball mills		●	chain bucket excavators	●	●
Reciprocating and process pumps		●	Concrete mixers, concrete casting units	●	●
Food industry			**Power generation**		
Centrifuges,		●	Reactor pumps		●
Drum driers,		●	Gas turbine starter sets	●	●
Diffusion towers		●	Pump storage plants		●
Sugar cane conveyors	●	●			
Mincing machines	●	●			
			Mechanical handling		
			Chain conveyors		●
Mechanical engineering			Plate conveyors		●
			Belt conveyors		●
Cutting off machines		●			
Mould drives		●			
Balancing drives		●	**Marine engineering**		
Screw presses		●	Winch drives		●
Test bench equipment		●			
Straightening machines		●			

FIG. P-142 Torque conversion applications. (Source: J. M. Voith GmbH.)

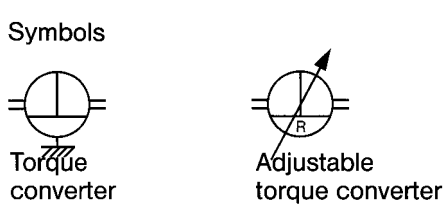
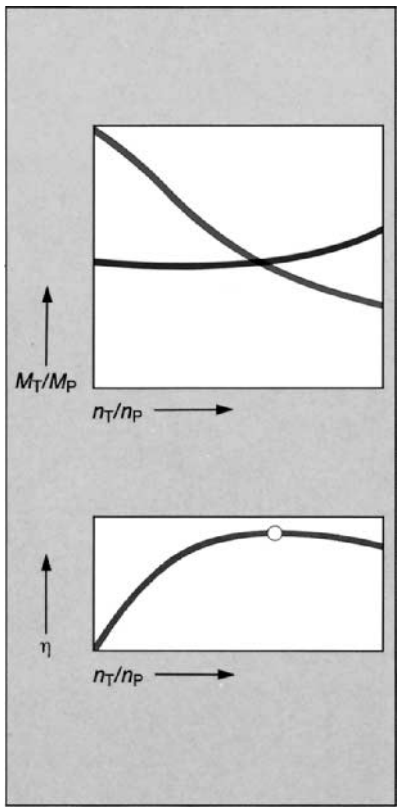
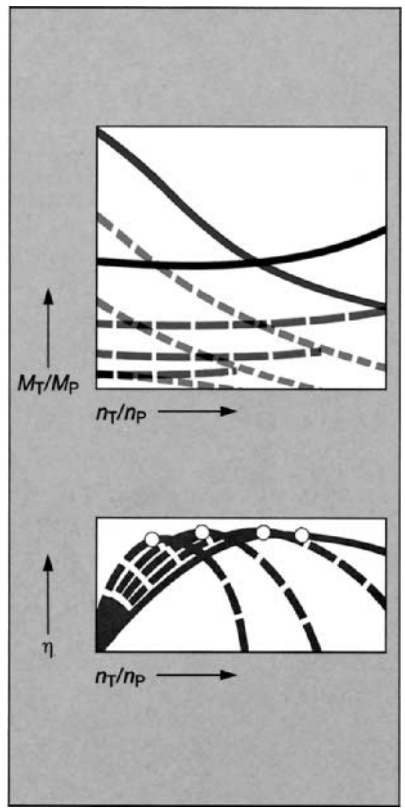


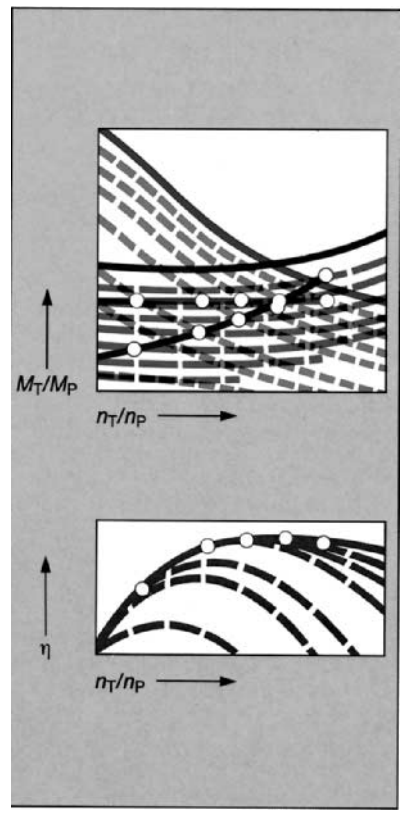
FIG. P-143 Symbols for a torque converter and an adjustable torque converter. (Source: J. M. Voith GmbH.)



144



145



146

FIG. P-144 Using a motor of constant speed and a torque converter without adjustment mechanism, jobs such as startups can be undertaken smoothly and steplessly along the "natural" characteristic curve of the converter. This is a starting process. The operating range is centered around peak efficiency. (Source: J. M. Voith GmbH.)

FIG. P-145 Using a variable-speed motor and a nonadjustable torque converter, a method of regulation is obtained that permits any desired curve to be operated corresponding to the input/output speed ratio. This case illustrates how the efficiency peak remains practically the same. (Source: J. M. Voith GmbH.)

FIG. P-146 Using a constant-speed motor and an adjustable torque converter, any operating point can be run within the characteristic range of the turbine, e.g., a constant torque can be held under fluctuating load or specific speeds in alternating succession. At first glance, an adjustable torque converter works uneconomically in specific ranges, because the efficiency decreases with progressively closed guide blades. This is deceptive. If one projects the operating points from the turbine characteristic range onto the efficiency curves, the result is a good overall efficiency. (Source: J. M. Voith GmbH.)

How the torque converter works

The pump impeller of the converter, a centrifugal pump, converts the torque M_P absorbed from the motor into kinetic energy in a fluid mass. See Fig P-147.

In the turbine wheel the fluid mass is decelerated producing the turbine torque M_T. The guide wheel (reaction member) absorbs the differential torque M_R between input and output torque that results in the multiplication of torque. The ratio and curve of torque multiplication M_T/M_P is conditioned by the form of the converter blades.

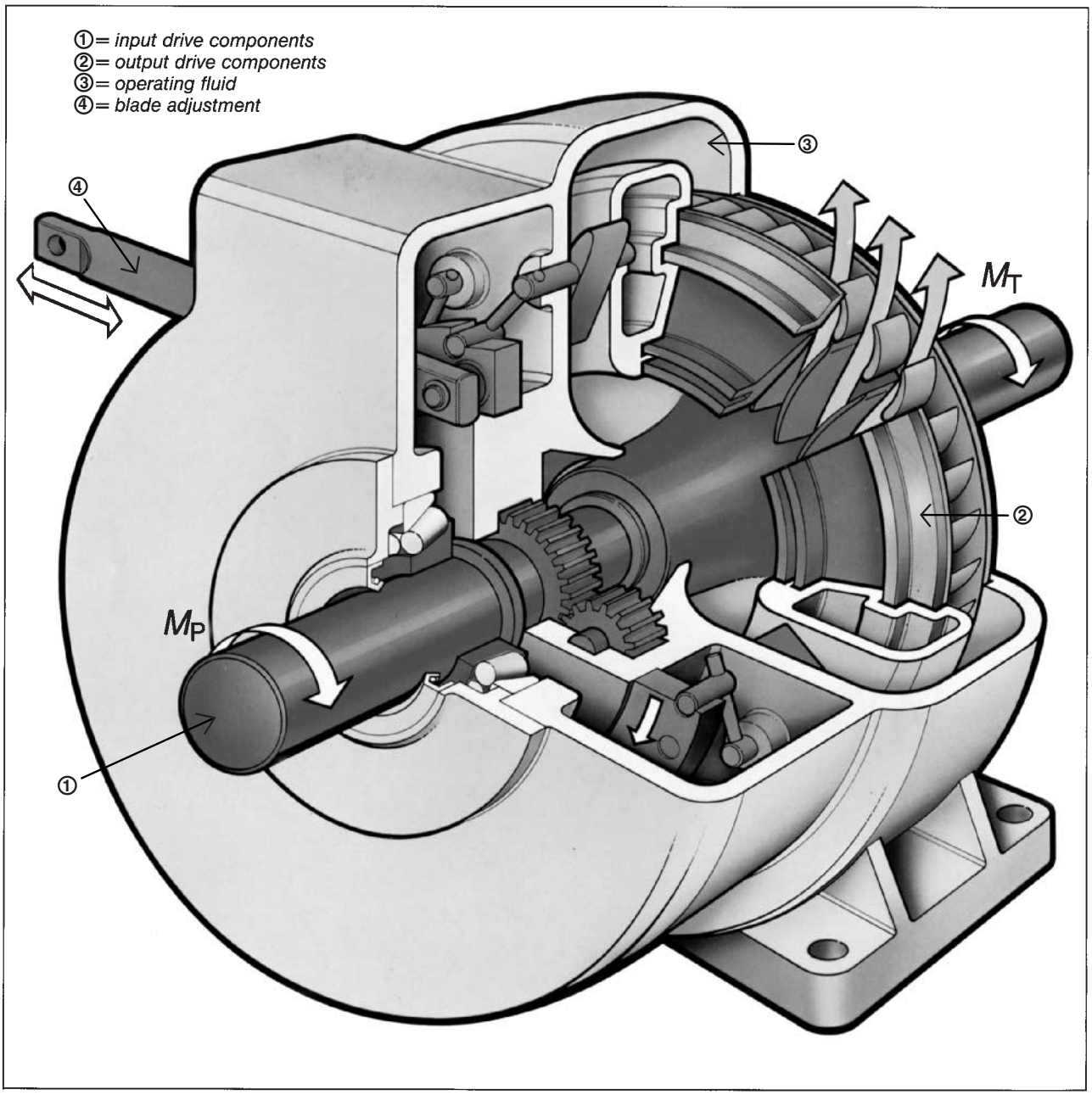


FIG. P-147 Torque converter components. (Source: J. M. Voith GmbH.)

For this reason, every converter design has its own characteristic family of curves.

The integral gear pump produces the necessary operating pressure and dissipates heat generated by directing the oil stream to the cooling unit.

Application case 1: Gas turbine starting

The gas turbine is loaded by its compressor and the mass of the generator. It cannot be started without assistance. As a starting motor either an electric motor can be

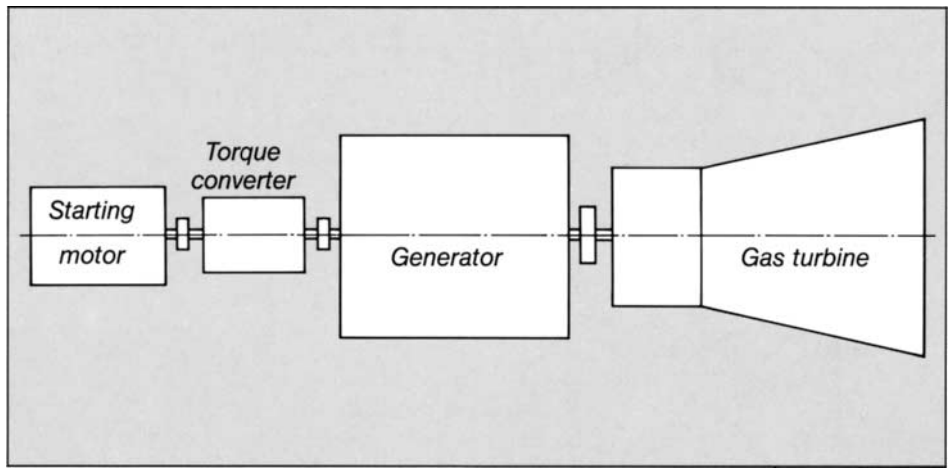


FIG. P-148 Schematic of a torque converter in a drivetrain. (Source: J. M. Voith GmbH.)

used, providing that sufficient power can be taken from the mains, or alternatively a diesel engine. See Fig. P-148.

Possible solutions. The torque converters shown in Figs. P-149 through P-152 can be employed wherever large masses have to be accelerated rapidly and smoothly, wherever high breakaway torque has to be overcome, and wherever speeds have to be automatically adapted to loads.

A torque converter, however, is not only an efficient but also an economical solution to a drive problem: low prime costs, low space requirements, and practically wear-free operation are facts which speak for this.

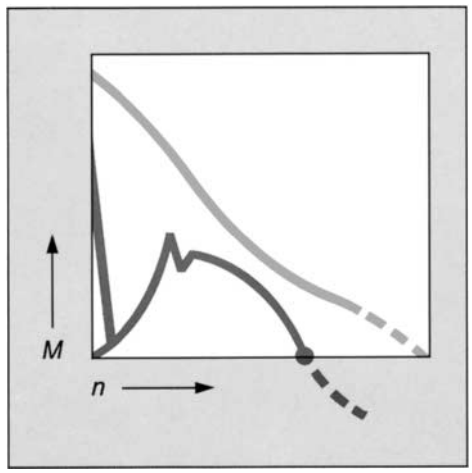
Torque converters for gas turbine starting sets are particularly adaptable. They can be supplied with a filling and draining facility of with a guide blade adjustment mechanism; in all cases, the blading can be matched to the torque loads to be handled. Previously supplied torque converters include those with ratings up to 4000 kW at speeds of 3600 1/min (60 Hz) and 3000 1/min (50 Hz).

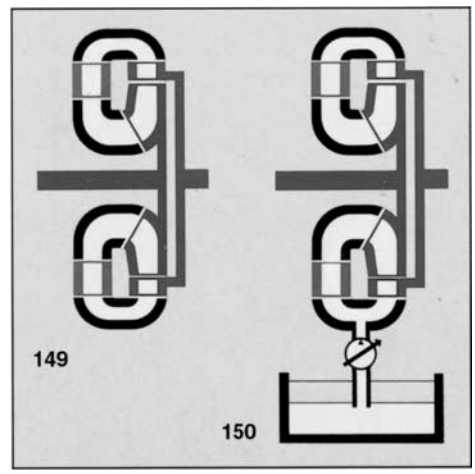
The operating fluid for the torque converter is taken from the lube oil system of the gas turbine. The oil tank simultaneously serves as a heat accumulator.

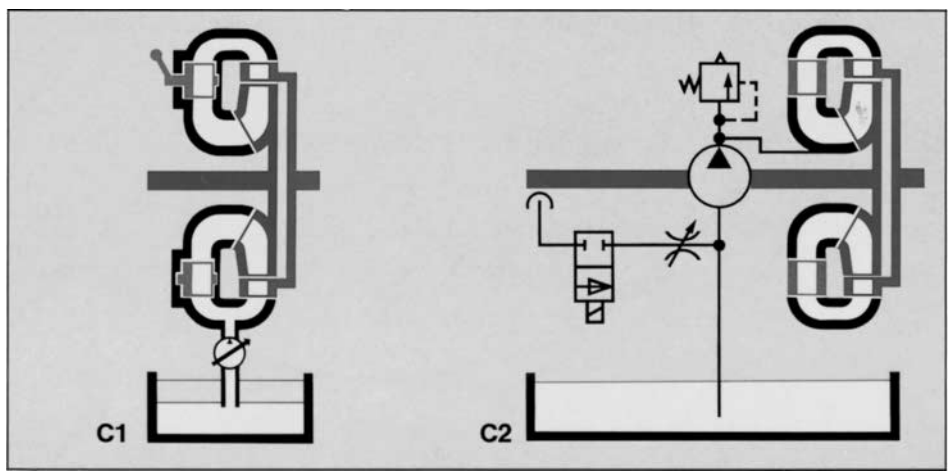
Application case 2: Variable-speed engine and fluctuating loads (Figs. P-153 through P-159)

Several diesel engines drive the drilling rods, lifting gear, and mud pumps through a chain compound under constantly fluctuating load. During drilling, speeds must be steplessly varied and shock loads absorbed. At the same time, the mud pumps press flushing and ballast material into the drill-hole at different rates and pressures to flush out the drill cuttings. The lifting gear works with high acceleration and retardation when the, often, several thousand meters long drilling rods need to be dismantled quickly. The automotive governed diesel engines, with their available speed and torque capacity, are incapable of fulfilling these tasks satisfactorily.

Possible solutions. Both types of converters are also suitable for other similarly loaded machines, e.g., heavy earth-moving machines, light shunting locomotives,







151

FIG. P-149 Torque converter with constant filling. The gas turbine is brought out of standstill at a high breakaway torque and is accelerated under increasing compression resistance to firing speed. After this, hot combustion gases in the turbine support the compressor so that with increasing speed the torque requirement decreases up to the self-sustaining speed of the turbine. When this speed has been reached, the starting equipment is disengaged by means of an SSS-clutch. The gas turbine oil tank serves as an accumulator for heat generated in the torque converter. (Source: J. M. Voith GmbH.)

FIG. P-150 Torque converter with filling and draining control. With the torque converter drained, the starting motor is accelerated up to rated speed. After this, the converter is filled and the gas turbine started as under Fig. P-149. To disengage the starting equipment, the torque converter is simply drained. A freewheeling device is dispensed with. (Source: J. M. Voith GmbH.)

FIG. P-151 Torque converter with filling and draining control and guide blade adjustment (C1). A torque converter equipped in this way fulfills the functions described under Figs. P-149 and P-150. Additionally, the gas turbine can be held constant at any required speed. A similar influence on speed is obtained by varying the circulating oil flow, e.g., by accurately dosed ventilation of the oil circuit (C2). (Source: J. M. Voith GmbH.)

scrapers, loader plants, bucket dredgers, cranes, and winches. The converter protects the plant from overload and overdrive, matches itself steplessly and automatically to arising resistance, and dampens shock and vibration between driving and driven machines. The use of a torque converter substantially increases the length of life reliability of a plant.

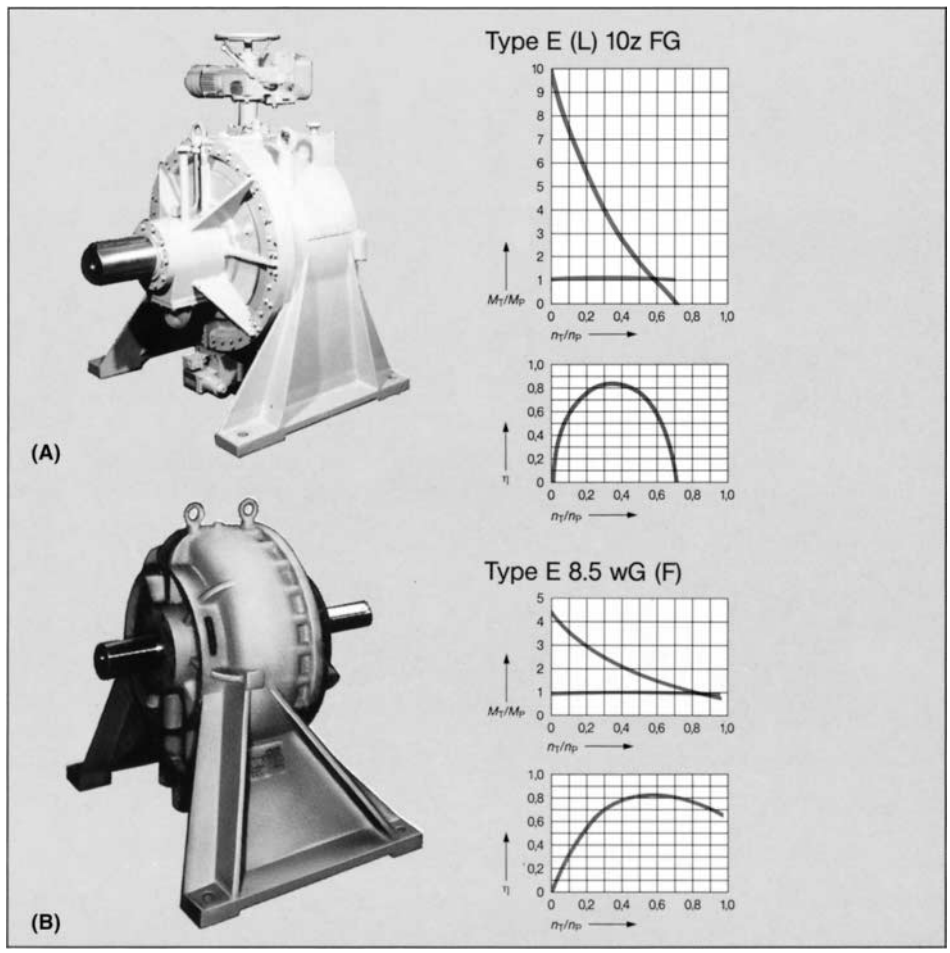


FIG. P-152 Torque converter models available. (*A*) Type E (L) 10z FG. Power rating = 1416 kW at 2950 1/min (gas turbine GTMS 9001 E, 50 Hz). Effective operating range 80 percent. Peak efficiency 34 percent. Optional: filling and draining facility or guide blade adjustment. (*B*) Type E 8.5 wG (F). Power rating = 560 kW at 210 1/min (gas turbine TG 20/B2, 50 Hz). Special feature: 4-fold torque multiplication. Optional: filling and draining facility. (Source: J. M. Voith GmbH.)

Application case 3: Speed and torque regulation (Figs. P-160 through P-165)

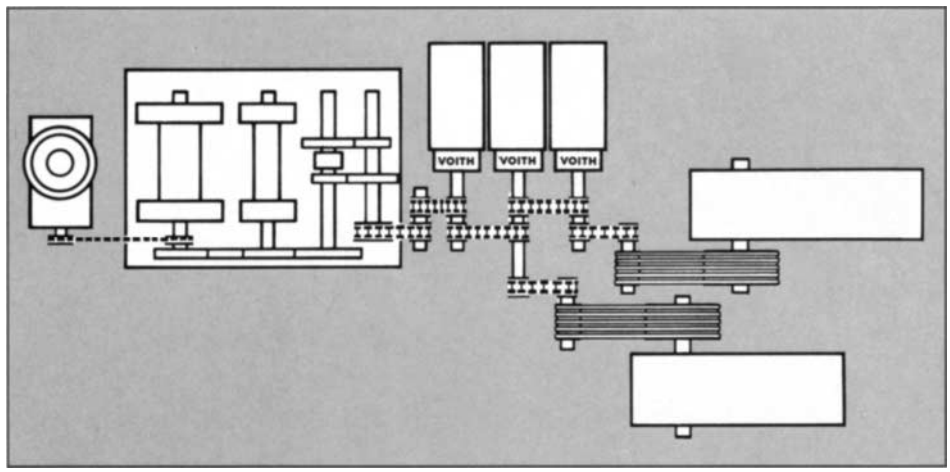
Reciprocating machines work at both low and high speed, mostly against a constant pressure, and therefore require a constant torque. Rates of delivery should be precisely and steplessly adjustable. High efficiency is desired throughout the entire operating range. With explosive media, flameproof protection is to be provided.

Possible solution. An electric motor with constant speed and an adjustable torque converter (torque converter with adjustable guide blade mechanism) form an ideal variable drive unit for reciprocating pumps. It is easy to regulate and therefore suitable for any process control; the efficiency is good when compared with that of other types of regulation. Figure P-142 demonstrates that the adjustable torque converter is used in reciprocating pump drives.

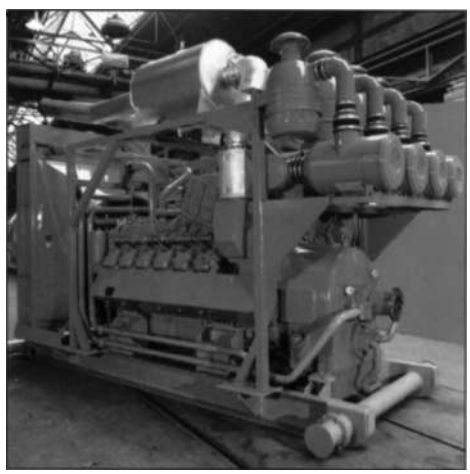
Torque converter (compact design Vorecon® type RWE)

In its most compact form, the RWE-type Vorecon comprises the main components of a hydrodynamic adjustable torque converter, a fixed planetary gear, and a revolving planetary gear in a common housing. (See Figs. P-166 and P-167.)

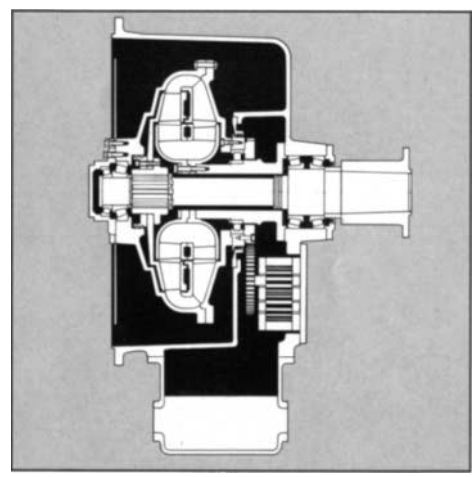
153



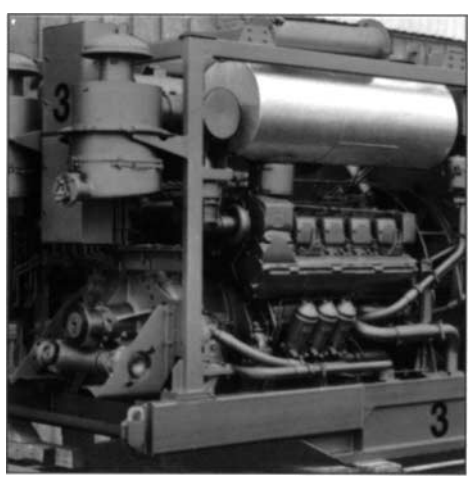
154



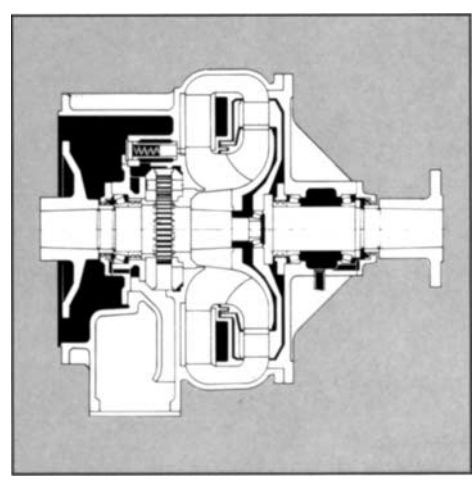
155



156



157



FIGS. P-153, P-154, P-155, P-156, P-157 A torque converter is mounted on the output end of each diesel engine. A torque converter compensates automatically fluctuations in torque, isolates vibrations, dampens shock loads, and can be overloaded. These qualities are of particular significance in multiple drive plants with widely differing loads. The waste heat generated in the converter is dissipated in an oil cooler. (Source: J. M. Voith GmbH.)

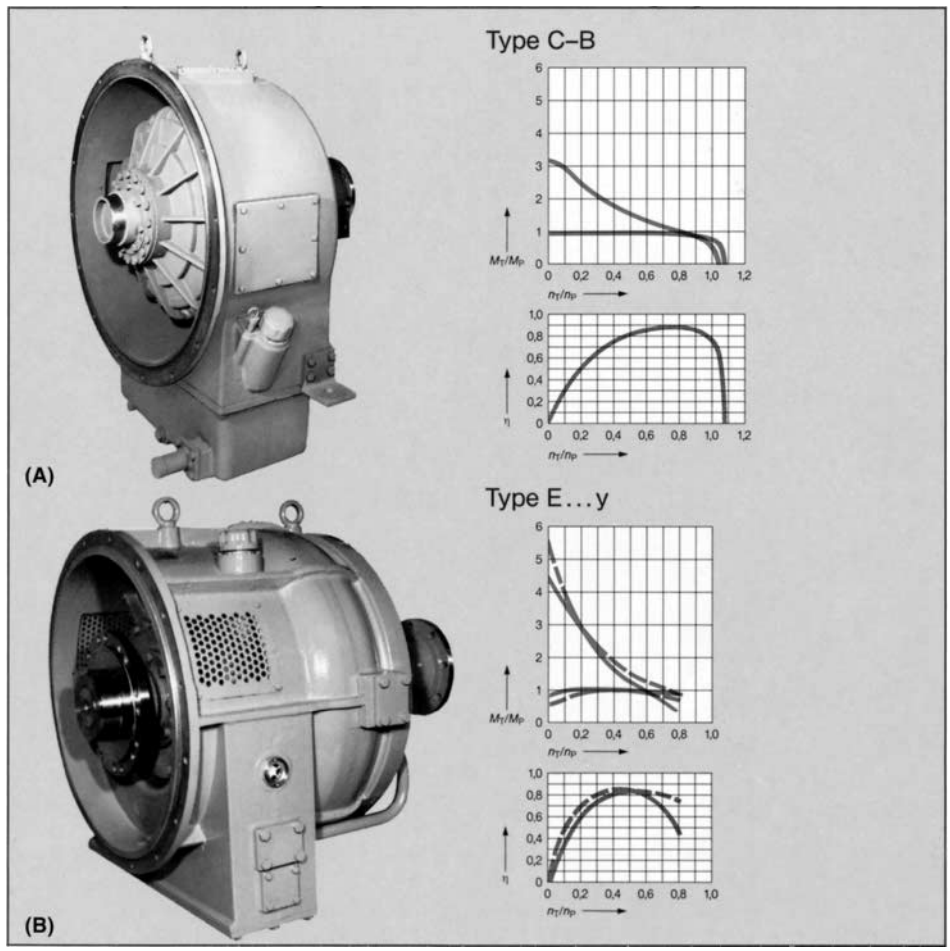


FIG. P-158 (*A*) Torque converter type C–B. Power range 50–1000 kW. Special features: Synchronous converter (no torque absorption at full output speed); therefore, no overload protection required. Effective operating range = 100 percent. Peak efficiency at 80 percent. Can be flange-mounted on a diesel engine; own lube oil system (oil sump and mechanical oil pump). Cooling via engine cooling unit or oil/air cooler (supplied as optional extra). (*B*) Torque converter type E. . .y. Power range 50–1000 kW. Special features: High starting torque multiplication, no engine lug-down. Effective operating range = 80 percent. Peak efficiency at 55 percent. Can be flange-mounted on a diesel engine; own lube oil system (oil sump and mechanical oil pump). Cooling via engine cooling unit or oil/air cooler (supplied as optional extra). (Source: J. M. Voith GmbH.)

Function. The Vorecon works on the principle of a power split.

The adjustable torque converter is used to "take off" a small part of the power from the main shaft and feed it to the revolving planetary gear via the fixed planetary gear.

By adjusting the position of the guide vanes of the torque converter, the output speed is varied.

The major portion of the power is transmitted purely mechanically and thus with hardly any loss via the main shaft and the planetary drive.

This produces a very high degree of efficiency for the entire unit over the speed range.

For type RWE the speed range is approximately 60 to 100 percent.

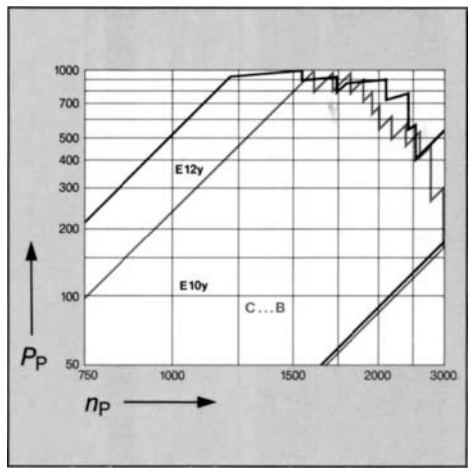


FIG. P-159 Power versus efficiency characteristics of C–B and E . . . y converters. (Source. J. M. Voith GmbH.)

Because of its short, coaxial form, the RWE is extremely space-saving and has very low capital cost.

Applications. The principal applications for this version are in the drives of boiler feed pumps, compressors, and other high-speed machinery with a limited speed range.

Expanded speed range application (Vorecon® type RWS)

In addition to the torque converter, fixed gear, and revolving gear unit, the RWS version has a hydrodynamic adjustable brake (retarder). (See Figs. P-168 and P-169.)

Function. Operation of the Vorecon type RWS can in principle be split into two operating ranges.

In operation range 1 the retarder is filled with oil and the torque converter is drained. Torque support of the superimposing system is affected by the retarder that has variable guide vanes. Adjusting the guide vanes alters the output speed.

On reaching the changeover point, the retarder is drained and the torque converter "takes over" operating range 2.

The changeover point can be fixed between 70 and 85 percent of the nominal speed by appropriate selection of gearing ratio, during the design phase.

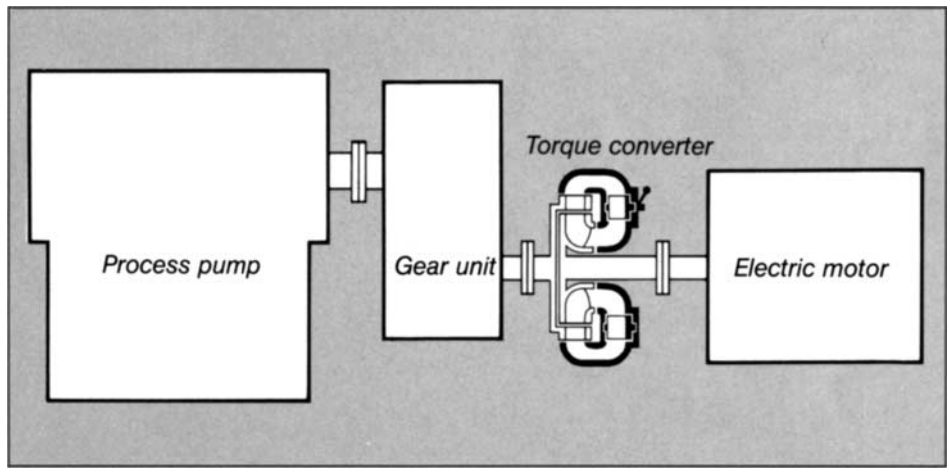
This means that the optimum efficiency of the drive can be adjusted to the load schedule of the plant.

The overall speed-regulating range of the Vorecon type RWS is approximately 45 to 100 percent.

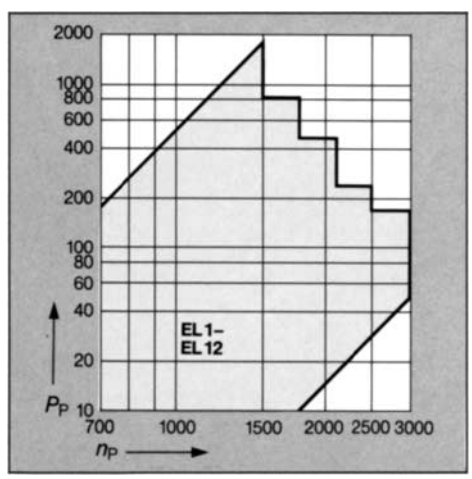
It is this expanded speed range, compared with the RWE, that characterizes the application areas of this version.

Relieved startup application (Vorecon® type RW)

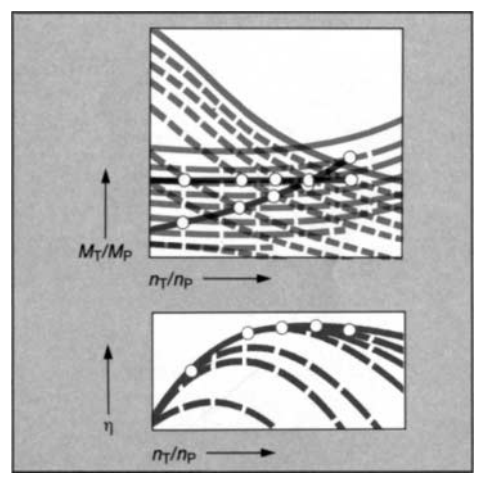
In this type of RW multistage variable-speed drive, a hydrodynamic variable-speed coupling with scoop tube and lockup clutch is incorporated in front of the torque converter.



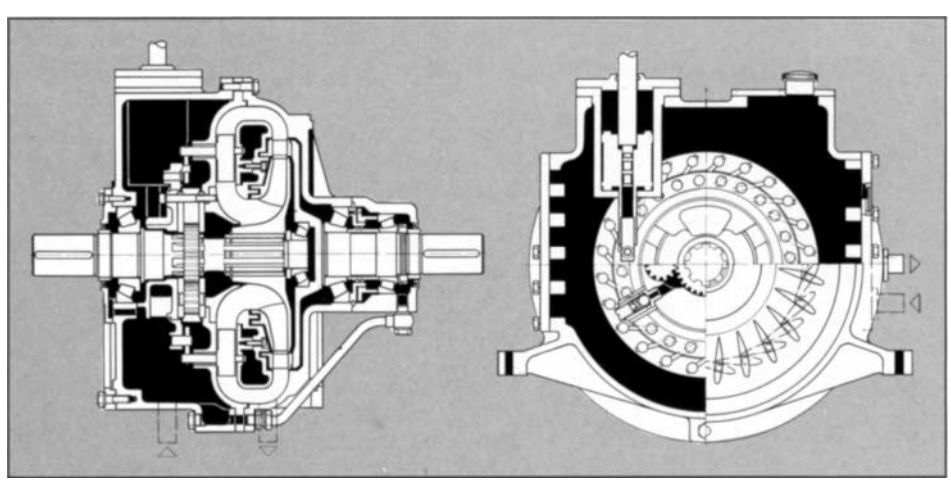
160



161



162

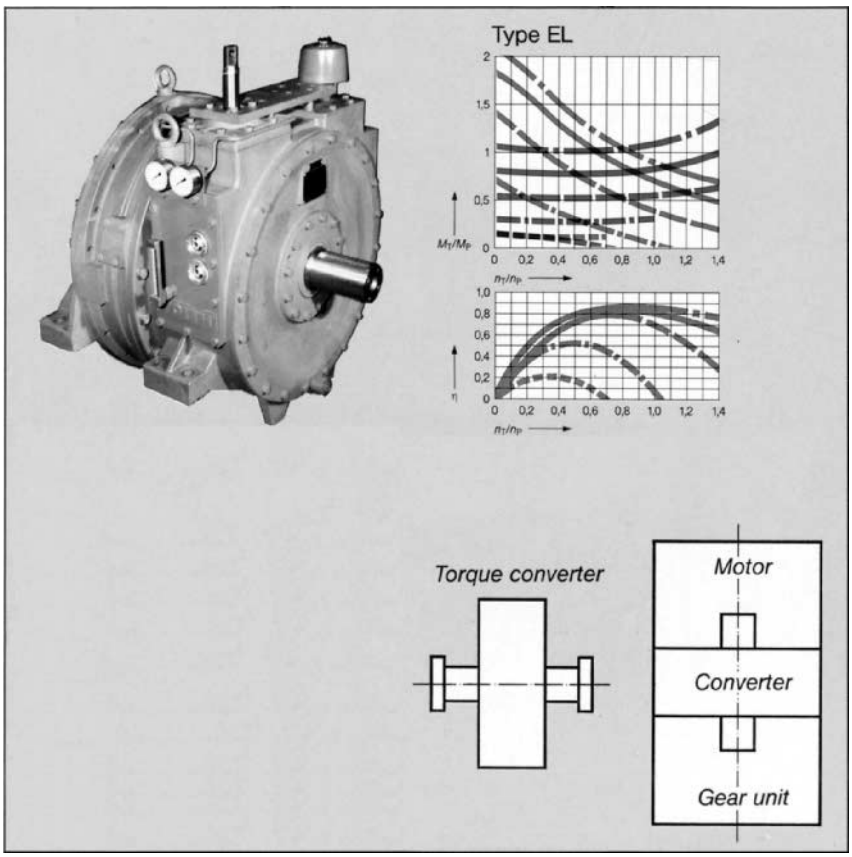


163

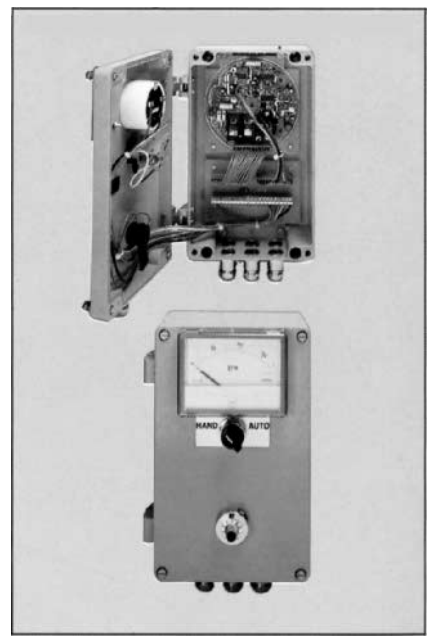
FIG. P-160 Schematic of a torque converter in a process pump train. (Source: J. M. Voith GmbH.)

FIGS. P-161, P-162 Characteristics of torque converter used in reciprocating machinery trains. (Source: J. M. Voith GmbH.)

FIG. P-163 Torque converter schematic (see also Fig. P-164). (Source: J. M. Voith GmbH.)



164



165

FIG. P-164 Adjustable torque converter, type EL. Power range 15–1000 kW. Special features: 2.5-fold torque multiplication; effective operating range = 140 percent; peak efficency with fully opened blades = 85 percent. Guide blade adjustment: manual actuation via push-rod or handwheel, push-button operated via servomotor drive, controlled and regulated electronically via governor or pneumatically via servocylinder. Versions: Input and output shafts horizontal or vertical shaft ends with key. (Source. J. M. Voith GmbH.)

FIG. P-165 Governors for adjustable torque converters link up guide blade adjustment and plant control. They have indicators for ideal and actual values and are supplied optionally as electronic variable-speed governors, or as constant-speed or torque governors. (Source. J. M. Voith GmbH.)

The features of the variable-speed coupling enable type RW to be used if a particularly wide range is required and/or if relieved startup of the drive motor is needed. (See Figs. P-170 and P-171.)

Function. Its function is also split into two operating ranges.

In operating range 1 the power flow passes through the variable-speed coupling and the revolving planetary gear. The speed is adjusted exclusively by changing the quantity of oil in the runner parts of the coupling, using the adjustable scoop tube. This covers the regulating from 10 to 80 percent of maximum speed.

In operating range 2 the lockup clutch is brought into operation and the converter is filled with oil. The converter pump wheel rotates at the same speed as the drive motor. Adjustment of the guide blades produces an increase in output speed from 80 to 100 percent.

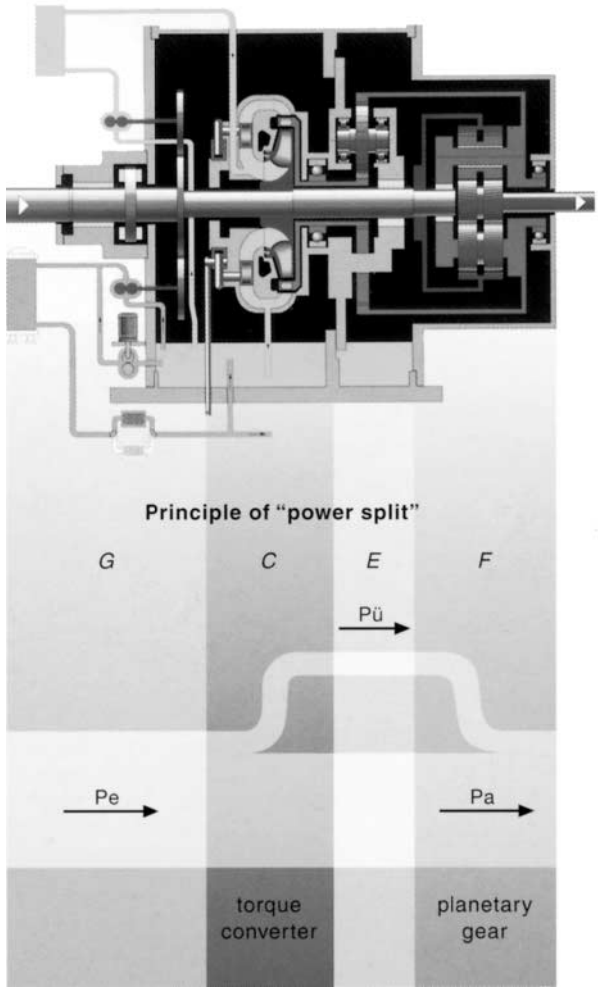


FIG. P-166 Design scheme of Vorecon type RWE. *C*, torque converter; *E*, fixed gear; *F*, revolving gear; *G*, oil supply; P_e, input power; $P_{ü}$, superimposed power; P_a, output power. (Source. J. M. Voith GmbH.)

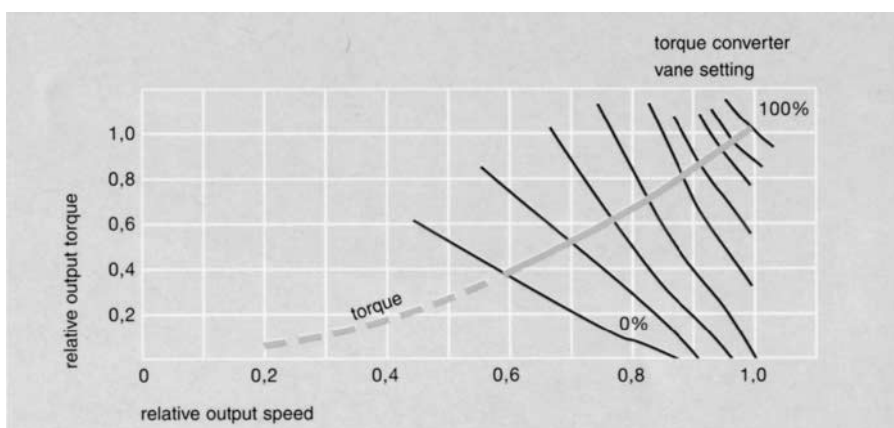


FIG. P-167 Performance characteristics (type RWE). (Source. J. M. Voith GmbH.)

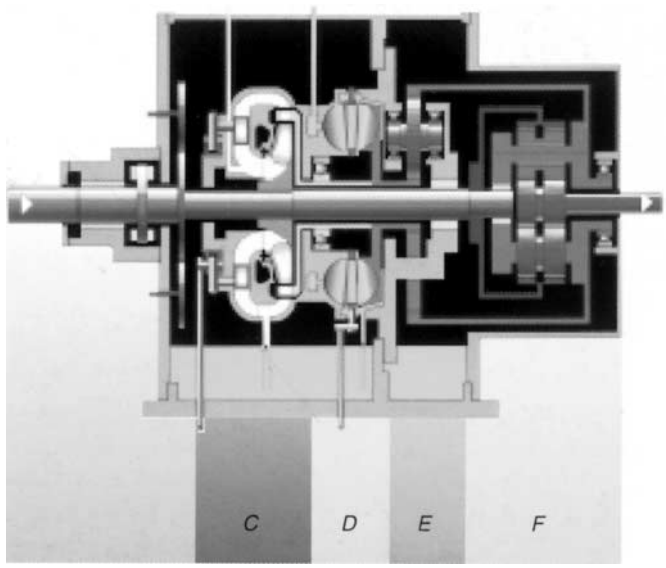


FIG. P-168 Design scheme of Vorecon type RWS. *C*, torque converter; *D*, hydrodynamic adjustable brake (retarder); *E*, fixed planetary gear; *F*, revolving planetary gear. Oil feed and control not illustrated. (Source. J. M. Voith GmbH.)

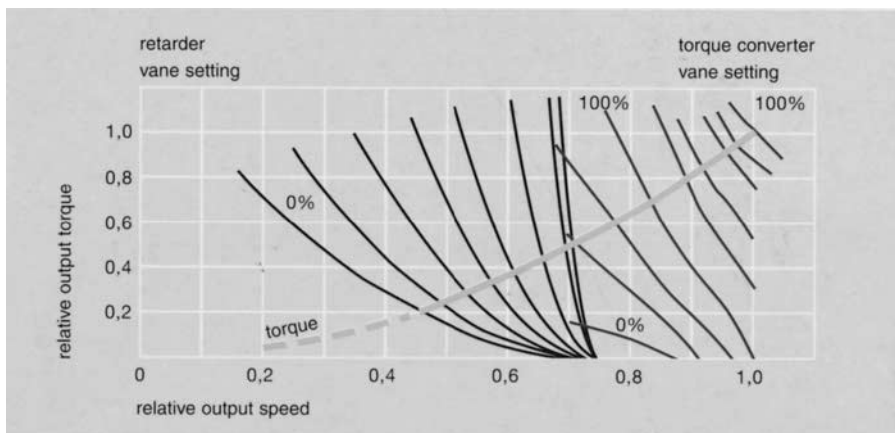


FIG. P-169 Performance characteristics (type RWS). (Source. J. M. Voith GmbH.)

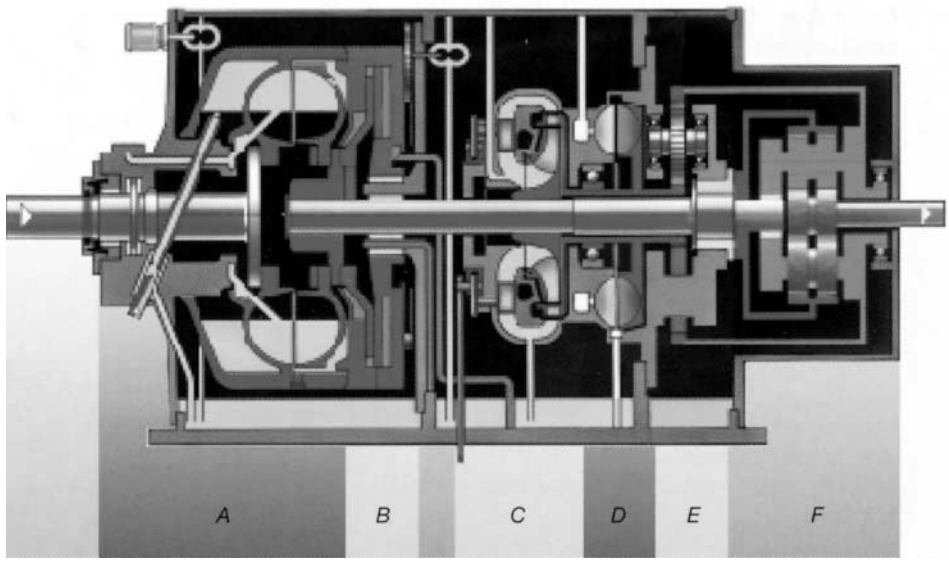
Because of its overall speed range from 10 to 100 percent, the RW multistage variable-speed drive is used particularly for drives of large fans and pumps, e.g., 100 percent boiler feed pumps.

Summary: Multistage variable-speed drives

The combination of hydrodynamic drive elements with a planetary gear and the principle of superimposed power transmission has helped improve the level of efficiency of variable-speed mechanical drive systems in areas not within the scope of other drive solutions. (See Figs. P-172 through P-174.)

The principle of power splitting makes it possible to achieve efficiencies of over 95 percent over wide ranges of powers from 1000 to 50,000 kW and speeds from 100 to 20,000 rpm.

(a)



(b)

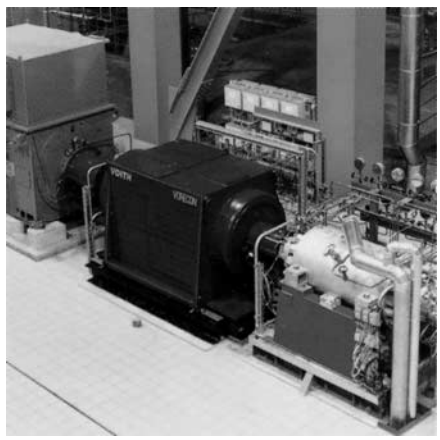


FIG. P-170 (*a*) Design scheme of Vorecon type RW. *A*, variable-speed coupling with scoop tube; *B*, lockup clutch; *C*, torque converter; *D*, hydrodynamic brake; *E*, fixed planetary gear; *F*, revolving planetary gear. (*b*) Vorecon multistage variable-speed drive type RW 14-12 F7 driving the boiler feed pumps at Mannheim power station (Germany). Input speed n_e = 1490 rpm; power requirement P_a = 8500 kW; output speed n_a = 5000 rpm. (Source. J. M. Voith GmbH.)

Observations indicate the following:

- Reliability: typical figures of 99.93 percent
- Availability including overhaul times: a typical figure is 99.83 percent
- Maintenance at intervals (TBOs) of up to 80,000 hours
- Precise adjustability and fast response times
- Compact design
- Vibration damping
- Can also be used in environments that are exposed to risk of explosion and that are dusty, at extreme temperatures

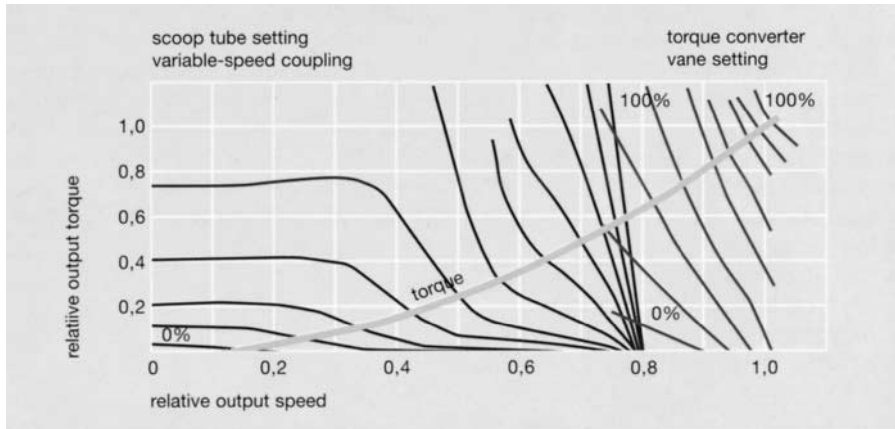


FIG. P-171 Performance characteristics (see Fig. P-170*b*). (Source: J. M. Voith GmbH.)

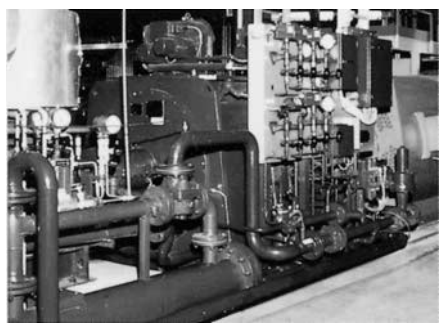


FIG. P-172 Vorecon multistage variable-speed drive type RWS 10-9 E2 driving the boiler feed pumps at Kajaani power station, Finland. Input speed n_e = 1490 rpm; power requirement P_a = 1430 kW; output speed n_a = 6000 rpm. (Source: J. M. Voith GmbH.)

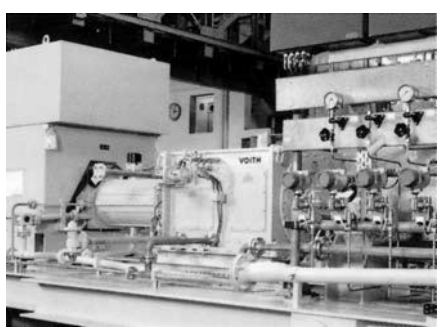


FIG. P-173 Vorecon type RWE 5 F3 driving a compressor in a Swedish refinery. Input speed n_e = 2980 rpm; power requirement P_a = 1530 kW; output speed n_a = 15,030 rpm. (Source: J. M. Voith GmbH.)

As previously noted, the multistage variable-speed drive is used in power stations, steel plants, and chemical and petrochemical industries. Specific applications include:

- Boiler and reactor feed pumps
- Coolant and condensate pumps

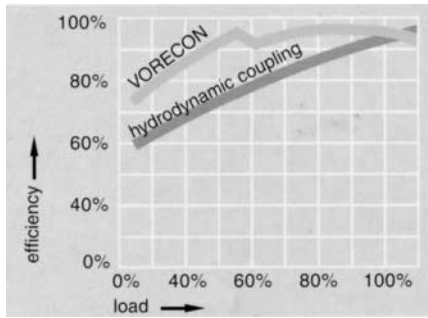


FIG. P-174 Efficiency versus load coupling characteristics (see Figs. P-172 and P-173). (Source: J. M. Voith GmbH.)

- Pipeline and loading pumps
- Piston pumps and compressors
- Boiler fans and dust extraction fans
- Centrifugal compressors
- Beater wheel mills
- Process compressors

A further application of the Vorecon, combined with an electric motor, is the replacement of gas or steam turbines when modernizing existing plants.

Torque-Limiting Safety Coupling

Features

- Adjustable release torque
- Selected release torque remains constant
- Precise point of release
- Minimum downtime
- Backlash-free power transmission
- Compact, low-weight design
- Low moment of inertia
- Minimal maintenance requirements

Design (see Figs. P-175 and P-176)

The Safeset® coupling consists of a twin-walled pressure sleeve. This can be pressurized up to 1000 bar using oil under high pressure, while a shear valve (shear tube) ensures that the system is completely sealed. Larger couplings have more than one shear tube.

A shear ring is fixed to the shaft. If any overload occurs, this shear ring breaks off the top of the shear valve.

The friction load surfaces are specially treated to prevent any wear when the coupling releases. To ensure that friction surfaces do not come into contact with

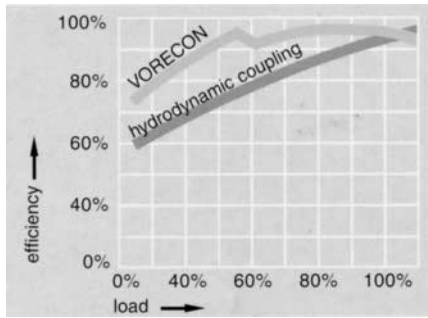


FIG. P-174 Efficiency versus load coupling characteristics (see Figs. P-172 and P-173). (Source: J. M. Voith GmbH.)

- Pipeline and loading pumps
- Piston pumps and compressors
- Boiler fans and dust extraction fans
- Centrifugal compressors
- Beater wheel mills
- Process compressors

A further application of the Vorecon, combined with an electric motor, is the replacement of gas or steam turbines when modernizing existing plants.

Torque-Limiting Safety Coupling

Features

- Adjustable release torque
- Selected release torque remains constant
- Precise point of release
- Minimum downtime
- Backlash-free power transmission
- Compact, low-weight design
- Low moment of inertia
- Minimal maintenance requirements

Design (see Figs. P-175 and P-176)

The Safeset® coupling consists of a twin-walled pressure sleeve. This can be pressurized up to 1000 bar using oil under high pressure, while a shear valve (shear tube) ensures that the system is completely sealed. Larger couplings have more than one shear tube.

A shear ring is fixed to the shaft. If any overload occurs, this shear ring breaks off the top of the shear valve.

The friction load surfaces are specially treated to prevent any wear when the coupling releases. To ensure that friction surfaces do not come into contact with

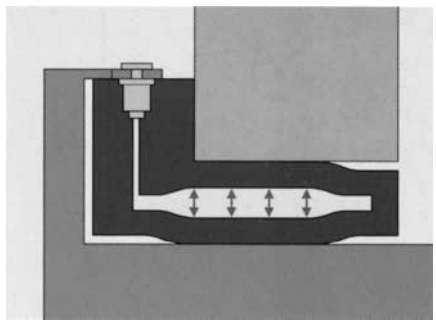


FIG. P-175 A hollow steel sleeve is expanded by oil under pressure; this produces a friction connection between a shaft and a hub. At overload the coupling instantaneously releases the oil pressure and interrupts the drive. (Source: J. M. Voith GmbH.)

each other after release, antifriction bearings are fitted between the components that rotate relative to one another. A plain bearing is sufficient for couplings operated at low speeds.

The area around the pressure surfaces and bearing is filled with a special oil. This oil supports constant friction coefficient and, in turn, precise release torque.

In contrast to shear element couplings, the release torque of the Safeset® coupling is not influenced by material fatigue, i.e., S-N data. The release torque always remains constant. (See Fig. P-177.)

Operation

To start operation of the coupling, connect a Safeset® pump to the oil charge port (see Fig. P-178) and release the shear valve slightly to allow oil to be pumped into the pressurized sleeve. The operator can select the required oil pressure using a calibration diagram that is provided for each coupling. (See Fig. P-179.)

The selected hydraulic pressure generates a defined friction load between the pressure sleeve and shaft. This pressure determines the maximum torque that can be transmitted (slip torque).

If the operating torque exceeds the selected slip torque, the shaft rotates within the pressure sleeve. The shear ring is fixed to the shaft, so it also rotates and breaks off the top of the shear tube. This causes an instantaneous drop in oil pressure and releases the friction connection. (See Fig. P-180.)

The Safeset® coupling can be returned to operation after a minimum of delay by replacing the tube and reapplying the pressure.

Since the release process does not cause any wear, no maintenance is required apart from a regular oil change.

Permitted temperature range is –20 to +60°C. Temperatures exceeding this range are possible with special measurements.

Operating procedure after overload

Follow Figs. P-181 through P-186.

Coupling types

Various design types and sizes are available to suit specific applications. See Figs. P-187 through P-191.

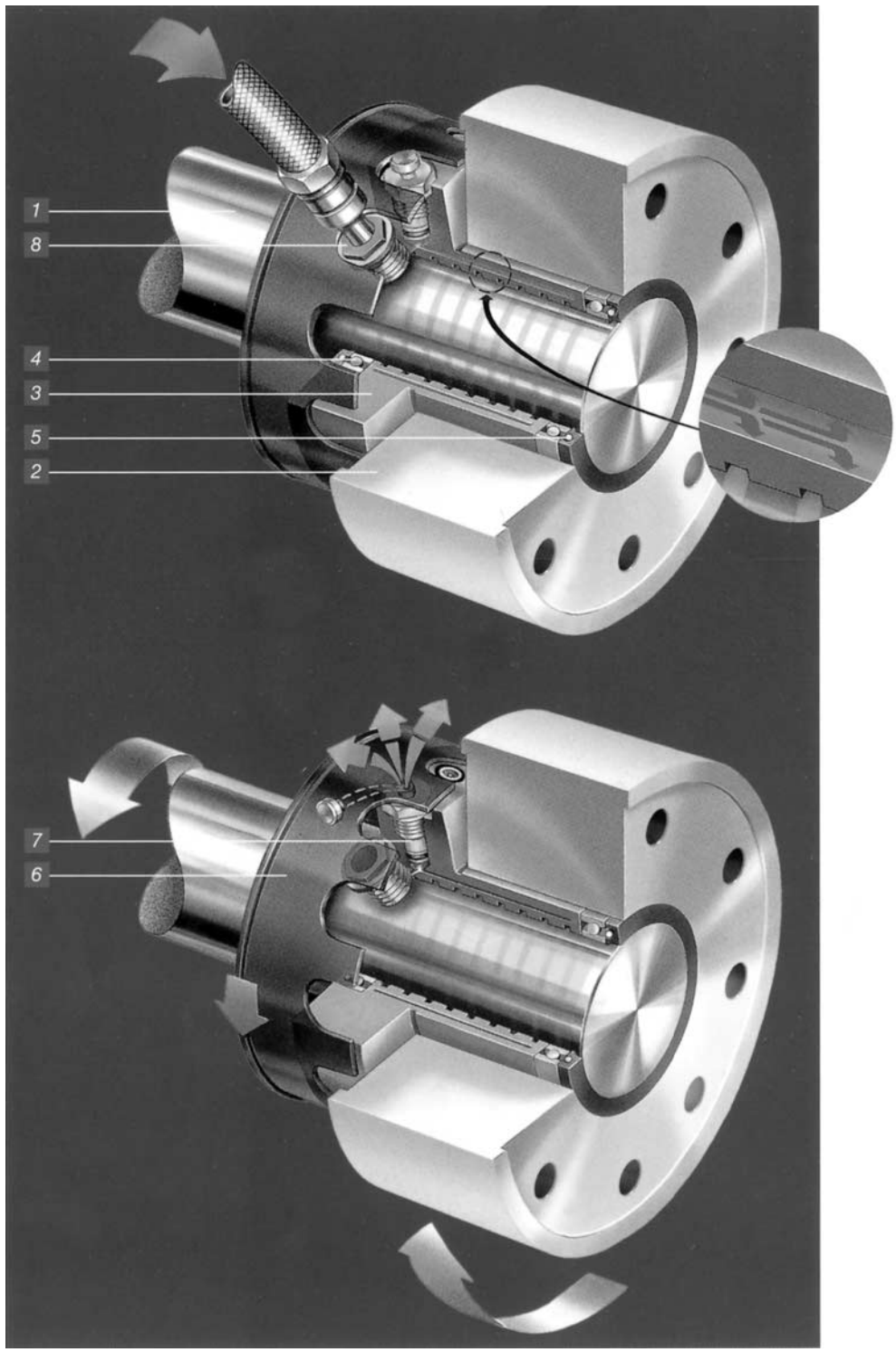


FIG. P-176 Safety coupling (Safeset) components. 1, shaft; 2, hub; 3, hollow steel sleeve; 4, antifriction bearings (on each side); 5, seals (on each side); 6, shear ring; 7, shear tube; 8, oil charge port. (Source: J. M. Voith GmbH.)

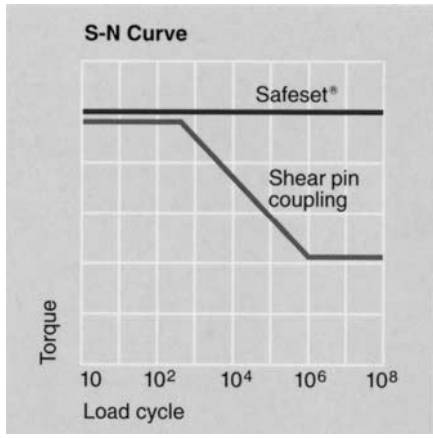


FIG. P-177 S–N curve for torque vs. load. (Source: J. M. Voith GmbH.)

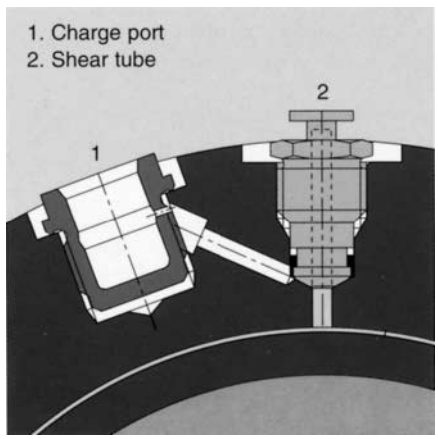


FIG. P-178 Shear tube. (Source: J. M. Voith GmbH.)

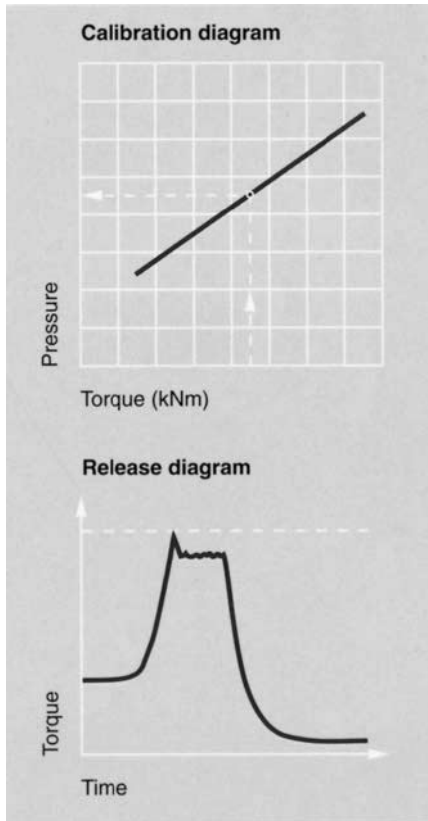
The coupling size is determined by the required release torque. This torque must be within the stated adjustment range. Practice has shown that in many cases, it is useful to have a torque safety margin of at least 20 percent.

The torque can be set below the lower limit, thereby reducing release torque accuracy. Less than 1/3 of the maximum value should never be used.

Safeset® ST series with sleeve bearing. For installation between a ground shaft and hub, with low operating speeds. This type is not axially located and cannot absorb axial thrust following release of the coupling. The sleeve bearing only allows limited radial and bending forces following release. (See Table P-21 and Fig. P-192.)

Safeset series SR-PF (heavy duty). This design is developed for heavy duties and preferably in conjunction with gear couplings, metal membrane couplings or universal joints. For this purpose connecting flanges on both sides are provided.

The couplings are designed and manufactured according to the customer's specifications.



179

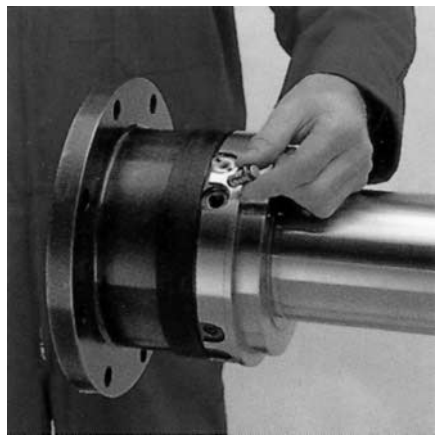
180

Calibration diagram (FIG. P-179) and release diagram (FIG. P-180). (Source: J. M. Voith GmbH.)

TABLE P-21 Safeset® ST Series with Sleeve Bearing

Size ST	M_A (kNm)	n_{max} (rpm)	d1 (mm)	d2 (mm)	d7 (mm)	L1 (mm)	L2 (mm)	L3 (mm)	G (kg)	J (kgm^2)
30	0.3–0.6	900	30	40	98	82	40	4	1.9	0.002
35	0.4–0.9	800	35	45	104	87	45	4	2.1	0.003
40	0.6–1.3	700	40	52	109	94	52	5	2.5	0.003
45	0.8–1.7	620	45	58	116	102	60	7	2.8	0.004
50	1.1–2.2	560	50	65	122	109	65	8	3.4	0.006
60	1.8–3.6	450	60	75	133	117	73	8	4.0	0.009
70	3.0–6.0	400	70	90	148	130	82	8	5.7	0.015
80	3.9–7.8	350	80	100	157	146	98	8	6.6	0.020
90	5.0–10.0	325	90	110	168	158	110	8	7.6	0.028
100	7.5–15.0	275	100	125	183	180	120	12	12	0.052
110	10.0–20.0	250	110	140	201	176	121	12	13	0.073
120	13.0–25.0	225	120	150	209	205	145	12	16	0.095
130	17.0–33.0	215	130	160	218	214	156	12	17	0.110
140	20.0–40.0	200	140	170	228	225	165	13	20	0.150
150	23.0–46.0	185	150	180	238	235	175	13	22	0.180
160	36.0–71.0	175	160	200	246	260	195	15	27	0.260
170	39.0–78.0	165	170	210	256	256	191	15	28	0.290
180	49.0–98.0	155	180	225	274	256	191	15	32	0.370
190	63.0–126.0	150	190	240	286	302	236	15	43	0.540
200	70.0–140.0	140	200	250	296	302	236	15	45	0.610
220	85.0–170.0	125	220	270	314	302	236	15	49	0.760

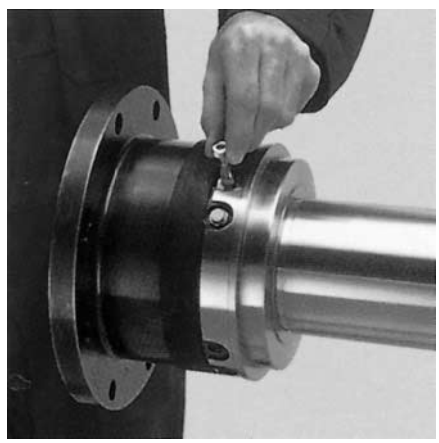
M_A, release torque—adjustment range; n_{max}, maximum permissible rpm; G, mass (weight); J, inertia moment.



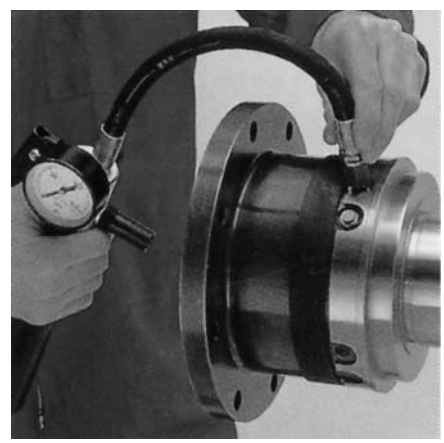
181



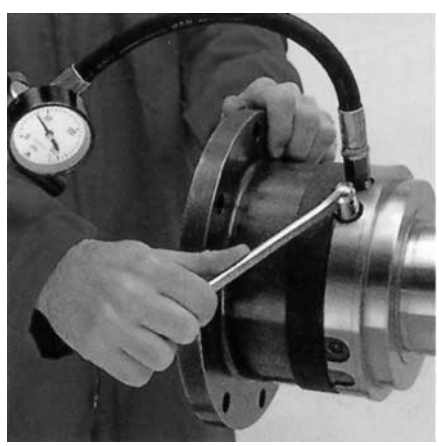
182



183



184



185



186

FIG. P-181 Remove the used shear tube. (Source: J. M. Voith GmbH.)

FIG. P-182 Insert a new shear tube. (Source: J. M. Voith GmbH.)

FIG. P-183 Remove the protection plug from the oil charge port. (Source: J. M. Voith GmbH.)

FIG. P-184 Connect the pump. Loosen the shear tube 1/4 of a turn. (Source: J. M. Voith GmbH.)

FIG. P-185 Pump in oil until the required pressure is attained. Tighten the shear tube. (Source: J. M. Voith GmbH.)

FIG. P-186 Release the pump pressure and disconnect the pump. Replace the protection plug in the oil charge port. (Source: J. M. Voith GmbH.)

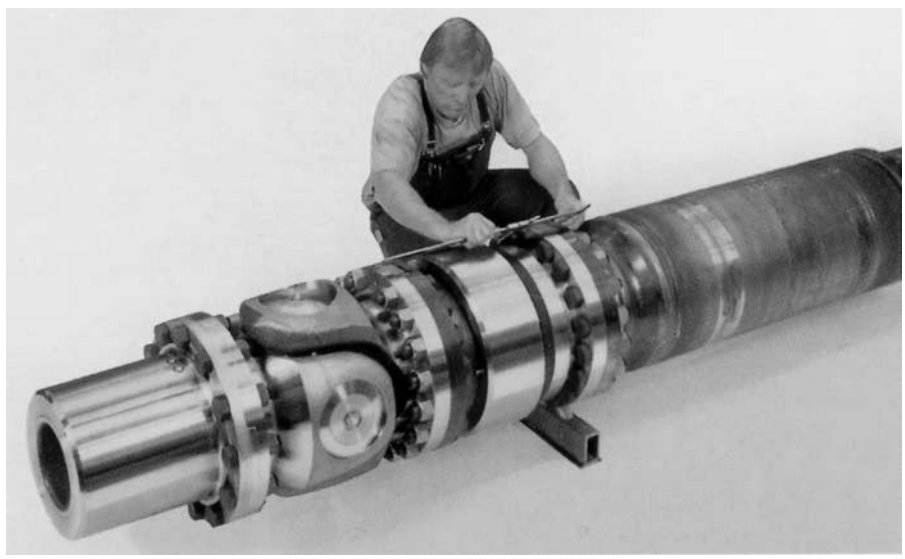
187



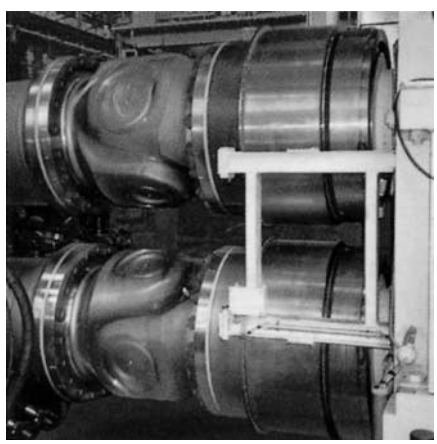


188

189



190





191

FIG. P-187 Assembling Safeset®-SR-F couplings in the workshop. (Source: J. M. Voith GmbH.)

FIG. P-188 Assembling Safeset®-SR-P couplings in the workshop. (Source: J. M. Voith GmbH.)

FIG. P-189 High-capacity universal joint shafts incorporating Safeset® couplings and Hygrip® connection coupling. (Source: J. M. Voith GmbH.)

FIG. P-190 High-capacity universal joint shafts and Safeset® couplings in a rolling mill stand. (Source: J. M. Voith GmbH.)

FIG. P-191 Safeset® couplings in a straightener. (Source: J. M. Voith GmbH.)

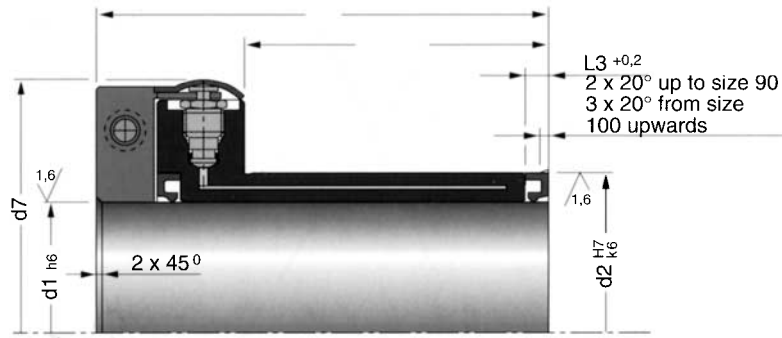


FIG. P-192 Safeset ST series. (Source: J. M. Voith GmbH.)

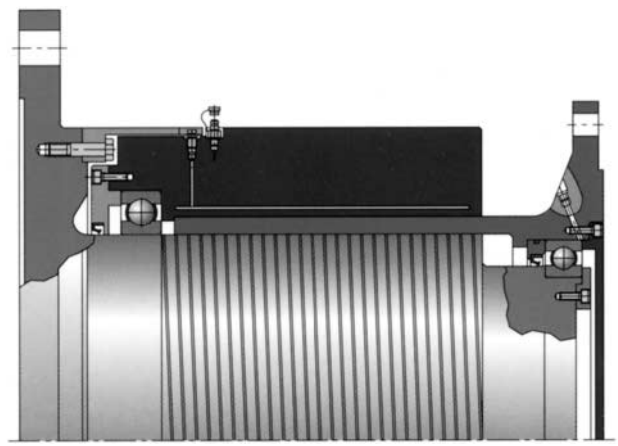


FIG. P-193 Safeset SR-PF 800 for a wind tunnel drive with a release torque of 7000 kNm. (Source: J. M. Voith GmbH.)

In this design the torque is not transmitted over the welded Safeset-ring but directly from a flange shaft to a flange sleeve. (See Figs. P-193 and P-194.)

Shear tubes. The couplings are equipped with 1–12 shear tubes of a suitable size depending on the application and the size of the coupling (Fig. P-195).

For external remote releases, e.g., on test benches, there are shear tubes with an extended head, if necessary.

Turbocoupling (Type "T")

The turbocoupling works on the Föettinger principle that was described earlier in this section. See also Fig. P-196. The power and speed of the drive motor determines coupling size (and model designation). See Figs. P-197 and P-198.

See Figs. P-199 through P-203 for special applications.

Different conditions require different starting procedures (characteristic curve) for the coupling. Important criteria in this respect are the mass moment of inertia, torque limitation, and frequency of startups.



FIG. P-194 Safeset SR-PF 710 with 4000 kNm release torque, during assembly for a rolling mill. (Source: J. M. Voith GmbH.)

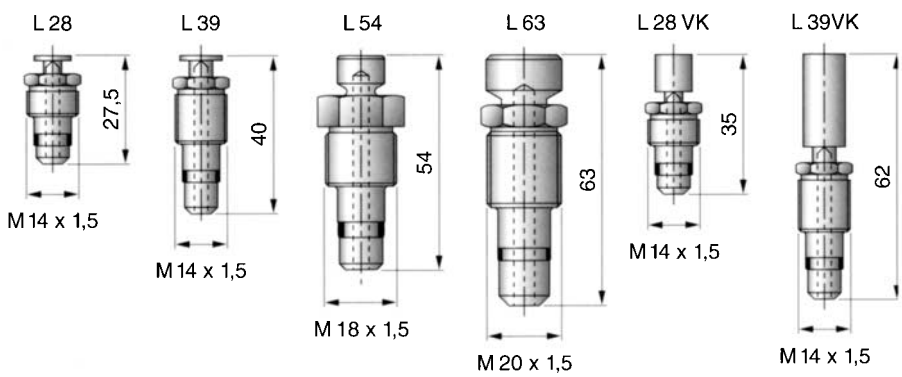


FIG. P-195 Standard sheer tubes. (Source: J. M. Voith GmbH.)

The turbocoupling "Type T" (this information source's designation) is the basic version of constant-fill couplings, consisting of pump wheel, turbine wheel, and outer shell (Figs. P-204 and P-205).

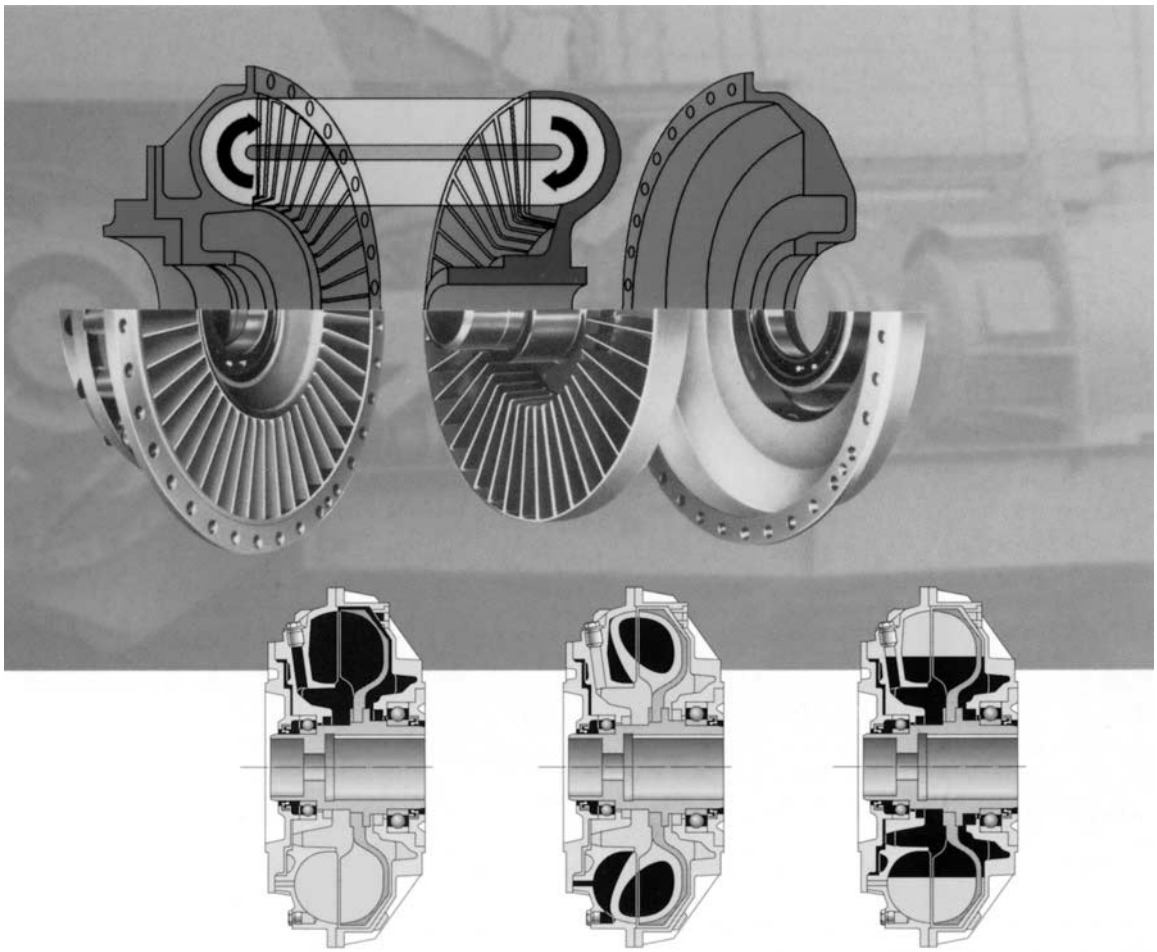
A further range has been created by the addition of other parts to this basic type.

The turbocoupling is normally mounted on the machine shaft or gearbox shaft to be driven (outer wheel drive). In order to compensate for any slight installation inaccuracies, a flexible connecting coupling is used to join coupling and input shaft.

Installation of this type of coupling is recommended when vibration and overload protection are required for motor and machine; they may also be used for simpler transmission systems in the lower performance range.

Applications include:

- Bucket-wheel equipment
- Excavators
- Mixing, kneading, and stirring machines



Stationary

All operating fluid in the coupling is static.

Starting

The pump-wheel accelerates the operating fluid as revolutions in the drive-motor increase, so that a circular current of fluid is created in the working chamber. The entire bladed surface of the turbine wheel is impacted and thus set in motion through kinetic energy. The torque development during the starting procedure is indicated by the characteristic curve of the coupling.

Operation at Normal Rating

Owing to the minimal difference in speed between the pump and turbine wheel, a constant flow is achieved in the coupling. Only the demand torque from the machine is transmitted.

Through skilful coordination of compensating chambers, such as the delay chamber and the annular chamber shell, the starting performance of the Turbo Coupling can be regulated (see characteristic curves in Fig. P-198).

FIG. P-196 Turbocoupling—components and operation. (Source: J. M. Voith GmbH.)

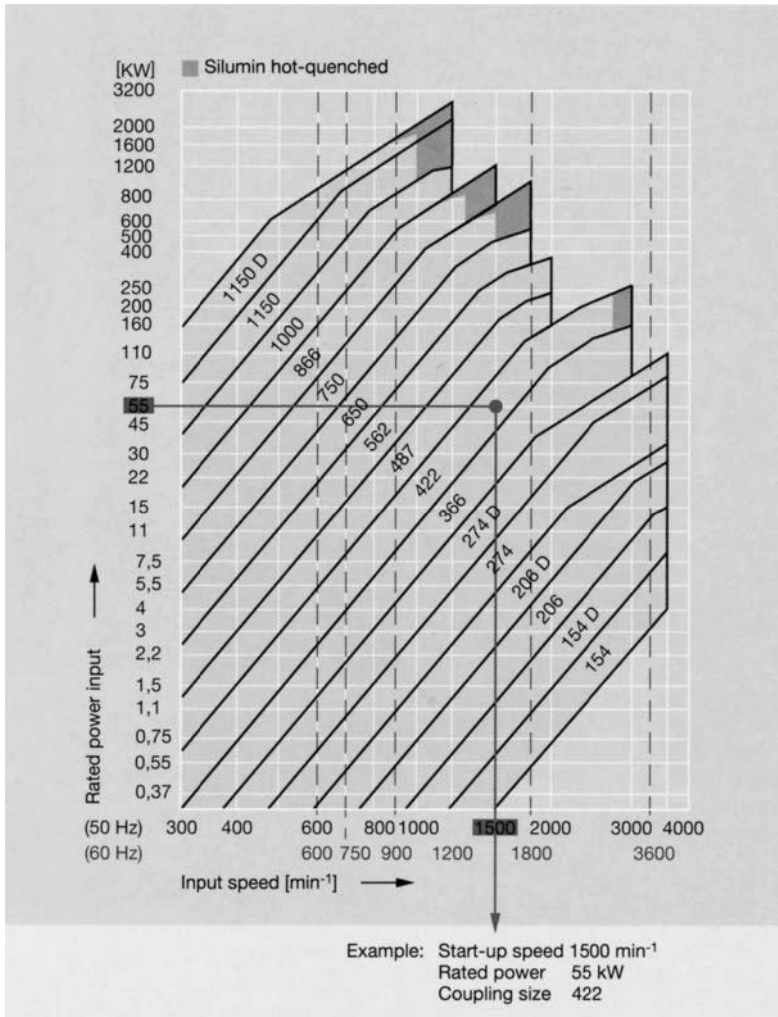


FIG. P-197 Turbocoupling characteristic. (Source: J. M. Voith GmbH.)

Type T
Type TV
Type TVV
Type TVVS

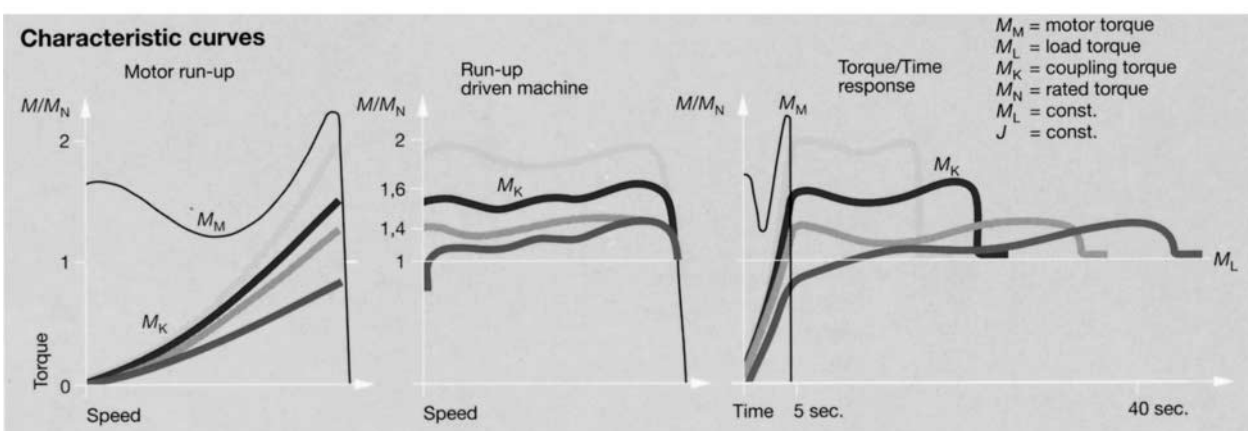


FIG. P-198 Turbocoupling characteristics of generic models. (Source: J. M. Voith GmbH.)

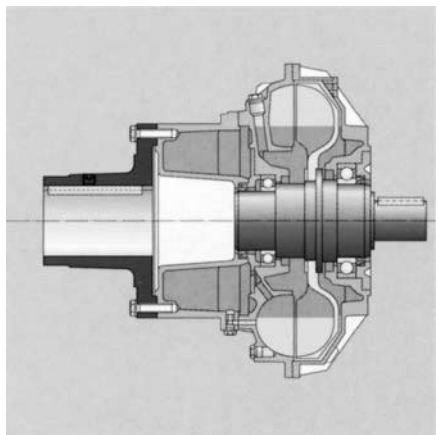


FIG. P-199 Turbocoupling with solid shelf and primary coupling flange. (Source: J. M. Voith GmbH.)

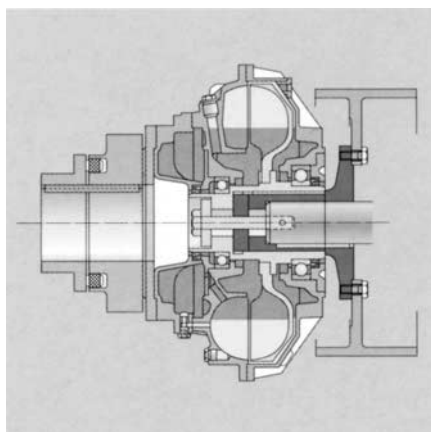


FIG. P-200 Turbocoupling with brake flange. (Source: J. M. Voith GmbH.)

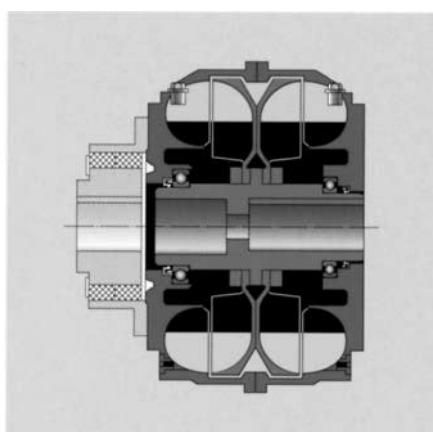


FIG. P-201 Turbocoupling with twin circuit "type DT." (Source: J. M. Voith GmbH.)

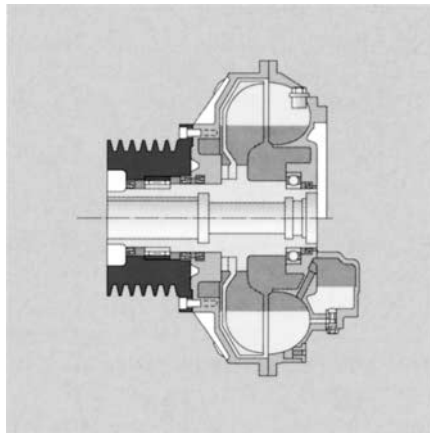


FIG. P-202 Pulley type coupling without bearing cover type TRI. (Source: J. M. Voith GmbH.)

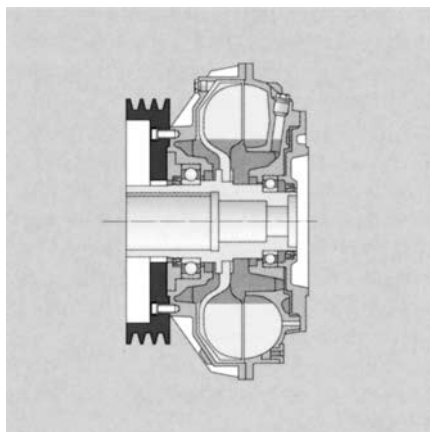


FIG. P-203 Turbocoupling with overhung pulley installation—type TR. (Source: J. M. Voith GmbH.)

Turbocoupling types "TV" and "TVV"

This version features a "delay-fill chamber" that is flange-connected to the outer wheel of the coupling (type TV) (see Figs. P-206 and P-207). At standstill, a proportion of the working fluid lies in this chamber, thus reducing the volume in the working chamber. Hence, on motor startup, a reduced-coupling torque is transmitted, while simultaneously providing an unloaded motor start. After the motor has run up, the working fluid flows from the delay-fill chamber into the working chamber, which smoothly accelerates the driven machine up to its operating speed.

Furthermore, if the application so demands, the delay-fill chamber can be further enlarged (type TVV), thus enhancing its effects and further reducing the coupling torque on motor start, as well as resulting in even longer and smoother startup of the driven machine.

In certain cases, the function of the delay chamber can be additionally improved through centrifugally controlled valves (type TVF) or through hydrodynamic refill (type TVY).

Applications include:

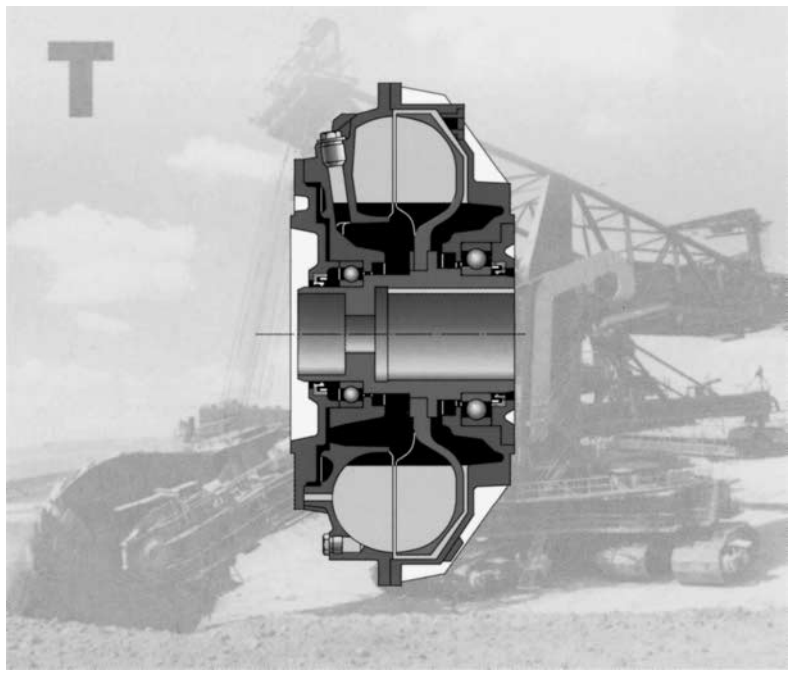
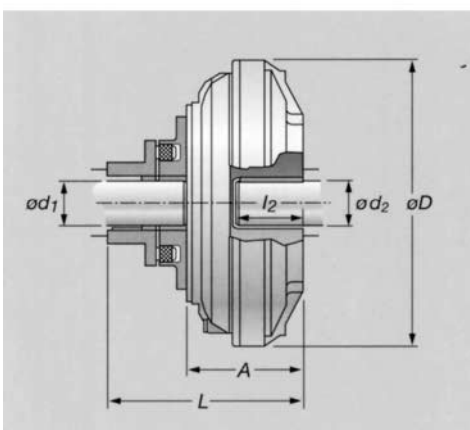


FIG. P-204 Section through turbocoupling type "T." (Source: J. M. Voith GmbH.)



Size	Type	A	D	L max.	d_1 max.	d_2 max.	l_2	Weight kg[1)]
154	T	80	190	143	32	28	60	4
154	DT	102	190	165	32	28	80	5
206	T	97	248	183	42	42	80	10
206	DT	137	248	223	42	42	115	13
274	T	135	328	291	65	55	90	21
274	DT	175	328	331	65	55	125	26
366	T	198	424	357	75	65	120	52
422	T	218	470	391	90	80	135	78
487	T	246	556	436	100	90	155	110
562	T	269	634	490	120	110	170	170
650	T	317	740	591	140	120	200	265
750	T	366	842	529	140	135	240	382
866	T	421	978	610	160	150	265	567
1000	T	441	1118	651	180	160	280	850
1150	T	505	1295	715	180	180	320	1165
1150	DT	830	1295	1040	180	170	550	1677

[1)] Weight with connecting coupling and max. oil filling.

FIG. P-205 Dimensions for the turbocoupling type "T" range. (Source: J. M. Voith GmbH.)

- Belt conveyors
- Chain conveyors
- Centrifuges, decanters
- Tube mills
- High-inertia machines

Turbocoupling type "TVVS"

A further development is the annular-chamber shell additional to the enlarged delay chamber (type TVVS). (See Figs. P-208 and P-209.)

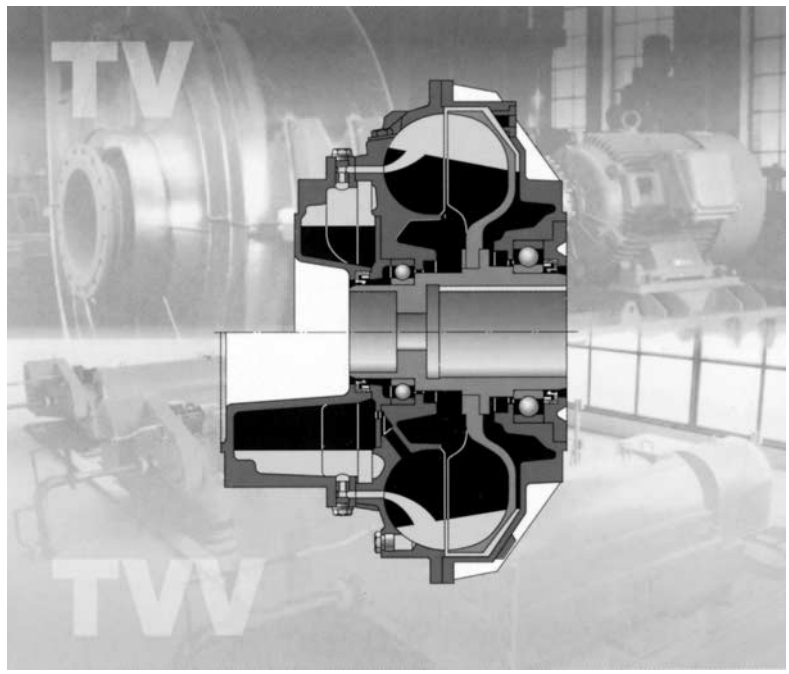
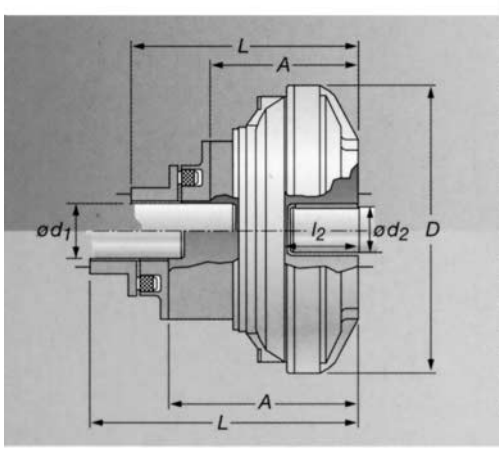


FIG. P-206 Section through turbocoupling types "TV" and "TVV." (Source: J. M. Voith GmbH.)



Size	Type	A	D	L	d_1 max.	d_2 max.	l_2	Weight kg[1)]
274	TV	172	328	328	65	55	90	32
274	TVV	204	328	360	65	55	90	33
274	DTV	244	328	403	65	55	105	38
366	TV	225	424	384	75	65	120	56
366	TVV	295.5		454.5	75	65	120	58
422	TV	257	470	430	90	80	135	80
422	TVV	335		508	90	80	135	83
487	TV	297	556	487	100	90	155	111
487	TVV	382		572	100	90	155	119
562	TV	333	634	554	120	110	170	172
562	TVV	428		649	120	110	170	182
650	TV	384	740	658	140	120	200	262
650	TVV	494		768	140	120	200	278
750	TV	440	842	603	140	135	240	390
750	TVV	567		730	140	135	240	408
866	TV	493	978	682	160	150	265	570
866	TVV	641		830	160	150	265	595
1000	TV	547	1118	757	180	160	280	870
1000	TVV	686		896	180	160	280	908
1150	TV	670	1295	880	180	180	320	1235
1150	TVV	883		1093	180	180	320	1289

[1)] Weight with connecting coupling and max. oil filling.

FIG. P-207 Dimensions for the turbocoupling types "TV" and "TVV." (Source: J. M. Voith GmbH.)

The additional chamber in the coupling shell enables further reduction of the starting torque. During the initial rotations of the startup procedure, centrifugal forces normally cause the outer chamber of the coupling to be completely filled with operating fluid from the bladed area.

In comparison with couplings without annular chamber, filling of the bladed area

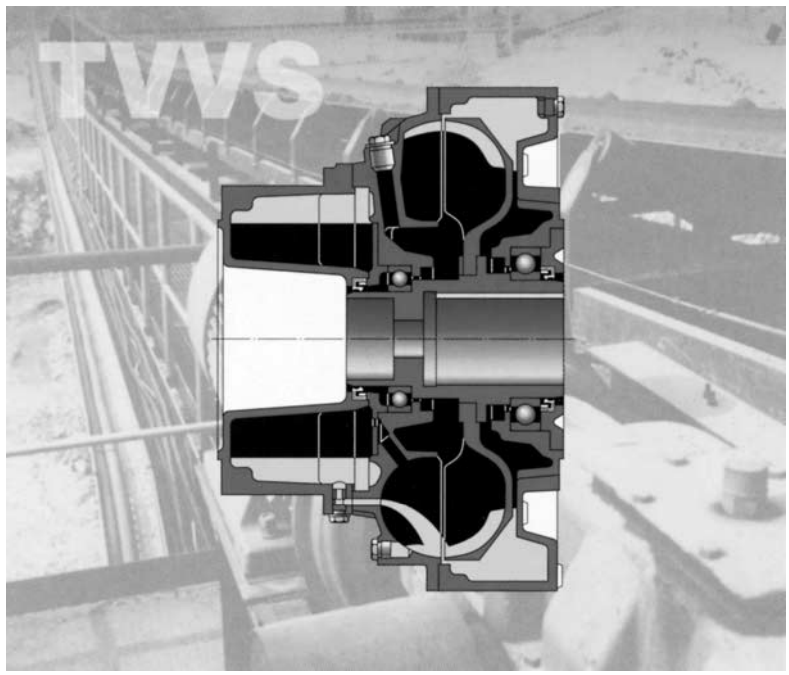
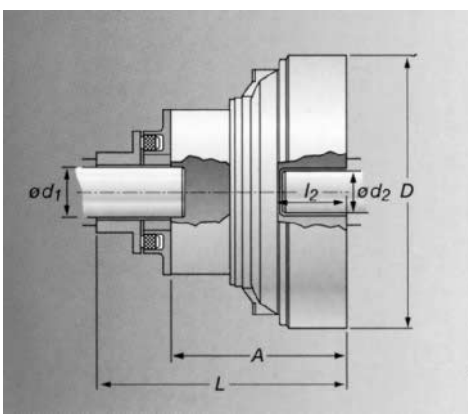


FIG. P-208 Section through turbocoupling type "TVVS." (Source: J. M. Voith GmbH.)



Size	Type	A	D	L	d_1 max.	d_2 max.	l_2	Weight kg[1)]
422	TVVS	335	470	508	90	80	135	87
487	TVVS	382	556	572	100	90	155	126
562	TVVS	428	655	649	120	110	170	192
650	TVVS	494	756	768	140	120	200	294
750	TVVS	567	872	730	140	135	240	425
866	TVVS	636	1012	825	160	150	265	645

[1)] Weight with connecting coupling and max. oil filling.

FIG. P-209 Dimensions for the turbocoupling type "TVVS." (Source: J. M. Voith GmbH.)

of a TVVS coupling is considerably slowed down, which, in turn, reduces the torque transmitted during motor runup.

The increase in torque then follows through gradual emptying of the fluid from the delay chamber into the bladed area.

The starting procedure can be adapted to the requirements of the application by adjustable nozzle screws.

This new concept for couplings was designed especially for conveyor belt drives. Through the gradual buildup of torque an automatic adaptation to belt load conditions is achieved.

Applications include:

- Belt conveyors

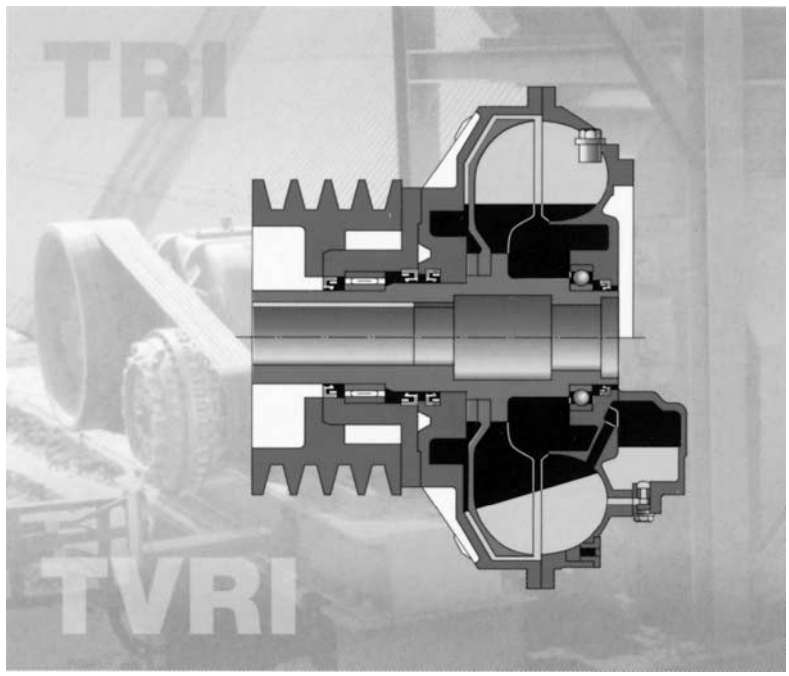
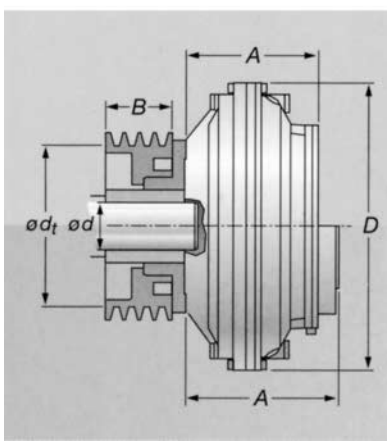


FIG. P-210 Section through turbocoupling type "TRI/TVRI." (Source: J. M. Voith GmbH.)



Size	Type	A	D	B max.	d_t min.	d max.	Weight kg[1)]
206	TRI	97	248	70	116	42	9
206	DTRI	137	248	70	116	42	12
274	TRI	137	328	100	150	55	25
274	TVRI	172	328	100	150	55	26
274	DTRI	175	328	135	165	60	33
274	DTVRI	242	328	135	165	60	38
366	TRI	198	424	145	160	65	47
366	TVRI	225	424	145	160	65	51
422	TRI	205	470	160	182	70	74
422	TVRI	257.5	470	160	182	70	76
487	TRI	246	556	201	233	90	110
487	TVRI	297	556	201	233	90	112
562	TRI	269	634	294	265	100	173
562	TVRI	333	634	294	265	100	175
650	TRI	317	740	–	423	105	256
650	TVRI	384	740	–	423	105	261

[1)] Weight with max. oil filling without pulley.

FIG. P-211 Dimensions for the turbocoupling type "TRI/TVRI." (Source: J. M. Voith GmbH.)

Turbocoupling type "TRI/TVRI"

The vee-belt or flat belt pulley that is flanged to the bearing cover allows various transmission ratios to be accommodated. If required, the pulley may be easily exchanged.

TRI- and TVRI-type turbocouplings are normally installed on the motor shaft in an overhung position. The belt force is supported by a bearing in the bearing cover on the coupling hub. (See Figs. P-210 and P-211.)

TRI-type turbocouplings can be installed both as starting aid and overload protection. Type TVRI with additional delay chamber is recommended if a particularly smooth startup is required.

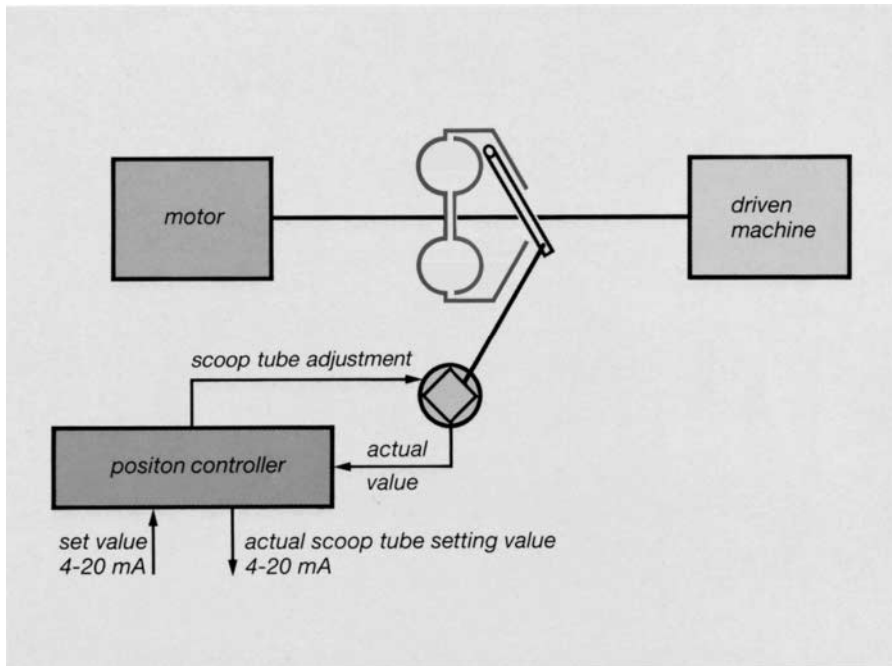


FIG. P-212 Position control circuit. (Source: J. M. Voith GmbH.)

Applications include:

- Centrifuges and decanters
- Fans
- Mixers
- Crushers

Integration of the variable-speed turbocoupling into a control circuit

Variable-speed turbocouplings serve to control the speed (rpm) or machinery. In many cases, the couplings are installed into an automatic process. (See Figs. P-212 and P-213.)

Position control circuit. Components:

- Position controller
- Actuator for continuous control
- Position feedback

Process control circuit. Components:

- Position controller
- Process controller

If the speed is used as a process input, or if it is to be displayed, the following components are required:

- Pole wheel
- Pickup

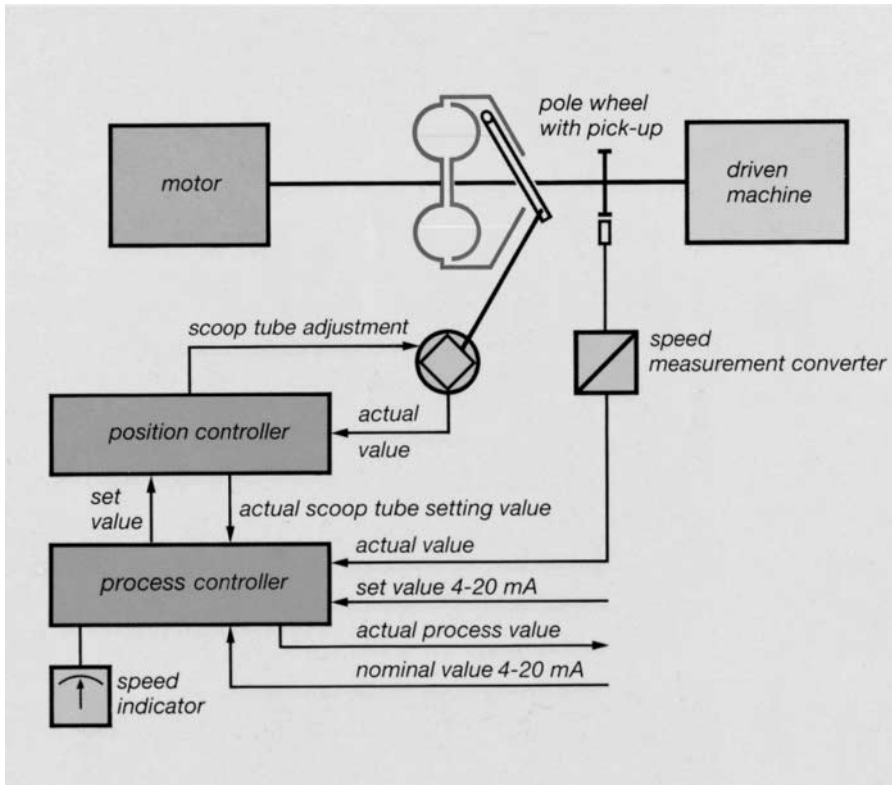


FIG. P-213 Process control circuit. (Source: J. M. Voith GmbH.)

- Converter
- Indicator

As with the speed (rpm), a process value (e.g., pressure, flow rate, etc.) can be installed into a control circuit. Then the process input is used as the set value in place of the speed.

Monitoring devices and accessories

Mechanical thermal switches (MTS). These switches are protection against overheating; fusible plugs are a standard fitment. See Fig. P-214.

In order to avoid loss of operating fluid through thermal overload, an MTS can be added.

On achieving the response temperature, the element activates a pin that then operates a switch.

Depending on the type of circuit, the signal can be used either as an alarm or to switch off the motor.

The circuit element has to be replaced after activation.

Noncontact thermal switch (BTS). Monitoring of coupling temperatures takes place without any contact. See Fig. P-215. After activation of the switch, no replacement of the element is required. It is ready for use as soon as the coupling has cooled down.

The signal can be used either as an alarm or to switch off the motor.

In some cases BTS can be used as a speed monitor.

Fitting and dismantling tool. Required for professional, safe fitting and dismantling. See Fig. P-216.

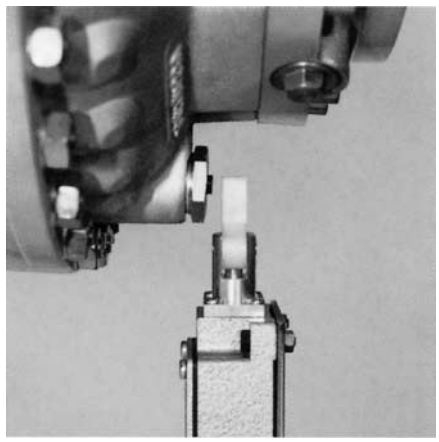


FIG. P-214 Mechanical thermal switches. (Source: J. M. Voith GmbH.)

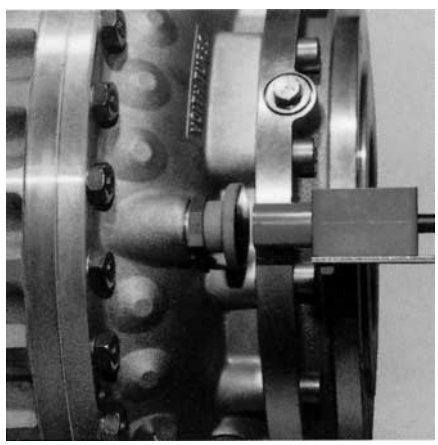


FIG. P-215 Noncontact thermal switch. (Source: J. M. Voith GmbH.)

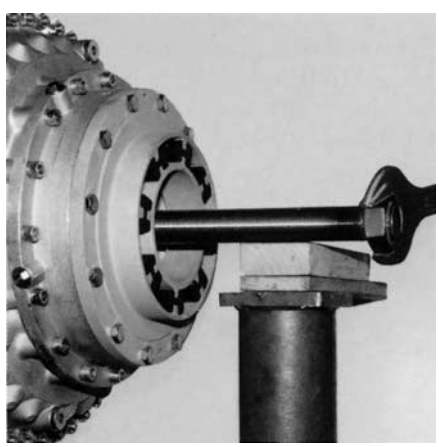


FIG. P-216 Fitting and dismantling fool. (Source: J. M. Voith GmbH.)

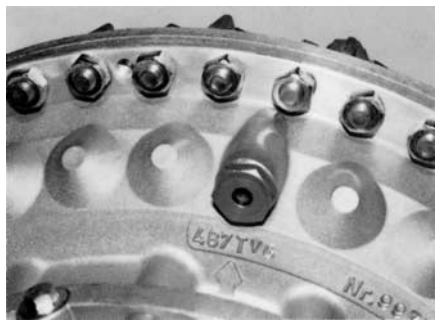


FIG. P-217 Sight glass. (Source: J. M. Voith GmbH.)

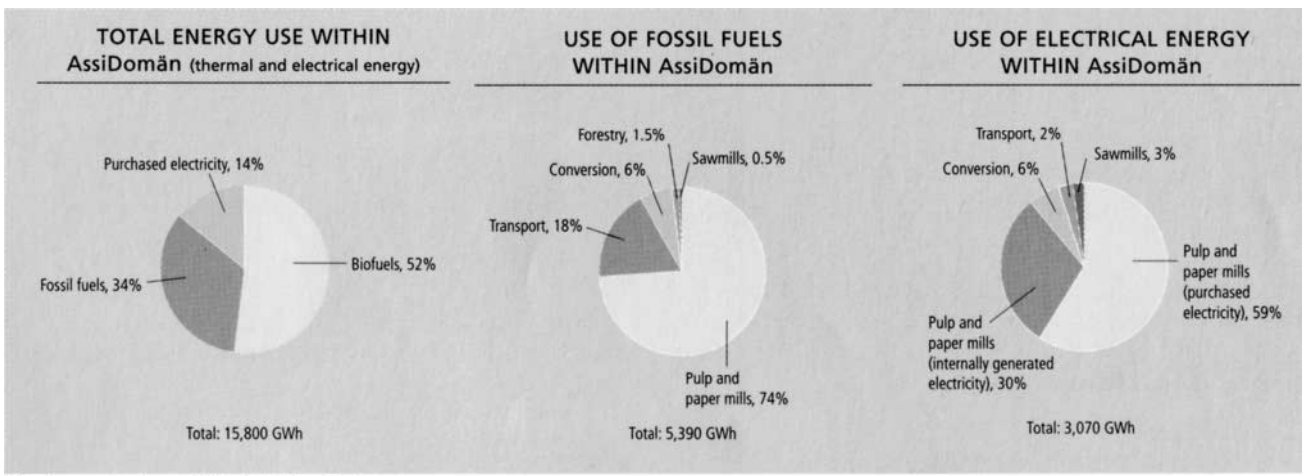


FIG. P-218 Corporate energy use. (Source: AssiDomän.)

As well as the mechanical fitting and dismantling tool, a hydraulic pressure device is available.

Sight glass. By fitting a sight glass, the fluid level in the coupling can be easily checked without opening the coupling. See Fig. P-217.

Pulp and Paper

The pulp and paper industry has its own institutes and handbooks. There are a few aspects that are worth noting in this book, including the fact that, like refining and most other branches of process engineering, aspects of paper's manufacture are coming under scrutiny in the increasing demand for operational optimization. Also, as in other industries, items that were formerly waste products are now being used as resources.

Energy Consumption of Pulp and Paper Mills*

Pulp and paper mills are big consumers of energy, but at the same time also major producers. See Fig. P-218.

* Source: AssiDomän, Sweden.

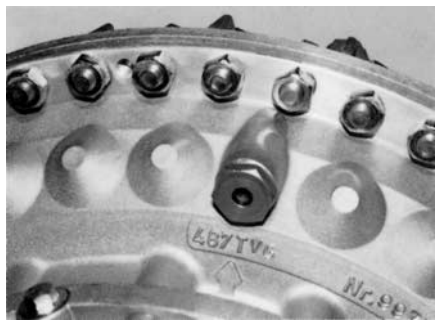


FIG. P-217 Sight glass. (Source: J. M. Voith GmbH.)

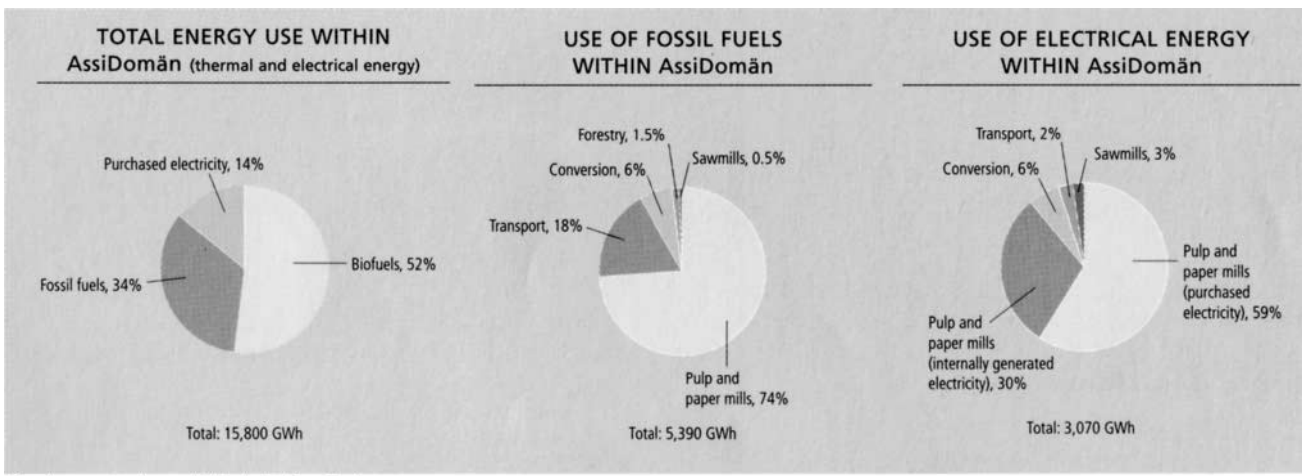


FIG. P-218 Corporate energy use. (Source: AssiDomän.)

As well as the mechanical fitting and dismantling tool, a hydraulic pressure device is available.

Sight glass. By fitting a sight glass, the fluid level in the coupling can be easily checked without opening the coupling. See Fig. P-217.

Pulp and Paper

The pulp and paper industry has its own institutes and handbooks. There are a few aspects that are worth noting in this book, including the fact that, like refining and most other branches of process engineering, aspects of paper's manufacture are coming under scrutiny in the increasing demand for operational optimization. Also, as in other industries, items that were formerly waste products are now being used as resources.

Energy Consumption of Pulp and Paper Mills*

Pulp and paper mills are big consumers of energy, but at the same time also major producers. See Fig. P-218.

* Source: AssiDomän, Sweden.



FIG. P-219 Energy from waste: Black liquor, which contains chemicals and dissolved organic matter from the wood, is used as an energy source in the pulp process. The liquor is evaporated and sprayed into the soda boiler, where it is burned. Energy and chemicals are recovered. The sulfate pulp mill is largely self-sufficient with regard to energy and over 95 percent of the chemicals are recovered. (Source: AssiDomän.)

In the manufacture of sulfate pulp, wood is cooked in white liquor. The purpose of the process is to dissolve lignin and other binders out of the wood and separate the cellulose fiber, which is used in paper manufacture. A waste product obtained from this process is black liquor, which contains organic matter dissolved out of the wood. The black liquor is reused in the pulp-making process by being burned in the soda recovery boiler. This generates virtually all the energy needed for pulp manufacture. At the same time, chemicals are recovered that are subsequently reused in the cooking process. Electrical energy (approximately one-third of the requirement) is generated in a steam turbine connected to a generator. In some cases, excess heat is produced, which is then sold to the district heating network. See Figs. P-219 and P-220.

Application Case Study 1: Biomass Fuel at a Paper Mill*

One of the main limiting factors when a green field pulp mill intends to increase its production is the capacity of the Tomlinson boiler by which the cooking chemicals are recovered. Since a boiler of this kind has a service life of some 30 years and represents a substantial investment, it is not an easy decision to replace it for a minor production increase. This study takes a novel approach to the problem. The excess black liquor is assumed to be recovered in a black liquor gasifier, an emerging technology for alternative black liquor recovery. Black liquor gasification (BLG) in itself has been studied considerably during the last decade. In this study it is combined with the gasification of biomass, i.e., bark and wood, which is already available at a pulp mill. The resulting hybrid energy system, with a gas turbine and a gas expander, offers interesting possibilities: good off-design performance and high efficiency with respect to the fuel quality.

Following the referendum in 1980, it was decided by the Swedish Parliament that all nuclear power would be phased out from the nation's energy supply system by the year 2010. That deadline has been pushed forward into the future, but there remains a need for novel ways to generate power. As hydropower is exploited nearly to its full extent already and fossil firing is ruled out for environmental reasons,

* Source: Adapted from extracts from Ahlroth and Svedberg, "Case Study on Simultaneous Gasification of Black Liquor and Biomass in a Pulp Mill," ASME paper 98-GT-350.

The illustration shows the path of the wood fibre from plant nursery to finished product – pulp, paper, packaging or sawn timber – for further transport to the customer.

On the way, resources in the form of raw materials and energy are used and emissions take place to air, water and land.

Resource use and emissions associated with AssiDomän's total activities in 1997 have been calculated and are reported in the table below.

Internally generated electricity

Purchased electricity

Emissions to air

Fossil fuels

Chemicals

RAW MATERIAL USE	
Harvested own forest, excl. standing timber, sm^3ub	6,190,000
Wood raw material to pulp and paper mills, sm^3ub	9,430,000
Plastic raw material, tonnes	31,500
Water, mill. m^3	167
ENERGY USE	
Biofuels, TJ	29,600
Fossil fuels, TJ	19,400
Purchased electricity, GWh	2,160
Internally generated electricity, GWh	920
EMISSIONS TO AIR [2]	
Sulphur dioxide, SO_2, tonnes	12,500
Nitrogen oxides, NO_x, tonnes	9,120
Carbon dioxide from fossil fuels, CO_2 fossil, tonnes	1,500,000
Carbon dioxide from biofuels, CO_2, biogenic, tonnes	2,880,000
EMISSIONS TO WATER	
Oxygen-demanding substances, COD, tonnes	48,700
Organochlorines, AOX, tonnes	78
Nitrogen, N, tonnes	545
Phosphorus, P, tonnes	67
RESIDUAL PRODUCTS AND WASTE	
Hazardous waste, tonnes	1,660
Landfilled material, tonnes	360,000

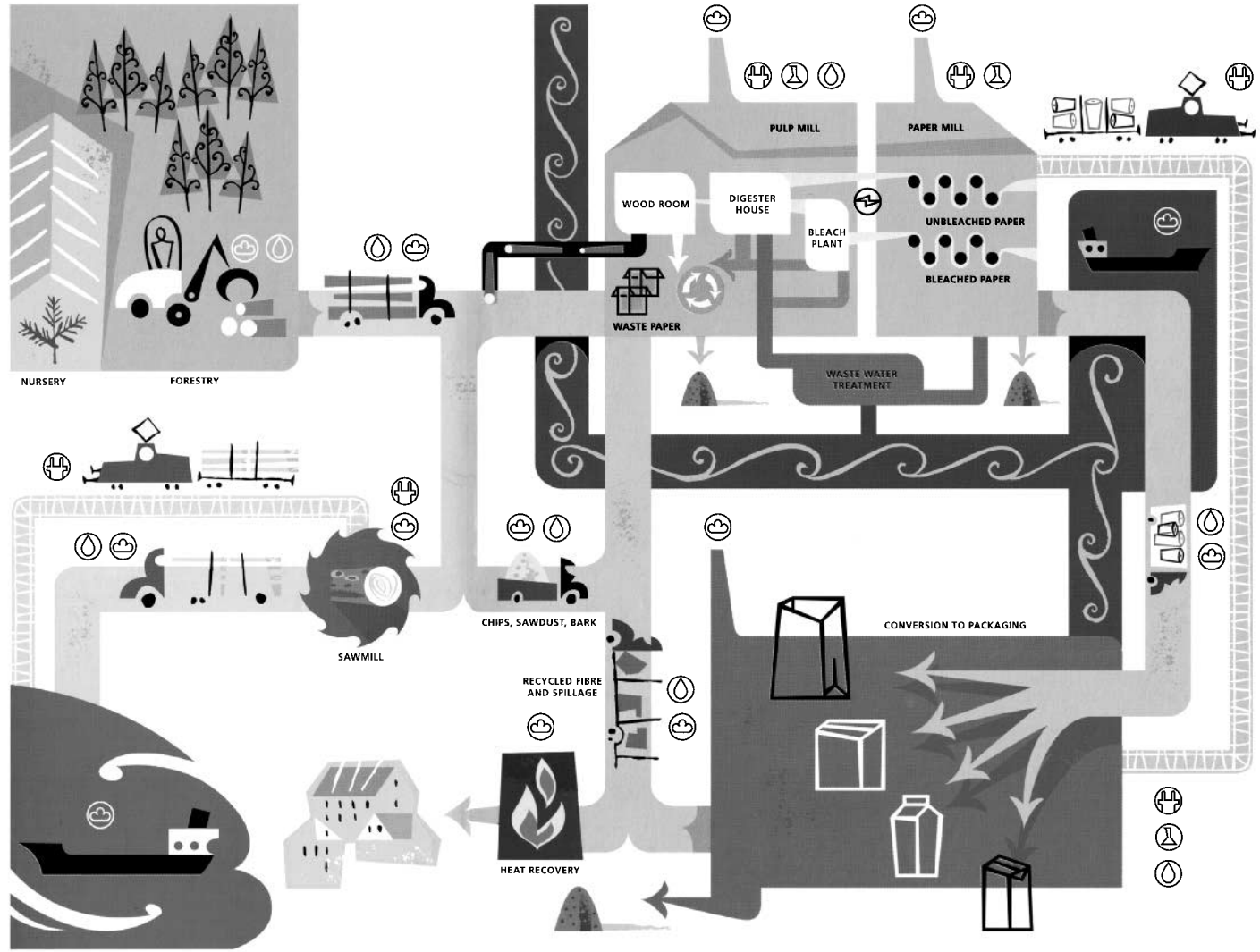


FIG. P-220 Industrial and eruptional flow differences. (Source: AssiDomän.)

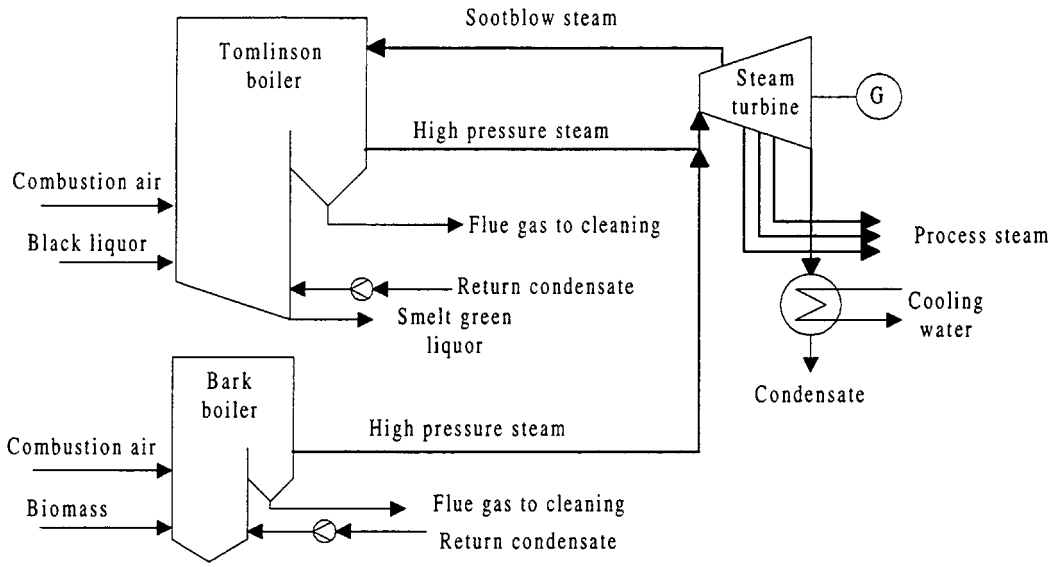


FIG. P-221 Original black liquor recovery system and bark boiler. (Source: Ahlroth and Svedberg.)

the most promising candidate is biomass firing. The industrial structure of Sweden makes this fuel an interesting option. The pulp and paper industry, for many years the backbone of Swedish industry, is already a major user of biomass in the form of black liquor, waste wood, and bark. In 1995, biomass contributed around 38.6 TWh of a total of 65.7 TWh of energy used in the pulp and paper industry in Sweden. Therefore it is logical to try to increase the use of biomass for power production in this particular sector, since the necessary fuel handling equipment is already built and the fuel is already available at a pulp mill. From a national viewpoint, use of indigenous fuels lowers the need for imported fuels and creates jobs in rural areas.

Pulp mills are sometimes faced with a serious problem resulting from an increased pulp production, namely, the inability to recover all the black liquor produced. Black liquor is the kraft pulping liquid that contains valuable cooking chemicals and dissolved substances from the wood raw material. It is classified as a renewable biomass-based fuel and better utilization of it is favored. The black liquor is combusted in a special type of boiler called a Tomlinson boiler primarily to recover the cooking chemicals, but also to produce process steam and power. This is done using a conventional Rankine steam turbine cycle. Thermal efficiency is generally poor, 15 percent is seldom exceeded. The boiler is large in size, represents a substantial investment, and has a service life of 30 years or more. Therefore, it is understandable that it acts as a bottleneck if pulp production is to increase beyond design capacity.

The objective of this study is to investigate how to increase biomass-based power production and simultaneously deal with insufficient black liquor recovery capacity. The study is based on the following scenario: a pulp mill with a capacity of 1160 t market pulp/day intends to increase its production to 1320 t market pulp/day. The existing Tomlinson recovery boiler is operating at maximum recovery capacity and has a long remaining service life ahead. The mill is also equipped with a bark boiler (see Fig. P-221). Steam produced by both boilers is led into a condensing turbine, producing power and process steam. The use of condensing turbines is

TABLE P-22 System Data on the Original Black Liquor Recovery System and Bark Boiler

Steam demand				
Saturated process steam	bar	3	5	12
Mass flow	t/h	24.6	206.7	76.5
Sootblow steam	bar	35		
Mass flow	t/h	11.7		
Steam production		Tomlinson boiler		Bark boiler
Mass flow	t/h	319.5		56
Pressure	bar	60		60
Temperarure	°C	450		450
Power production		Condensing steam turbine		
Condensing pressure	bar	0.37		
Isentropic efficiency	%	80		
Power output	MW	47.9		
Fuel		Black liquor		Biomass
Tomlinson boiler	t/h	95		
Dry content	%	75		
Bark boiler	t/h			23.2
Moisture content	%			15

rather unusual; however, today most Swedish pulp mills have only back pressure turbines.

System data are shown in Table P-22. With the enlargement, the flow of black liquor increases from 95 t DS/h (75 percent dry substance content) to 108 t DS/h (75 percent dry substance content), but as stated, the additional black liquor must be recovered by other means than the Tomlinson boiler. This is done using a small retrofit BLG unit. Contrary to contemporary studies, the low Btu gas from the gasifier is combusted in a duct burner instead of in a gas turbine and the exhaust is led to a one-pressure heat recovery steam generator (HRSG) for combined heat and power (CHP) production. The condensing turbine is converted into a back pressure turbine for CHP production. Also, the bark boiler is replaced with an airblown biomass gasifier (BIG) operating at about atmospheric pressure. The product gas is combusted in a gas turbine and the exhaust is used as preheated air in the duct burner (see Fig. P-222).

Modeling the process

Four cases are simulated for the hybrid energy system shown in Fig. P-222:

- The reference case, which represents the normal operation of the BLG and the BIG
- The low heat demand case, where no steam is drawn from the steam turbine as process steam; all steam is used for power production

The remaining two cases are off-design cases:

- The BLG suffers a malfunction and gets off line
- The BIG suffers a malfunction and gets off line

Equipment assumptions for the various unit operations in the hybrid energy system are shown in Table P-23. The individual unit operations will be described in detail.

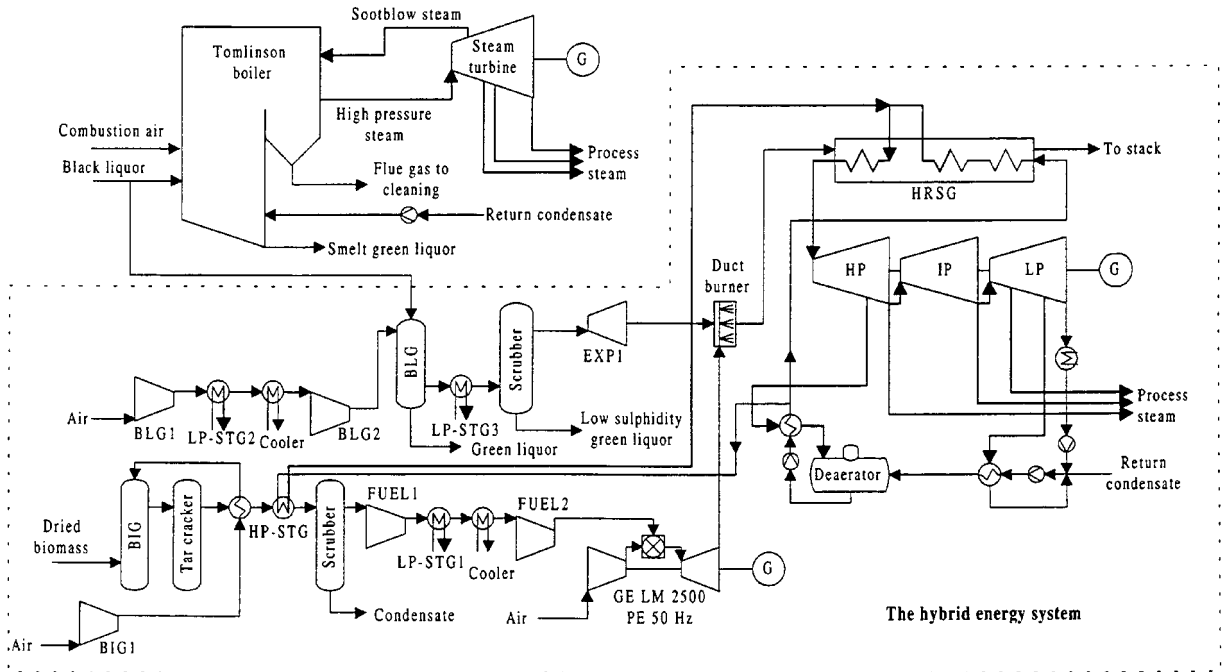


FIG. P-222 The hybrid energy system and the modified Tomlinson boiler with the back pressure turbine. (Source: Ahlroth and Svedberg.)

The black liquor gasifier (BLG). The BLG in this study is a Kvaerner Chemrec airblown atmospheric gasifier. At present, one unit of this kind has been built at the Frövi kraft mill in Sweden (100 t DS/day) and one is situated at New Bern (350 t DS/day) in the United States. Recently, the Swedish pulp manufacturer AssiDomän decided to build the world's first full-sized, pressurized, oxygen-blown BLG in Piteå in northern Sweden. The BLG in the hybrid-energy system studied would have a capacity of 312 t DS/day. The gasification in itself is not simulated since the purpose of this study is to investigate the hybrid-energy system, not the gasification. The reason for not fixing the black liquor synthesis gas in the gas turbine is to avoid the need for high-performance gas cleaning. The cooking chemicals in the black liquor contain large amounts of sodium, which must be kept away from the sensitive gas turbine expander blades. The amount of fine dust in raw synthesis gas is unacceptably high for use in a gas turbine. Therefore, expensive hot dust cleaning is needed. On the other hand, a duct burner is less sensitive to sodium and other fine particulates present in the synthesis gas, making the hot gas filter unnecessary.

The BLG is treated as a "black box" in the simulation. Only the parts that interact with the rest of the system are included, such as compressors, heat exchangers, and steam generators, where process steam is raised.

The biomass gasifier (BIG). The BIG is a Termiska Processer AB (TPS) atmospheric gasifier operating at 900°C. TPS of Nyköping, Sweden, is a small Swedish development company specializing in biomass gasification technology. A 12 MW TPS gasifier using refuse-derived fuel is operational at Greve-in-Chianti, Italy. In the same way as stated above, the BIG is modeled as a "black box." The input flow of wet biomass (50 percent moisture content) is dried to 15 percent moisture content

TABLE P-23 Equipment Assumptions for the Hybrid System

Compressors		FUEL1	FUEL2	BIG1	BLG1	BLG2
Working fluid		BIG syngas	BIG syngas	Air	Air	Air
Inlet pressure	bar	1,3	7,1	1	1	16
Outlet pressure	bar	7,1	39	1,3	16	25
Power consumption	MW	2,6	2,6	0,2	2,5	0,3
Isentropic efficiency	%	78	78	78	78	78
Turbines		**EXP1**	**HP**	**IP**	**LP**	
Working fluid		BLG syngas	Steam	Steam	Steam	
Inlet pressure	bar	23,5	105	12	5	
Outlet pressure	bar	1	12	5	0,37	
Power production	MW	1,1	9,5	2,3	2,6	
Isentropic efficiency	%	63	80	80	80	
Extractions: Pressure	bar		17//12	5	3//1	
Extractions: Mass flow	t/h		9//10,5	24,4	1,8//2,7	
Steam Generators		**HP-STG**	**LP-STG1**	**LP-STG2**	**LP-STG3**	
Steam production	t/h	11,5	1,4	2,5	1,6	
Pinch gas/water	°C	31	20	20	20	
Steam quality		Sat. 105 bar	Sat. 5 bar	Sat. 5 bar	Sat. 3 bar	
Total pressure drop	bar	1	0,25	0,5	0,5	
Calc. surface area	m^2	894	423	351	570	

HSRG		Superheater	Evaporator	Economizer
Mass flow	t/h	72,7	61,2	61,2
Inlet pressure	bar	105,1	106,1	106,1
Inlet temperature	°C	315	300	188
Outlet pressure	bar	105	105,1	106,1
Outlet temperature	°C	540	315	300
Pinch gas/water	°C		41	56
Calc. surface area	m^2	1,170	3,392	6,027

Gas Turbine		Model Based on the General Electric LM 2,500 PE 50 Hz Gas Turbine
Compressor mass flow	kg/s	40
Isentropic efficiency	%	88
Outlet temperature	°C	358
Outlet pressure	bar	13,7
Turbine mass flow	kg/s	49,4
Isentropic efficiency	%	90
TTT	°C	1,177
Inlet pressure	bar	13,2
Net power output	MW	18,8

All pumps have an isentropic efficiency of 82 percent.

with hot flue gas prior to gasification. Heat generated by the gasification process is used to raise high-pressure steam for the CHP plant and process steam for the pulp mill.

The gas turbine. The gas turbine is modeled based on a standard General Electric LM2500 PE 50 Hz. The model is taken from the gas turbine library in GateCycle™ and modified. The pressure ratio and the turbine geometry is unchanged but the flow of combustion air is reduced to 40 kg/s. This is due to the low heating value of the gasified biomass product gas. More than 60 mole% of the product gas is inert

gas, so in order to increase the turbine inlet temperature (TIT) to a reasonable level, the airflow is reduced. With 40 kg/s of air, the TIT is 1177°C. The total flow of gas entering the gas turbine expander is 49.2 kg/s when gasified biomass gas is used.

The duct burner. The syngas from the BLG is combusted with the gas turbine exhaust as the oxygen source. The resulting flue gas has a temperature of 900°C when leaving the burner. This temperature is the practical maximum inlet temperature for auxiliary fired HRSGs.

The HRSG unit. The heat recovery steam generator is a simple one pressure unit producing 20.2 kg/s of superheated steam (105 bars, 540°C). After the boiler feed water (BFW) heater, the water flow is split into two different streams and 3.2 kg/s is drawn from the main flow and led to the HP-STG steam generator connected to the BIG. The remaining 17 kg/s is led to the HRSG unit where it flows through the economizer and the steam generator. The two streams reunite prior to the superheater and they are heated to the final temperature together.

The CHP plant. The turbine set consists of three turbines (HP, IP, and LP, respectively) with extractions at 3, 5, and 12 bars for heat production. The unit is modeled based on the existing CHP plant in the city of Enköping, Sweden, with similar performance and steam data.

The Tomlinson boiler and the back pressure turbine. The mass flow of steam generated in the Tomlinson boiler, 319.5 t/h, is unchanged by the installation of the hybrid energy system. The omission of the bark boiler and the higher exit pressure, 3 bars, from the turbine reduces the power output from 47.9 MW to 35.6 MW. The change from a condensing turbine to a back pressure turbine is necessary, since the steam produced by the Tomlinson boiler is only enough to cover part of the process steam demand and the entire sootblow steam demand.

Results. The simulation is performed using GateCycle™, a program package by Enter Software. Input data are shown in Table P-24. Process and sootblow steam is assumed to be saturated. All pressures are absolute pressures. Steam production is calculated using a minimum pinch point of 20°C between gas and liquid phases in the steam generators. The studied cases are numbered as follows:

1. The original black liquor recovery system and bark boiler.
2. Hypothetical status quo case.
3. The reference case for the hybrid energy system.
4. The low heat demand case for the hybrid energy system.
5. The BIG malfunction case for the hybrid energy system.
6. The BLG malfunction case for the hybrid energy system.

In the first case, no simulation is needed. The performance of the original system, shown in Figure P-221, is taken from Table P-23.

The power output and the steam production for the original system is extrapolated to match the increased flow of black liquor and biomass after the enlargement of the pulp production. This case is called “hypothetical status-quo case” and it is used for comparison only. The power output in this case is 54.7 MW. In the reference case, the hybrid-energy system and the modified recovery system shown in Fig. P-222 are simulated. In the off-design cases, the calculated surface area from case number 3 is kept constant while all other parameters are subject to

TABLE P-24 Input Data to the Simulated Hybrid Energy System

Steam demand				
Saturated process steam	bar	3	5	12
Mass flow	t/h	28	235	87
Sootblow steam	bar	35		
Mass flow	t/h	11.7		
Fuel		Black liquor	Biomass	
			Wet	Dry
Total flow	t/h	108	26.4	15.5
Tomlinson boiler	t/h	95		
Black liquor gasifier	t/h	13		
Dry content	%	75		
Moisture content	%		50	15

Gasification products

Composition after gas cleaning prior to FUEL1 and EXP1

Component in mole%	Biomass	Black liquor
CO	19.33	14
CO_2	13.73	12.2
CH_4	3.17	1.5
C_2H_4	1.05	
C_2H_6	0.02	
H_2	14.88	16.6
H_2O	3.26	6.1
N_2	44.02	49.5
H_2S		0.0002
Ar	0.54	
Temperature	°C 30	110
Pressure	bar 1.3	23.5
Flow	kg/s 9.42	8.1
LHV	kJ/kg 5,032	3,698

Ambient conditions				
Temperature	°C	5		
Pressure	bar	1.013		
Relative humidity	%	50		

change. In the low heat demand case, some surplus process steam is produced in steam generators connected to the BIG and BLG compressors and the hot black liquor synthesis gas exiting the gasifier. For the BIG malfunction case there is a slight steam deficit for 3 and 5 bar steam and very low power production. Apart from this, the mill should be able to operate at full capacity.

In case of BLG breakdown it is assumed that pulp production is lowered to 1160 t/d. Usually a pulp mill has a short-term buffer against interruptions in black liquor recovery, but in this case the BLG breakdown is presumed to be long term. Steam is produced by cooling of the hot gas exiting the FUEL1 compressor in the biomass gasifier unit. The gas turbine and the HRSG are operated as a cogeneration plant. The higher net power output from the gas turbine in this case is due to the fact that the air compressors BLG1 and BLG2 in the BLG are not operating for the moment. The results are shown in Table P-25 and Figs. P-223 and P-224.

Discussion. From a pulp and paper industry viewpoint, production of pulp and paper is the primary mission, while power generation is of secondary importance.

TABLE P-25 Simulation Results for the Hybrid System and the Tomlinson Boiler

Power Production	Power Output (MW)		Thermal Efficiency (%)
	Gas turbine	Steam turbine	
Reference case	11.8	13.8	33.1
Low heat demand case	11.8	17.8	38.2
Biomass gasifier malfunction		2.1	7.0
Black liquor gasifier malfunction	13.5	8.1	45.6
Tomlinson boiler	35.6		
Steam Production	Saturated Steam Produced in t/h		
Pressure in bar	3	5	12
Reference case	3.4	28.3	10.5
Low heat demand case	1.6	3.9	
Biomass gasifier malfunction	1.6	16.4	10.5
Black liquor gasifier malfunction		1.4	
Tomlinson boiler	24.6	206.7	76.5
Steam demand	28	235	87
Production Minus Demand (t/h)	"+" is surplus and "–" is deficit		
Reference case	Balance	Balance	Balance
Low heat demand case	+1.6	+3.9	Balance
Biomass gasifier malfunction	–1.8	–11.9	Balance
Black liquor gasifier malfunction	Balance[1]	+1.4	Balance[1]

1. Steam demand is lower in this case.

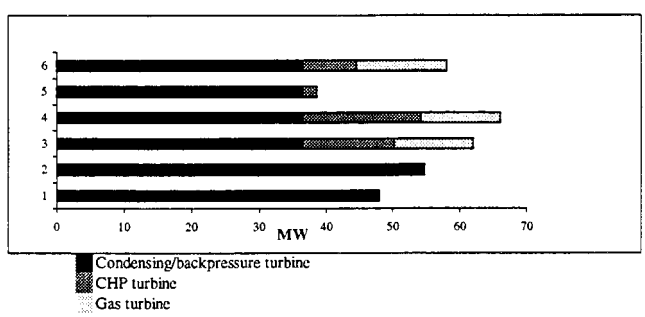


FIG. P-223 Power output for different scenarios. (Source: Ahlroth and Svedberg.)

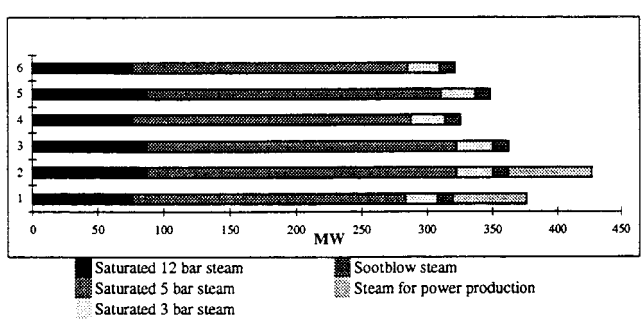


FIG. P-224 Steam production for different scenarios. (Source: Ahlroth and Svedberg.)

It is therefore noticeable that in all studied cases with a hybrid energy system instead of a conventional one, the process steam demand for the pulp mill is satisfied at all points of operation, except when the biomass gasifier suffers a malfunction. The introduction of the hybrid-energy system, with black liquor gasification as a unit operation, enables an increase in pulp production as planned.

The thermal efficiency of the hybrid system is 33 percent for the reference case based on the LHV for the gasified fuels shown in Table P-24. Compared to modern gas turbine combined cycle plants, with efficiencies up to nearly 60 percent this might seem as mediocre performance. It should be kept in mind, however, that the fuel used is difficult to refine to such a degree that it can be used in a gas turbine and that it has a low heating value. Secondly, comparing the efficiency of the Tomlinson boiler and the steam turbine cycle to the efficiency of the hybrid-energy system, the potential of the gas turbine in the pulp and paper industry is demonstrated. For a 12 percent increase in fuel input to the pulp mill, the conventional Tomlinson boiler with a steam cycle can deliver only 6.8 MW more power, whereas the hybrid-energy system, thanks to its gas turbine, can deliver 13.3 MW more power. This will make gas turbines an interesting option for pulp mills when biomass and black liquor gasification reaches full-scale commercial breakthrough.

The hybrid energy system is restricted in many ways, since it has to fit in an existing energy system with certain demands. The decision not to use black liquor synthesis gas in the gas turbine also limits the power output of the system. Still, it performs well. In order to significantly increase the power output of a pulp mill, gasification of all the black liquor in an IGCC is needed. For a pulp mill with equal flow of black liquor, 108 t DS/h, a power output of 96.6 MW is reported. This does not include gasification of biomass, as in this study. The total power output for such a combined system would be 122.2 MW.

Reference and Additional Reading

1. Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Pulsation Dampeners

Flow irregularities in fluid flow can cause pulsation in flow that is audible and causes vibration in process machinery. These irregularities can be corrected with pulsation dampeners, which essentially are vessels that provide the fluid (usually this problem is most common with gases) with enough residence time to steady the flow.

A description of typical pulsation dampeners follows.*

Preferable manufacturing facilities are ASME-certified and are registered with the National Board of Boiler and Pressure Vessel Inspectors, with fully accredited "U" and "R" stamps. Specialty services include fabrication with a wide range of materials to pressures in excess of 10,000 psi. Multicylinder dampeners can be built with tolerances of $\pm 1/32$ in.

See Table P-26 and Figs. P-225 through P-232.

Pumps

Pumps are the most common piece of equipment in any plant or process. They vary considerably in operating parameter ranges and scope. Manufacturers can customize a design for any operation with specific requirements, but, for the most

* Source: Peerless, USA.

TABLE P-26 Improved Efficiency Means Lower Costs

Problem/Solution	Discussion	Equipment	Benefits
Pulsation/ vibration	Waves of compression (pulsation) set in motion by the periodic intake and discharge of gas or liquid from reciprocating compressors or pumps can couple with resonant equipment frequencies and produce damaging vibrations, resulting in safety hazards, shortened equipment life, and increased downtime and maintenance/replacement costs.	Pulsation Dampeners	■ Reduce or eliminate equipment downtime/failure. ■ Increase safety: Reduce/ eliminate possibility of rupture caused by equipment vibration fatigue. ■ Lengthen equipment life by eliminating rapidly cycling pressures. ■ Lower compressor piping costs. ■ Increase the accuracy of in-line measurement equipment without vibrations. ■ Save fuel: Less horsepower required to run nonvibrating systems. ■ Simplify and reduce frequency of maintenance. ■ Protect equipment from liquid condensate. ■ Protect machinery from solids.
Solution	The installation of pulsation dampeners or combination units eliminates these problems. Pulsation dampeners attenuate harmful pulsations and reduce damaging vibrations by introducing a series of choking and expansion volumes.		
Dynamic pressure drop	Fluctuating flow due to acoustic resonance can cause significant additional pressure drop and/ or meter error.	Pulsation dampeners	
Solution	The use of proven acoustic control techniques diminishes excessive pulsation by effectively decoupling cylinder excitation from the resonant response of the piping.		
Solid/liquid entrainment	Formation of condensate and the presence of solid particles may cause exchanger fouling and/or compressor cylinder damage.	Combination dampener/ separators	
Solution	The combination dampener/separator effectively removes both solids and liquids in a singgle vessel, providing maximum protection at minimum cost.		

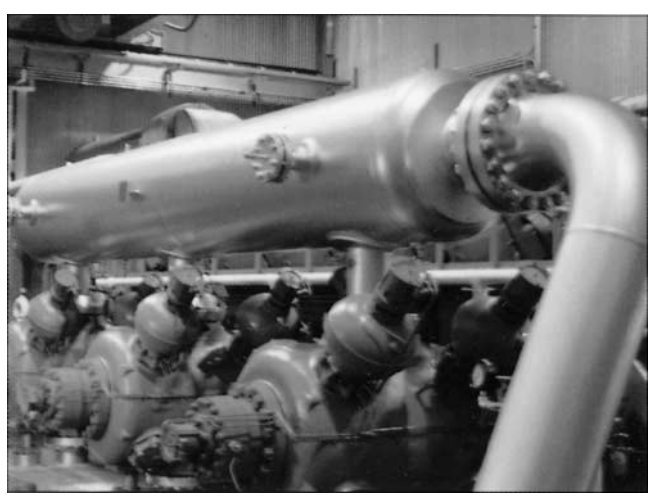


FIG. P-225 Natural gas transmission compressor station is typical of facilities where pulsation dampening equipment provides reliable long-term service. (Source: Peerless.)

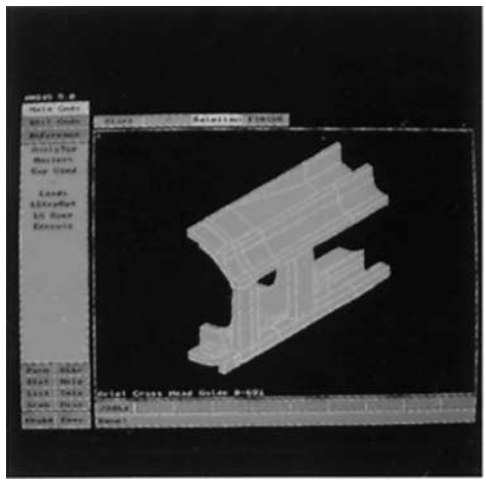


FIG. P-226 General finite-element software is used to analyze structures such as this crosshead guide. (Source: Peerless.)

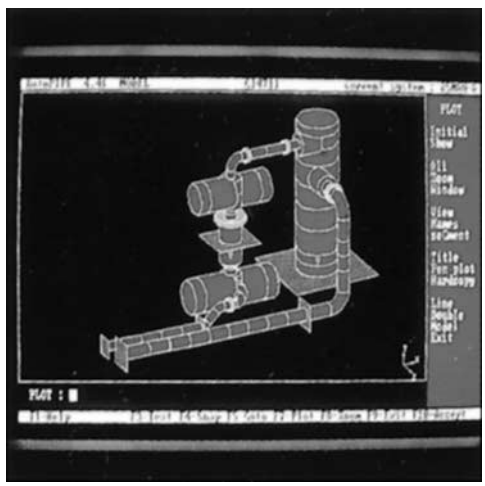


FIG. P-227 Specialized finite-element software is used to perform mechanical analysis of manifolds and complex piping systems. (Source: Peerless.)



FIG. P-228 Welding certified to the highest standards to the use of specified materials is prudent. (Source: Peerless.)

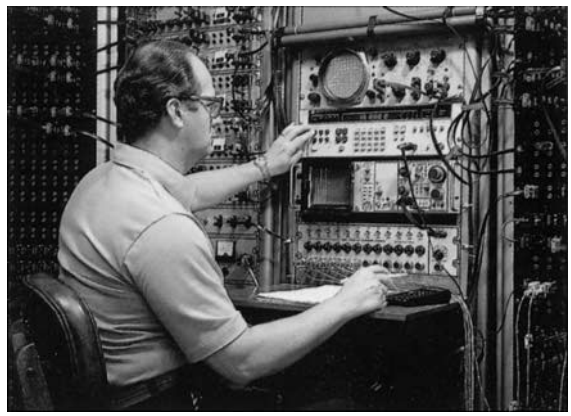


FIG. P-229 Analog simulators have been utilized to perform hundreds of pulsation studies resulting in thousands of successful structural piping designs. (Source: Peerless.)



FIG. P-230 An acoustical and mechanical engineer performs a field survey using state-of-the-art signal analyzers. The company is known for its field service and works closely with customers to identify, solve, and correct vibration problems in compressor and pump installations (Source: Peerless.)



FIG. P-231 Pulsation dampeners in low-pressure hydrogen service. These dampeners, installed in a fluid catalytic cracker (FCC) unit, ensure low refinery maintenance costs. (Source: Peerless.)

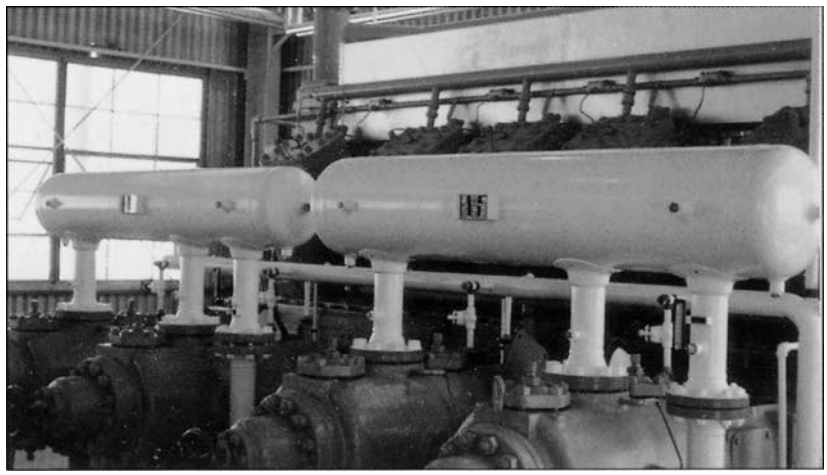


FIG. P-232 The pulsation dampeners are configured for dual nozzle suction manifolds. Installed in a natural gas reciprocating compressor transmission station in the northeastern United States, they reduce damaging vibrations and provide many years of trouble-free service at reduced costs. (Source: Peerless.)

part, picking a pump out of a catalogue will be possible. After that, one has to choose between manufacturers, based on previous experience with items such as after-sales service, longevity of components, time between overhauls, and so forth. There are various types of pumps such as centrifugal, piston, reciprocating, rotary, gear, lobe, and several others.

To help illustrate how varied features and types can be, information on common pump types in the process sector is included. The sources in this section are primarily three different manufacturers, so contrasting operational/design philosophies can be observed.

Pump Theory*

Centrifugal pumps

The centrifugal pump is one of the most versatile types of machinery for industry. Every plant has in operation a multitude of pumps of this type, and modern civilization could not be visualized without this equipment.

Compared with other types of pumps, e.g., reciprocating and rotary pumps, centrifugal pumps operate at relatively high speeds and consequently are smaller and lighter when designed for comparable capacity and head. Required floor space, weight, initial cost, and building costs are therefore reduced.

Owing to their relatively high speed, centrifugal pumps are usually direct-connected to the driver, the majority being electric-motor driven. Having no reciprocating parts, centrifugal pumps are inherently balanced. There are no internal rubbing parts, and because running clearances are relatively large, wear is minimized. The liquid is delivered in a steady stream so that no receiver is needed to even out pulsations.

In contrast to positive-type displacement pumps, centrifugal pumps develop a limited head at constant speed over the operating range from zero to rated capacity, and excessively high pressures cannot occur. They can therefore be started against

* Source: Demag Delaval, USA.

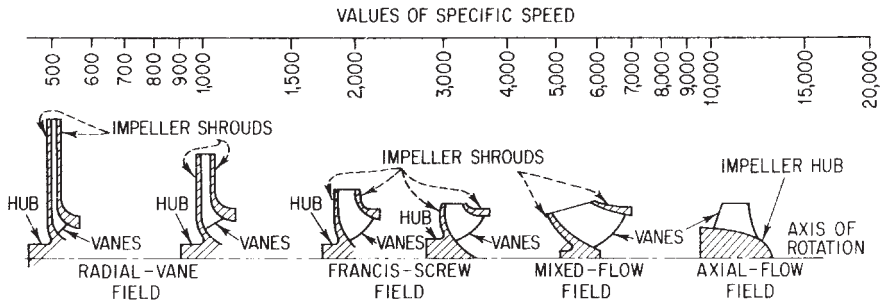


FIG. P-233 Profile of several pump-impeller designs, ranging from the low-specific-speed radial flow on the left to the high-specific-speed impeller design on the right, placed according to where each design fits on the specific-speed scale. (Source: Hydraulic Institute.)

a closed discharge valve but should be operated at this condition for a minimum period (see subsection "Minimum Flow-Through Pump"). Generally, the bearings are located outside the casing, so that the liquid does not come in contact with the lubricating oil and is not contaminated by it.

Classification. There are three general classes of pumps, depending on the configuration of the pump impellers:

- Centrifugal or radial-flow pump
- Mixed-flow pump
- Axial-flow pump

Those classes can be subclassified according to

- Number of stages: single-stage pump, multistage pump
- Arrangement of liquid inlet: single-suction pump, double-suction pump
- Position of shaft: horizontal pump, vertical pump (dry-pit type), vertical pump (submerged type)

Specific speed. Specific speed is a correlation of pump capacity, head, and speed at *optimum* efficiency, which classifies pump *impellers* with respect to their geometric similarity corresponding to the classification mentioned above (see also Fig. P-233).

Specific speed is a number usually expressed as*

$$\text{Specific speed } N_s = \frac{N\sqrt{Q}}{H^{3/4}} \quad \text{or} \quad N_S = \frac{N\sqrt{Q}H^{1/4}}{H}$$

where N_s = specific speed
N = rotative speed, rpm
Q = flow, gal/min, at or near optimum efficiency
H = head, ft per stage

The specific speed of an impeller is defined as the revolutions per minute at which a geometrically similar impeller would run if it were of such a size as to discharge one gallon per minute against one foot head.

*The value of $H^{3/4}$ may be found in Table P-28.

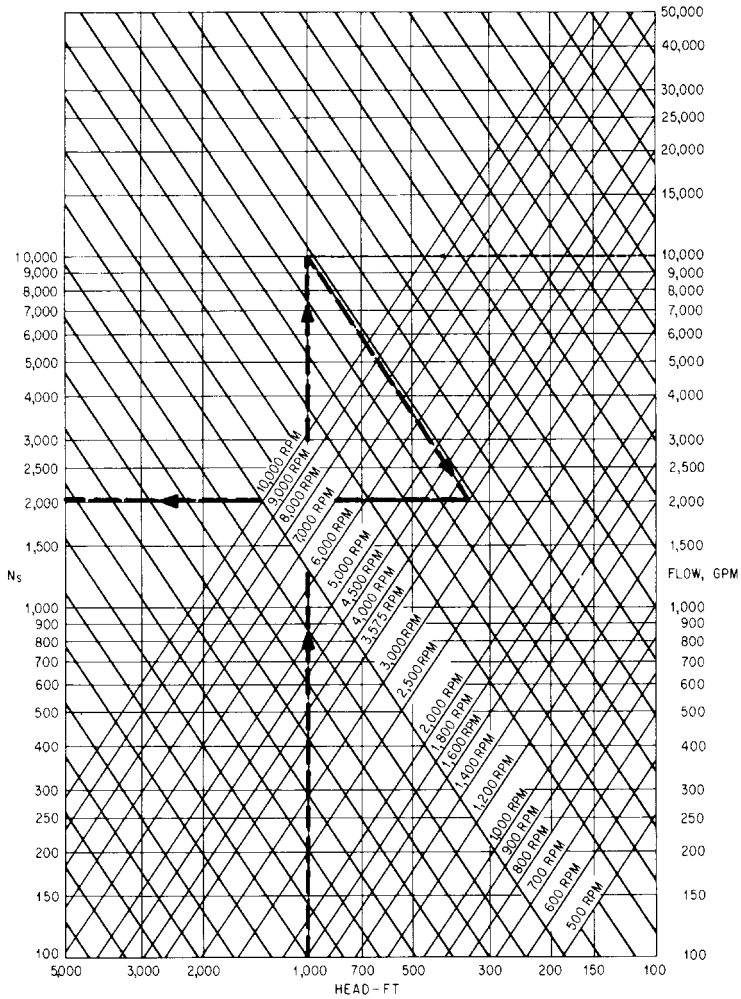


FIG. P-234 Diagram for determination of specific speed. Using the diagram, plot the head-capacity point; move from this point parallel to heavy lines to correct speed; from there move horizontally to the left and read specific speed. Example (dashed lines): H = 1,000 ft; Q = 10,000 gal/min; N = 3575 rpm; N_s = 2015. (Source: Demag Delaval.)

Specific speed is indicative of the shape and characteristics of an impeller, and it has been found that the ratios of major dimensions vary uniformly with specific speed. Specific speed is useful to the designer in predicting required proportions and to the application engineer in checking the suction limitation of pumps.

Impeller form and proportions vary with specific speed, as shown in Fig. P-233.

Pumps are traditionally divided into three classes: the centrifugal or radial-flow, the mixed-flow, and the axial-flow, but it can be seen from Fig. P-233 that there is a continuous change from the radial-flow impeller, which develops pressure principally by the action of centrifugal force, to the axial-flow impeller, which develops most of its head by the propelling or lifting action of the vanes on the liquid.

In the specific-speed range of approximately 1000 to 4000, double-suction impellers are used as frequently as single-suction impellers.

Figure P-233 gives values of $H^{3/4}$ for the accurate determination of the specific speed N_s.

Figure P-234 may be used to find specific speed with sufficient accuracy for practical purposes without calculating the head to the three-fourths power. The point located by plotting the total head and capacity in gallons per minute at the

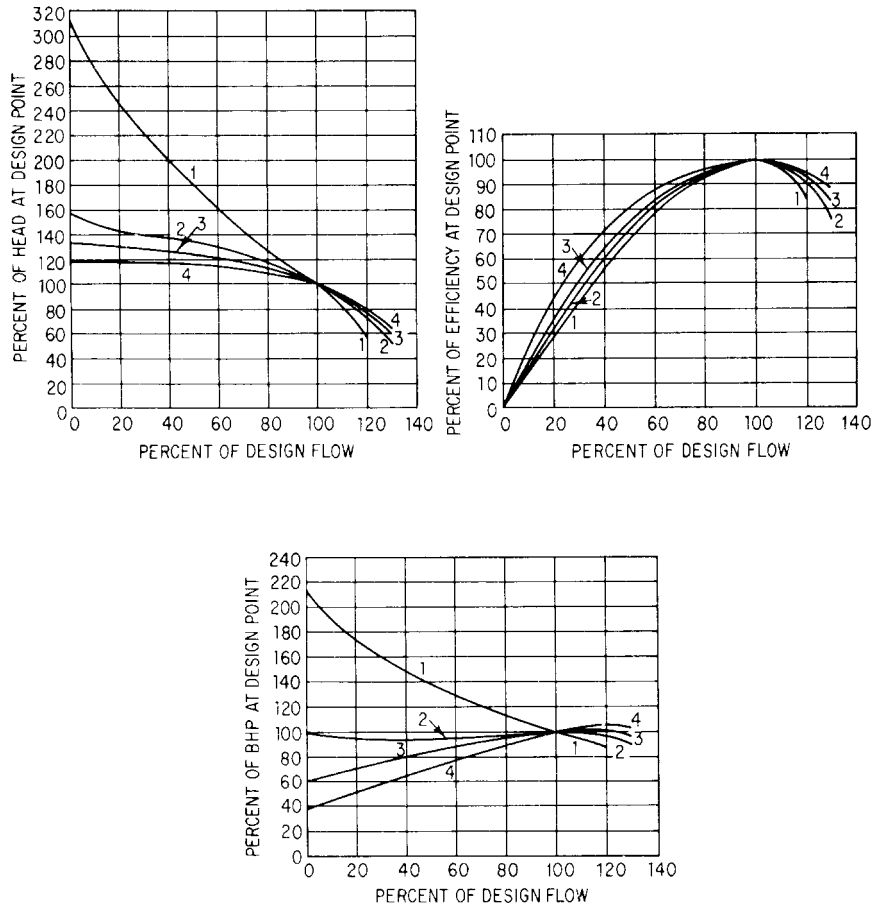


FIG. P-235 Comparison of performance curves for various types of impellers: (1) propeller; (2) mixed-flow; (3) Francis; (4) radial. (Source: Demag Delaval.)

design point is moved parallel to the sloping lines to the correct speed in revolutions per minute. The specific speed is read at the left of the diagram. The procedure is illustrated by the heavy dashed lines.

For double-suction impellers, total flow is used in the calculation, although historically there has been considerable use of half of the flow in the equation.

For multistage pumps, the head per stage is used in the specific-speed equation. Generally, this is the total head of the pump divided by the number of stages.

Impeller performance curves are intimately related to their types or specific speeds. Higher-specific-speed impellers operating at partial loads have higher heads, require more horsepower, and have lower efficiency. This is illustrated in Fig. P-235 on a percentage basis for the three general types mentioned above.

Hydraulics

Definition of static, pressure, and velocity heads. One of the most useful relationships of hydraulics is the *continuity equation*, which is based upon the principle that after steady conditions in any system have been established, the weight flow of fluid per unit of time passing any point is constant. Since most liquids are practically incompressible, this may be put in equation form as $Q = AV$, where Q = flow, ft^3/s; A = cross-sectional area, ft^2; and V = velocity, ft/s. This equation may be rewritten in the form $V = 0.321Q/a$, where V = velocity, ft/s; Q = volume flow, gal/min; and a = area of pipe, in^2. This equation is of importance in determining the velocity of the fluid at various points in either the piping or the pump itself.

Another important term is *head*, which is the energy contained in a pound of fluid. It is the height to which a column of the same fluid must rise to contain the same amount of energy that a unit weight of the fluid has under the conditions considered. It may appear in any of three interchangeable forms.

The *potential head* is based upon the elevation of the fluid above some arbitrarily chosen datum plane. Thus, a column of fluid z ft high contains an amount of energy due to its position and is said to have a head of z ft of fluid.

The *pressure head* is the energy contained in a unit weight of the fluid due to its pressure and equals P/γ, where P is the pressure, lb/ft^2, and γ is its specific weight, lb/ft^3. If an open manometer tube is set perpendicular to the flow, the fluid in it will rise to a height equal to P/γ. This is referred to as *static-pressure head*.

The *kinetic* or *velocity head* is the energy contained in a unit weight of the fluid due to its motion and is given by the familiar expression for kinetic energy $V^2/2g$, where V = velocity, ft/s, and g = acceleration due to gravity (32.17 ft/s^2). Values of this velocity head for various velocities are given in Table P-27. The head may be determined by taking the difference between a reading obtained by a pitot tube facing the flow and a tube at right angles to the flow at the same location.

The total energy of the fluid is equal to the sum of the three heads, or

$$\frac{P}{\gamma} + \frac{V^2}{2g} + z = H$$

Since energy cannot be created or destoyed, H is constant at any point of a closed hydraulic system (losses are neglected). This equation is known as *Bernoulli's theorem*. The various forms of head may vary in magnitude at different sections, but if losses are neglected, their sum is always the same.

When liquid flows through a pipe, there will be a drop in pressure or head due to friction losses.

The *total head H* developed by a centrifugal pump is the measure of energy increase of the liquid imparted to it by the pump and is the difference between the total discharge head and the total suction head. Expressed in equation form,

$$H = h_d - h_s$$

where H = total pump head, ft
h_d = total discharge head, ft, above atmospheric pressure at datum elevation
h_s = total suction head, ft, above atmospheric pressure at datum elevation

Note: h_d and h_s are negative if the corresponding pressures at the datum elevation are below the atmospheric pressure. The common datum is taken through the pump centerline for horizontal pumps and at the entrance eye of the suction impeller for vertical-shaft pumps.

Suction lift h_s exists when the total suction head is below atmospheric pressure. The total suction lift as determined on test is the static pressure (vacuum) as measured by a mercury column expressed in feet of the liquid being pumped less the velocity head at the point of gauge connection. This is equivalent to the static lift plus entrance and friction losses in the piping if the water supply level is below the centerline of the pump. In the case of water supply level above the pump centerline and at atmospheric pressure, suction lift will exist if the entrance and friction losses in the suction piping are greater than the static head.

Suction head h_s exists when the total suction head is above atmospheric pressure. Total suction head as determined on test is the gauge reading expressed in feet of the liquid being pumped plus the velocity head at the point of gauge connection.

TABLE P-27 Values of $H^{3/4}$

Head	$H^{3/4}$	Head	$H^{3/4}$	Head	$H^{3/4}$	Head	$H^{3/4}$
0	0	50	18.8	225	58.1	1,050	184.4
1	1.00	52	19.4	230	59.0	1,100	191.0
2	1.68	54	19.9	235	60.0	1,150	197.4
3	2.28	56	20.5	240	61.0	1,200	203.8
4	2.83	58	21.0	245	62.0	1,250	210.2
5	3.34	60	21.6	250	62.9	1,300	216.4
6	3.83	62	22.1	260	64.8	1,350	222.7
7	4.30	64	22.6	270	66.6	1,400	228.8
8	4.75	66	23.2	280	68.4	1,450	234.9
9	5.20	68	23.7	290	70.2	1,500	241.0
10	5.62	70	24.2	300	72.0	1,550	247.0
11	6.03	72	24.7	310	73.9	1,600	252.9
12	6.45	74	25.2	320	75.7	1,650	258.8
13	6.85	76	25.7	330	77.4	1,700	264.7
14	7.24	78	26.2	340	79.2	1,750	270.5
15	7.62	80	26.8	350	80.9	1,800	276.3
16	8.00	82	27.3	360	82.6	1,850	282.0
17	8.38	84	27.7	370	84.4	1,900	287.7
18	8.73	86	28.2	380	86.1	1,950	293.4
19	9.09	88	28.7	390	87.8	2,000	299.0
20	9.45	90	29.2	400	89.1	2,100	310.2
21	9.80	92	29.7	410	91.1	2,200	321.2
22	10.2	94	30.2	420	92.8	2,300	332.1
23	10.5	96	30.7	430	94.4	2,400	342.8
24	10.8	98	31.1	440	96.1	2,500	353.5
25	11.2	100	31.6	450	97.7	2,600	364.1
26	11.5	105	32.8	460	99.3	2,700	374.5
27	11.8	110	33.9	470	101	2,800	384.9
28	12.2	115	35.0	480	103	2,900	395.1
29	12.5	120	36.2	490	104	3,000	405.3
30	12.8	125	37.4	500	106	3,100	415.4
31	13.1	130	38.5	520	109	3,200	425.4
32	13.5	135	39.6	540	112	3,300	435.3
33	13.8	140	40.6	560	115	3,400	445.2
34	14.1	145	41.8	580	118	3,500	455.0
35	14.4	150	42.8	600	121	3,600	464.7
36	14.7	155	43.9	620	124	3,700	474.4
37	15.0	160	45.0	640	127	3,800	483.9
38	15.3	165	46.0	660	130	3,900	493.5
39	15.6	170	47.1	680	133	4,000	502.9
40	15.9	175	48.1	700	136	4,100	512.3
41	16.2	180	49.2	720	139	4,200	521.7
42	16.5	185	50.2	740	141	4,300	531.0
43	16.8	190	51.2	760	145	4,400	540.2
44	17.1	195	52.2	780	148	4,500	549.4
45	17.4	200	53.2	800	150	4,600	558.5
46	17.7	205	54.2	850	157	4,700	567.6
47	18.0	210	55.1	900	164	4,800	576.6
48	18.2	215	56.2	950	171	4,900	585.6
49	18.5	220	57.1	1,000	178	5,000	594.6

The total suction head is equivalent to the static head less entrance and friction losses in the suction piping.

The *total discharge head* h_d as determined on test is the pressure gauge reading expressed in feet of liquid being pumped plus the velocity head at the point of gauge connection. It is thus equivalent to the static discharge head plus all losses in the discharge piping if the discharge water level is at atmospheric pressure.

In the case of a discharge into a closed vessel under pressure, the total discharge head is equivalent to the static head corresponding to the water level in the vessel,

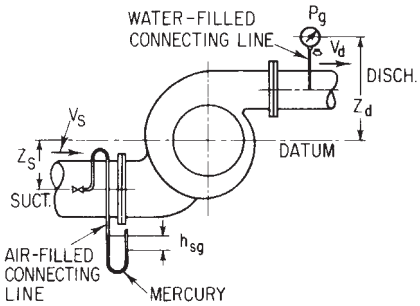


FIG. P-236 Determination of head. (Source: Demag Delaval.)

plus the gauge pressure expressed in feet of liquid corrected to this water level, plus the friction losses in the discharge piping.

If the pump discharges to a level below the pump centerline or into a vessel under vacuum, the static head and pressure are taken as negative.

Determination of head. Figure P-236 shows the usual arrangement for determining the total head developed by a pump working with suction pressure below atmospheric. Suction head is measured by means of a mercury U tube that is connected to the suction through an *air*-filled tube. Discharge head is measured by means of a Bourdon gauge connected to the pump discharge through a *water*-filled tube.

The total head for the arrangement shown in Fig. P-236 is derived from the following expressions:

$$h_s = -\frac{\gamma m}{\gamma} h_{sg} - z_s + \frac{V_s^2}{2g}$$

$$h_d = \frac{144 P_g}{\gamma} + z_d + \frac{V_d^2}{2g}$$

$$H = h_d - h_s$$

$$= \frac{144 P_g}{\gamma} + z_d + \frac{V_d^2}{2g} + \frac{\gamma m}{\gamma} h_{sg} + z_s - \frac{V^2}{2g}$$

$$= \frac{144 P_g}{\gamma} + \frac{\gamma m}{\gamma} h_{sg} + z_d + z_s + \left(\frac{V_d^2}{2g} - \frac{V_s^2}{2g} \right)$$

where H = total pump head, ft
P_g = discharge gauge reading, lb/in^2
h_{sg} = suction gauge reading, ftHg
γ = specific weight of liquid being pumped, lb/ft^3
γm = specific weight of mercury, lb/ft^3
V_d = average velocity in discharge pipe, ft/s
V_s = average velocity in suction pipe, ft/s
z_d and z_s = elevation, ft

For water at 68°F, 1 lb/in^2 = 2.3107 ft, and $(144/\gamma)P_g = 2.3107\ P_g$. The specific gravity of mercury at 68°F is $\gamma m/\gamma = 13.57$. Therefore, for water and mercury at 68°F, the formula simplifies to

$$H = 2.3107 P_g + 13.57 h_{sg} + z_d + z_s + \left(\frac{V_d^2}{2g} - \frac{V_s^2}{2g} \right)$$

Example: Determine the head developed, given a pump with a 10-in suction and 8-in discharge with a capacity of 2000 gal/min. The suction U tube reads 11 inHg and the discharge pressure gauge 50 lb/in^2. The distance from the pump centerline to the U-tube connection is 7 in and to the center of the discharge gauge, +18 in.

Therefore,

$$P_g = 50 \qquad V_d = 12.8\,\text{ft/s}$$

$$h_{sg} = \frac{11}{12}\text{ or } 0.916\text{ ft} \qquad V_s = 8.17\,\text{ft/s}$$

$$z_d = 1.5\text{ ft} \qquad \frac{V_d^2}{2g} = 2.55$$

$$z_s = \frac{7}{12}\text{ or } 0.583\text{ ft} \qquad \frac{V_s^2}{2g} = 1.03$$

Substituting in the formula,

$$H = 2.3107 \times 50 + 13.57 \times 0.916 + 1.5 + 0.583 + (2.55 - 1.03)$$
$$= 115.5 + 12.4 + 1.5 + 0.6 + 1.5 = 131.5\text{ ft}$$

Determination of power required. The work required for pumping depends on the total head and the weight or volume of the liquid to be pumped in a given time. This will give a theoretical liquid horsepower (Lhp) as expressed in the following formula:

$$\text{Lhp} = \frac{wH}{33{,}000}$$

where w = lb liquid pumped per minute
H = total head, ft of liquid

When the liquid is water at 68°F weighing 62.318 lb/ft^3, the formula becomes

$$\text{Lhp} = \frac{\text{gal/min} \times H}{3960}$$

If a liquid other than water is pumped or water at a temperature other than 68°F, the formula must be corrected for the specific gravity of the liquid so that

$$\text{Lhp} = \frac{\text{gal/min} \times H \times \text{sg}}{3960}$$

where sg = specific gravity of the liquid referred to water at 68°F.

If the total head is expressed as pounds per square inch, the specific gravity does not enter into the final equation, which is as follows:

$$\text{Lhp} = \frac{\text{gal/min} \times (\text{total head, lb/in}^2)}{1714}$$

The theoretical horsepower is less than the actual or brake horsepower because of losses in the pump such as friction and leakage. The efficiency of the pump is therefore the ratio of the liquid-horsepower output to the brake-horsepower (bhp) input, or

$$\text{Efficiency, percent} = \frac{\text{Lhp}}{\text{bhp}} \times 100$$

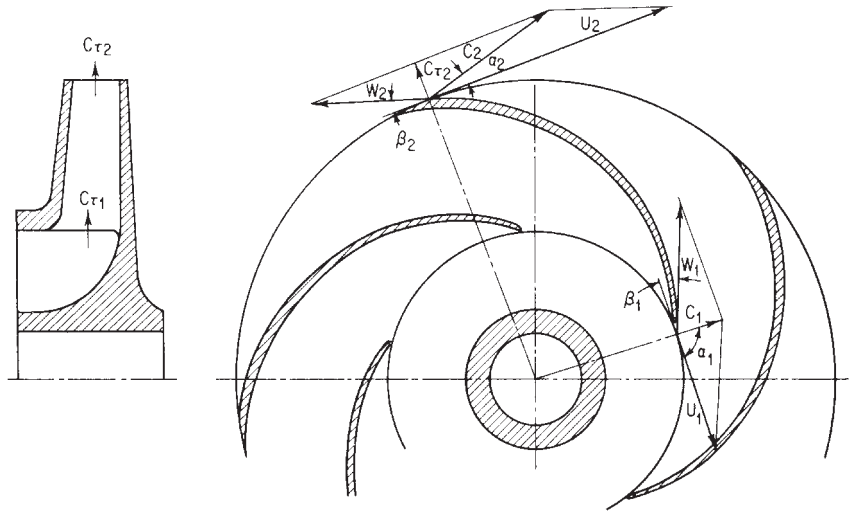


FIG. P-237 Wheel geometry. (Source: Demag Delaval.)

and the bhp required at the coupling will be

$$\text{bhp} = \frac{\text{Lhp}}{\eta}$$

where η = pump efficiency, percent/100.

If the entire unit is considered, it is necessary to include the efficiency of the driver and any other associated equipment to arrive at the overall efficiency.

Pump performance. When studying the flow of liquid through a pump impeller, three types of velocities must be considered. The first is the velocity of a point on the impeller and is designated by the symbol U. The second is the velocity of the fluid relative to the casing, known as the absolute velocity and designated by the symbol C. The third is the velocity of the liquid relative to the impeller, known as the relative velocity and designated by the symbol W. The relative velocity W is found by taking the vector difference of the absolute velocities U and C. Subscripts are generally used with these symbols, the subscript 1 designating the velocities at the inlet to the vanes and the subscript 2 at the outlet of the vanes. The angle between the vectors **U** and **W** is designated as β and is the angle that the vane makes with a tangent to the impeller. The angle α between vectors **C** and **U** represents the angle at which the fluid enters or leaves the wheel. This is illustrated in Fig. P-237 for a typical impeller.

The ideal head developed by the wheel is given by the equation

$$H_i = \frac{U_2^2 - U_1^2}{2g} + \frac{W_1^2 - W_2^2}{2g} + \frac{C_2^2 - C_1^2}{2g}$$

where the first term represents the head due to the centrifugal action, the second that due to the change in the relative velocity, and the third that due to the change in the absolute velocity. The first two terms represent the pressure head that is developed in the impeller, while the last is the velocity head developed in the impeller and converted into pressure in the volute or diffuser. This expression may also be written

$$H_i = \frac{1}{g}(U_2 C_2 \cos\alpha_2 - U_1 C_1 \cos\alpha_1)$$

The actual head developed by the pump will be less than the ideal owing to fluid friction and shock losses and to a circulatory flow that takes place between the vanes.

The basic formula can be expressed in a simplified form for the actual head developed by a pump impeller at rated design conditions as follows:

$$H = \mu \frac{U_2^2}{g}$$

where H = actual head, ft
U_2 = impeller-tip velocity, ft/s
g = acceleration of gravity, ft/s^2 (32.17 ft/s^2)
μ = pressure coefficient (varying from 0.45 to 0.52 for radial-type impellers)

From this it follows that the impeller diameter will be

$$D_2 = \frac{1300}{n}\sqrt{\frac{H}{\mu}}$$

where D_2 = impeller diameter, in
n = pump speed, rpm

The above equations give the basis for estimating the effect of speed and impeller changes on the head. The weight or volume flow is directly proportional to the revolutions per minute of the pump. It may be observed from the above equations that the head is proportional to the square of the rpm. The horsepower required to drive the pump is the product of the head and weight flow divided by a constant. The efficiency of a pump is little affected by reasonable changes in either the speed or the impeller diameter.

The net effect of changes in the outside diameter of the impeller is similar to that of varying the speed of the unit.

The effect of changes in operating conditions may be summarized by the following equations, where the subscript a refers to the condition while b refers to the new.

$$\text{Volume or weight flow } Q_b = Q\frac{n_b}{n_a}\frac{D_b}{D_a}$$

$$\text{Head } H_b = H_a\left(\frac{n_b}{n_a}\right)^2\left(\frac{D_b}{D_a}\right)^2$$

$$\text{Horsepower } \mathrm{hp}_b = \mathrm{hp}_a\left(\frac{n_b}{n_a}\right)^3\left(\frac{D_b}{D_a}\right)^3$$

$$\text{Efficiency } \eta_b = \eta_a$$

where n = speed, rpm
D = impeller outside diameter, in

These relations may be applied over the entire range of a pump characteristic curve but should be used only for relatively small changes in speed or impeller diameter.

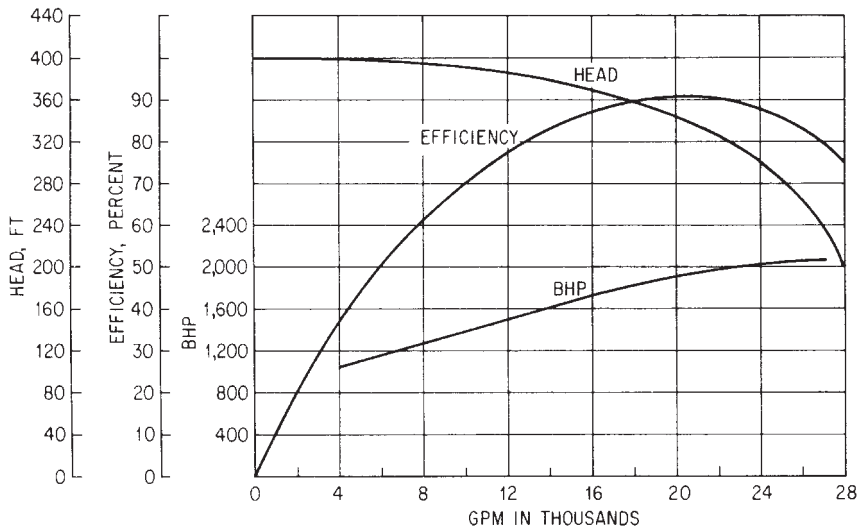


FIG. P-238 Performance curve for a 24/20 centrifugal pump at 885 rpm. (Source: Demag Delaval.)

Example: Assume that a pump is delivering 2500 gal/min of water against a head of 150 ft when running 1760 rpm with an efficiency of 81 percent. The bhp is then 117. The outside diameter of the impeller is $13^1/_2$ in. What would the performance be if the impeller diameter were reduced to 13 in and the pump speeded up to 1800 rpm?

$$Q_b = Q\frac{n_b D_b}{n_a D_a} = 2{,}500\,\frac{1{,}800}{1{,}760}\,\frac{13}{13.5} = 2{,}460\,\text{gal/min}$$

$$H_b = H_a\left(\frac{n_b}{n_a}\right)^2\left(\frac{D_b}{D_a}\right)^2 = 150\left(\frac{1{,}800}{1{,}760}\right)^2\left(\frac{13}{13.5}\right)^2 = 146.7\,\text{ft}$$

$$\text{hp}_b = \text{hp}_a\left(\frac{n_b}{n_a}\right)^3\left(\frac{D_b}{D_a}\right)^3 = 117\left(\frac{1{,}800}{1{,}760}\right)^3\left(\frac{13}{13.5}\right)^3 = 112\,\text{hp}$$

$$\eta_b = \eta_a = 81\ \text{percent}$$

Figure P-238 shows a typical performance curve for a centrifugal pump. It illustrates the variations of head, power, and efficiency as a function of capacity at constant speed. These curves are called *characteristic curves* of pump performance, and they are determined experimentally for each pump type.

Figure P-239 shows diagrammatically the characteristic curves of a pump at two speeds N_1 and N_2, where points of similar flow conditions are related according to the formulas given above.

Parallel and series operation. When pumping requirements are variable, it may be desirable to install several small pumps in parallel rather than use a single large pump. When the demand drops, one or more smaller pumps may be shut down, thus allowing the remainder to operate at or near peak efficiency. If a single pump is used with lowered demand, the discharge must be throttled (for constant speed), and it will operate at reduced efficiency. Moreover, when smaller units are used, opportunity is provided during slack-demand periods for repairing and maintaining each pump in turn, thus avoiding the plant shutdowns that would be necessary

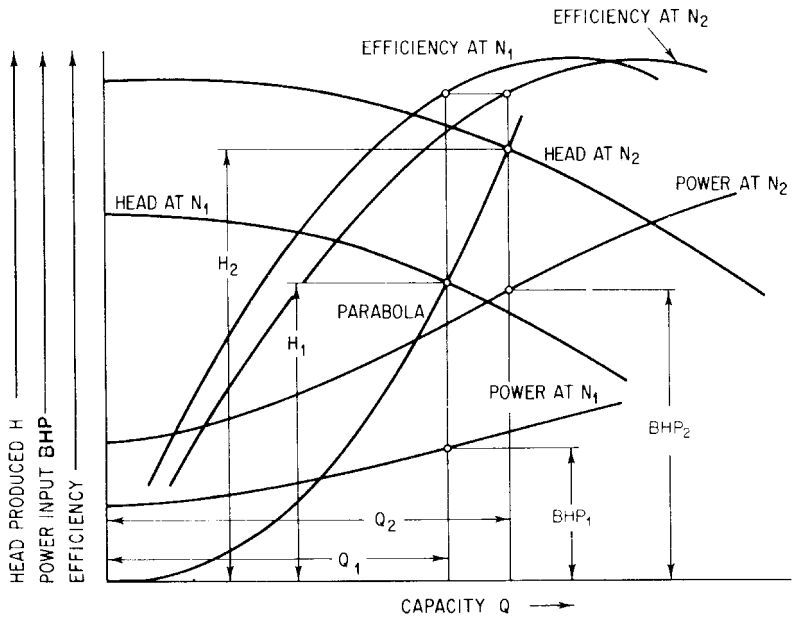


FIG. P-239 Characteristic curves at two speeds N_1 and N_2. (Source: Demag Delaval.)

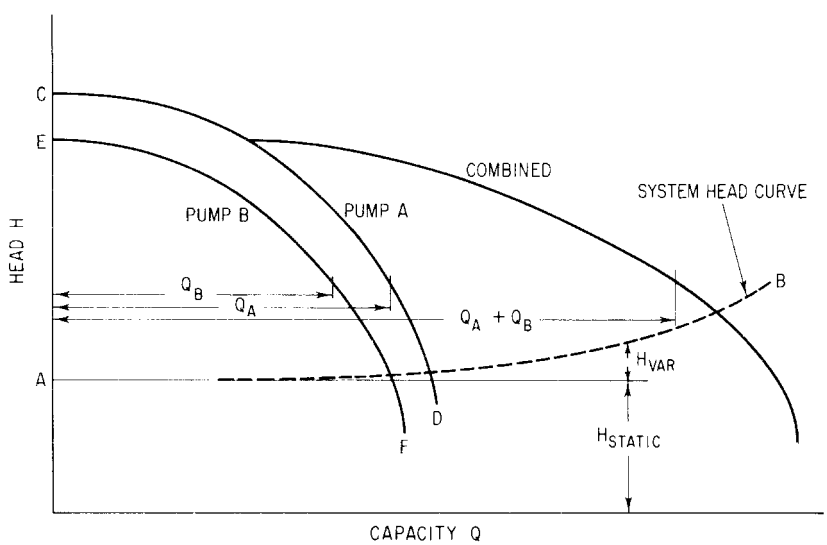


FIG. P-240 Head-capacity curves of pumps operating in parallel. (Source: Demag Delaval.)

with single units. Similarly, multiple pumps in series may be used when liquid must be delivered at high heads.

In planning such installations, a head-capacity curve for the system must first be drawn. The head required by the system is the sum of the static head (difference in elevation and/or its pressure equivalent) plus the variable head (friction and shock losses in the pipes, heaters, etc.). The former is usually constant for a given system, whereas the latter increase approximately with the square of the flow. The resulting curve is represented as line *AB* in Figs. P-240 and P-241.

Connecting two pumps in parallel to be driven by one motor is not a very common practice, and offhand such an arrangement may appear more expensive than a

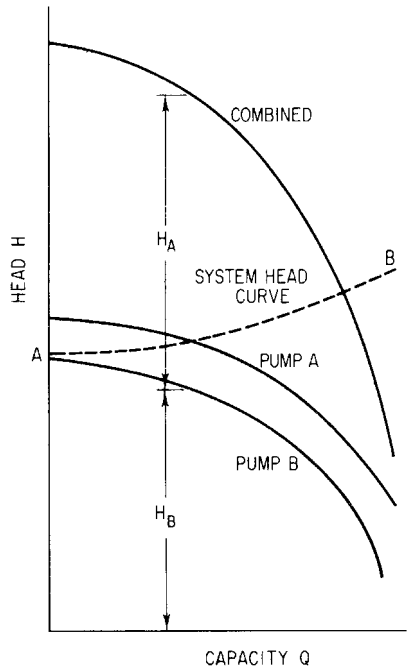


FIG. P-241 Head-capacity curves of pumps operating in series. (Source: Demag Delaval.)

single pump. However, it should be remembered that in most cases it is possible to operate such a unit at about 40 percent higher speed, which may reduce the cost of the motor materially. Thus, the cost of two high-speed pumps may not be much greater than that of a single slow-speed pump.

For units to operate satisfactorily in parallel, they must be working on the portion of the characteristic curve that drops off with increased capacity in order to secure an even flow distribution. Consider the action of two pumps operating in parallel. The system head-capacity curve *AB* shown in Fig. P-240 starts at *H* static when the flow is zero and rises parabolically with increased flow. Curve *CD* represents the characteristic curve of pump *A* operating alone; the similar curve for pump *B* is represented by *EF*. Pump *B* will not start delivery until the discharge pressure of pump *A* falls below that of the shutoff head of *B* (point *E*). The combined delivery for a given head is equal to the sum of the individual capacities of the two pumps at that head. For a given combined delivery head, the capacity is divided between the pumps and designated as Q_A and Q_B. The combined characteristic curve shown on the figure is found by plotting these summations. The combined bhp curve can be found by adding the bhp of pump *A* corresponding to Q_A to that of pump *B* corresponding to Q_B and by plotting this at the combined flow. The efficiency curve of the combination may be determined by dividing the combined power $\gamma(Q_A + Q_B)H/550$ by the corresponding combined bhp (Q taken as ft^3/s).

If two pumps are operated in *series*, the combined head for any flow is equal to the sum of the individual heads as shown in Fig. P-241. The combined bhp curve may be found by adding the horsepowers given by the curves for the individual pumps. Points on the combined efficiency curve are found by dividing the combined fluid horsepower $(H_A + H_B)Q\gamma/550$ by the combined bhp; Q is again in ft^3/s.

Pump performance for viscous liquids. As the viscosity of the liquid being handled by a centrifugal pump is increased, the effect on the performance is a marked increase in the bhp, a reduction in the head, and some reduction in the capacity.

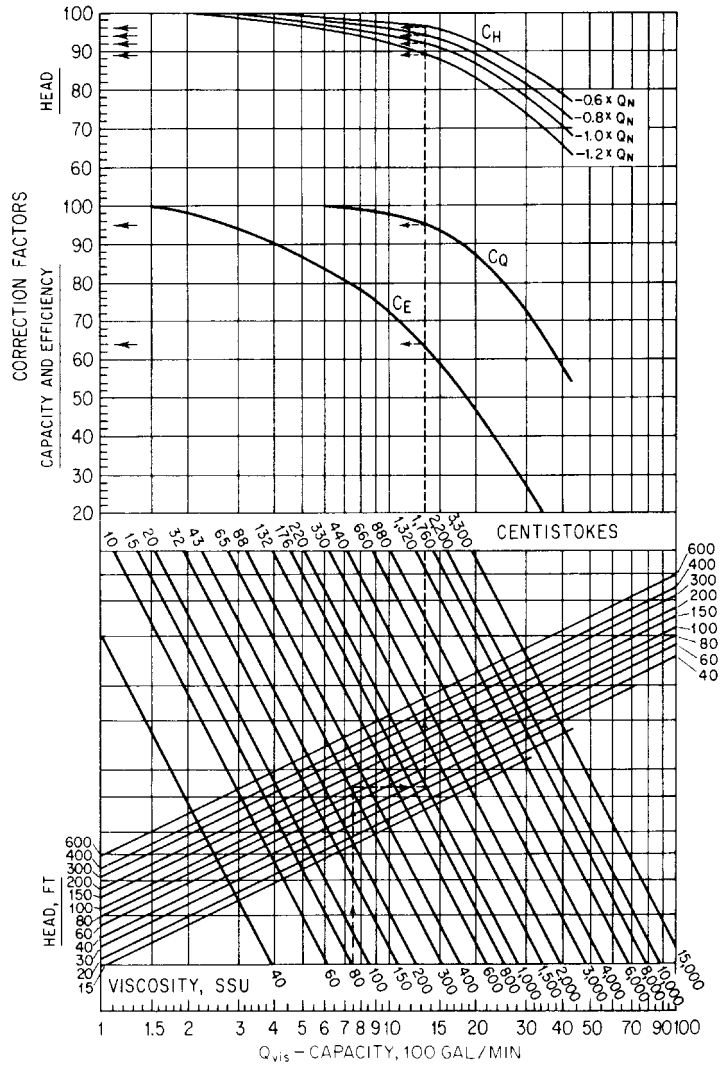


FIG. P-242 Performance correction diagram. (Source: Hydraulic Institute.)

Figure P-242 is taken from the *Hydraulic Institute Standards* and may be used to estimate the magnitude of these effects for a particular liquid. The diagram should be used only for the conventional radial-type impeller and should not be extrapolated or used for nonuniform liquids such as gels, slurries, or paper stock or when the net positive suction head is inadequate.

The procedure for selecting a pump for a given head-capacity-viscosity requirement when the desired capacity and head of the viscous liquid and the viscosity and specific gravity at the pumping temperature are known is as follows. Enter the diagram at the base with the desired viscous capacity Q_{vis} and proceed upward to the desired viscous head H_{vis} in feet of liquid. For multistage pumps use the head per stage. Proceed horizontally (either left or right) to the fluid viscosity, and then go upward to the correction curves. Divide the viscous capacity Q_{vis} by the capacity correction factor C_Q to get the approximate equivalent water capacity Q_w. Divide the viscous head H_{vis} by the head correction factor C_h from the curve labeled "$1.0 \times Q_N$" to get the approximate equivalent water head H_w. Having this new equivalent water head-capacity point, select a pump in the usual manner. The viscous efficiency and the viscous bhp may then be calculated.

To illustrate, assume that it is desired to select a pump to deliver 750 gal/min at 100 ft total head of a liquid having a viscosity of 1000 SSU and a specific gravity of 0.90 at the pumping temperature. Enter the diagram at 750 gal/min, go up to 100 ft head, over to 1000 SSU, and then up to the correction factors:

$$C_Q = 0.95 \qquad Q_w = \frac{Q_{\text{vis}}}{C_Q} = \frac{750}{0.95} = 790 \text{ gal/min}$$

$$C_H = 0.92 \text{ for } Q_N = 1.0 \quad H_W \frac{H_{\text{vis}}}{C_H} = \frac{100}{0.92} = 108.8 \text{ ft head, or roughly } 109 \text{ ft}$$

Select a pump for a water capacity of 790 gal/min at 109 ft head. The selection should be at or close to the maximum efficiency point for water performance. If the pump selected has an efficiency with water of 81 percent at 790 gal/min, then the efficiency for the viscous liquid will be

$$E_{\text{vis}} = C_E \times E_W = 0.635 \times 81 \text{ percent} = 51.5 \text{ percent}$$

The bhp for pumping the viscous liquid will be

$$\text{bhp}_{\text{vis}} = \frac{Q_{\text{vis}} \times H_{\text{vis}} \times \text{sg}}{3{,}960 \times E_{\text{vis}}} = \frac{750 \times 100 \times 0.90}{3{,}960 \times 0.515} = 33.1 \text{ hp}$$

The procedure to determine the pump performance on a viscous liquid when its performance with water is known is as follows. From the efficiency curve locate the water capacity ($1.0 \times Q_N$) at which maximum efficiency is obtained. From this capacity determine the capacities $0.6 \times Q_N$, $0.8 \times Q_N$, and $1.2 \times Q_N$. Enter the diagram at the bottom with the capacity at best efficiency ($1.0 \times Q_N$), go upward to the head developed (in one stage) H_w at this capacity, then horizontally (either left or right) to the desired viscosity, and then proceed upward to the various correction curves. Read the values of C_E and C_Q and of C_H for all four capacities. Multiply each capacity by C_Q to obtain the corrected capacities. Multiply each head by its corresponding head correction factor to obtain the corrected heads. Multiply each efficiency value by C_E to obtain the corrected efficiency values, which apply at the corresponding corrected capacities. The head at shutoff can be taken as approximately the same as that for water. Calculate the viscous bhp from the equation

$$\text{bhp}_{\text{vis}} = \frac{Q_{\text{vis}} \times H_{\text{vis}} \times \text{sg}}{3{,}960 \times E_{\text{vis}}}$$

Net positive suction head. If the pressure at any point inside a pump drops below the vapor pressure corresponding to the temperature of the liquid, the liquid will vaporize. These bubbles of vapor will be carried along to a point of higher pressure where they will suddenly collapse. This phenomenon is known as *cavitation*. It is accompanied by removal of metal in the pump, reduced flow, loss in efficiency, and noise and hence should be avoided. It occurs around the pump suction and inlet edge of the vanes when the absolute suction pressure is low.

The net positive suction head (NPSH) of a pump is the equivalent total head at the pump centerline corrected for vapor pressure. It is found from the equation

$$\text{NPSH} = H_p = H_z - H_{vp} - H_f$$

H_p is the head corresponding to the absolute pressure on the surface of the liquid from which the pump draws. This will be the barometric pressure if the tank is

open to the atmosphere or the absolute pressure in the closed tank or condenser from which the pump takes liquid. In a deaerating heater, it is normally the saturated pressure at the existing water temperature. H_z is the height in feet of the fluid surface above or below the pump centerline. If above, it is considered to be plus since the suction head is then increased; if below, it is minus. H_{vp} is the head corresponding to the vapor pressure at the existing temperature of the liquid. H_f is the head lost because of friction and turbulence between the surface of the liquid and the pump suction flange.

In designing a pump installation and purchasing a pump, there are two types of NPSH to be considered. One is the *available* NPSH of the system, and the other is the *required* NPSH of the pump to be placed in the system. The former is determined by the plant designer and is based upon the pump location, fluid temperature, etc., while the latter is based upon suppression pump tests of the manufacturer. To secure satisfactory operating conditions, the available NPSH must be greater than the required NPSH. In higher-energy pumps such as boiler-feed pumps, values of NPSHA should be 1.5 to 2.0 times larger than NPSHR (as normally measured with a 3 percent drop in head).

The calculation of available NPSH will be illustrated by two examples. The head corresponding to a given pressure is given by the equation $H_p = 2.31\,p/\text{sg}$, where p is the pressure in pounds per square inch and sg the specific gravity of the liquid.

Assume that water at 80°F is to be pumped from a sump. The unit is located at an altitude of 800 ft above sea level, and the suction lift (from water surface to pump centerline) is 7 ft. The pipe losses amount to 1 ft head. What is the available NPSH?

The atmospheric pressure at an altitude of 800 ft is 14.27 psia. The specific gravity of the water at 80°F is 0.9984, and the vapor pressure is 0.5069 psia.

$$H_p = \frac{2.31p}{\text{sg}} = \frac{2.31 \times 14.27}{0.9984} = 32.97 \text{ ft}$$

$$H_z = -7 \text{ ft (negative since it is a lift)}$$

$$H_{vp} = \frac{2.31p}{\text{sg}} = \frac{2.31 \times 0.5069}{0.9984} = 1.17 \text{ ft}$$

$$H_f = 1 \text{ ft}$$

$$\text{NPSH} = H_p + H_z - H_{vp} - H_f = 32.97 - 7 - 1.17 - 1 = 23.80 \text{ ft}$$

Determine the available NPSH of a condensate pump drawing water from a condenser in which a 28-in vacuum, referred to a 30-in barometer, is maintained. The friction and turbulence head loss in the piping is estimated to be 2 ft. The minimum height of water in the condenser above the pump centerline is 5 ft. The absolute pressure in the condenser is 30 − 28 = 2 inHg, or 0.982 lb/in^2. The corresponding specific gravity is 0.9945.

$$H_p = H_{vp} = \frac{2.31p}{\text{sg}} = \frac{2.31 \times 0.982}{0.9945} = 2.28 \text{ ft}$$

$$H_f = 1 \text{ ft} \qquad H_z = 5 \text{ ft}$$

$$\text{NPSH} = H_p + H_z - H_{vp} - H_f = 2.28 + 5 - 2.28 - 2 = 3 \text{ ft}$$

A third example is that of a deaerating heater having a water level 180 ft above the pump centerline. The water temperature is 350°F. The pipe friction loss is

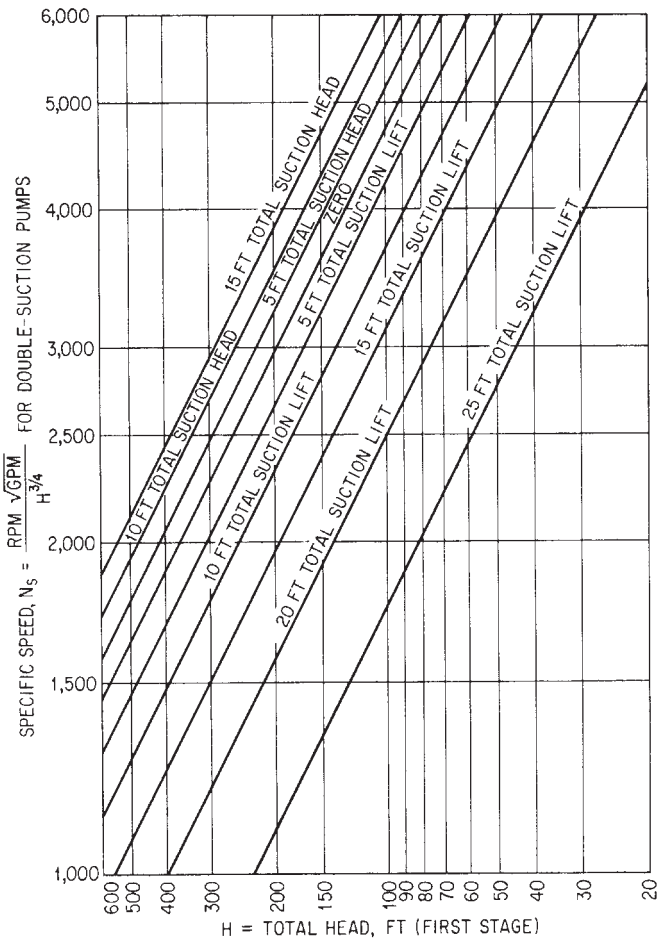


FIG. P-243 Upper limits of specific speeds: double-suction pumps handling clear water at 85°F at sea level. (Source: Hydraulic Institute.)

12 ft. Since the water in the deareator is in a saturated condition, this absolute pressure on the surface liquid equals the vapor pressure at 350°F. Then

$$\text{NPSH} = H_p + H_z - H_{vp} - H_f$$
$$= \frac{2.31 \times 134.6}{0.892} + 180 - \frac{2.31 \times 134.6}{0.892} - 12 = 168 \text{ ft}$$

The required NPSH must be determined in most cases by means of suppression tests. The Hydraulic Institute has prepared a series of diagrams to estimate this required head (see Figs. P-243 to P-246 inclusive). These diagrams are not to be considered as the highest values that can be obtained by careful design, but they may be used for estimating as they represent average results of good present-day practice.

The use of the diagrams is simple and may be illustrated by an example. A double-suction pump operating at 3600 rpm delivers 1000 gal/min against a total head of 200 ft. What should the minimum NPSH be for satisfactory operation? The specific speed as found from Fig. P-234 is 2200. By referring to Fig. P-243, the point corresponding to this specific speed for double-suction pumps and a total dynamic head of 200 ft gives a 12-ft suction lift as the safe maximum. If the same conditions

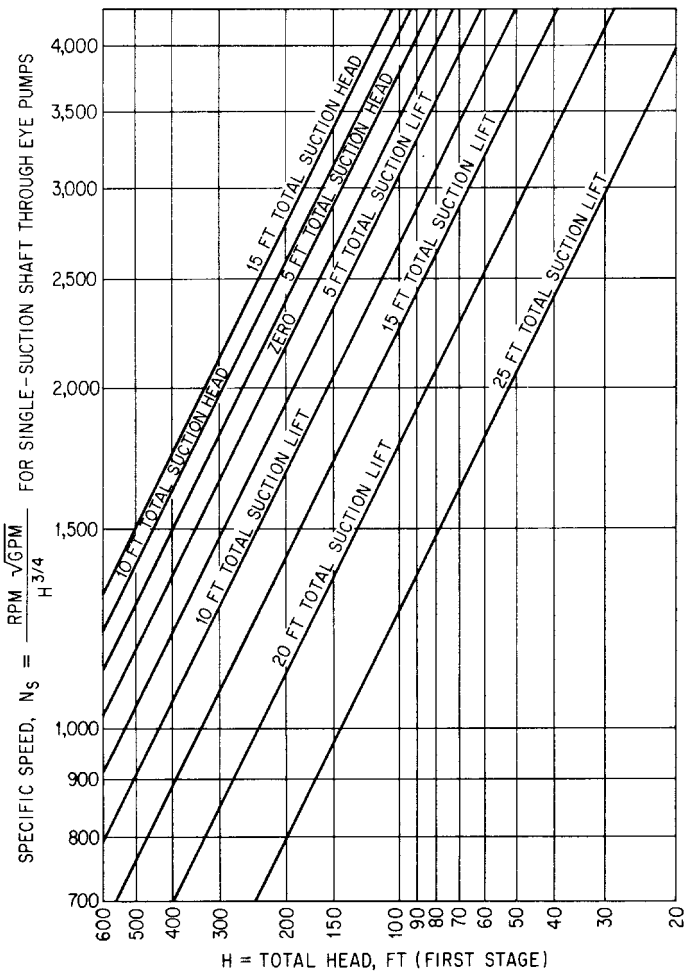


FIG. P-244 Upper limits of specific speeds: single-suction shaft through eye pumps handling clear water at 85°F at sea level. (Source: Hydraulic Institute.)

were applied to a single-suction pump with the shaft through the eye of the impeller, the safe minimum suction condition would require at least a 1 ft positive head (i.e., the suction head would have to be at least +1 ft rather than –12 ft; hence, the required NPSH would be 35 ft instead of 22 ft).

These curves are based upon handling clear water at 85°F and sea-level barometric pressure. If the water temperature is higher, the difference in head corresponding to the difference in vapor pressures between 85°F and the temperature of the water pumped should be subtracted from the suction lift or added to the suction head. Also, if the unit is to be located above sea level, the difference in head corresponding to the difference in atmospheric pressures should be subtracted from the suction lift or added to the suction head.

Thus, in the above example, if the water temperature is 140°F and the plant is located at an altitude of 2000 ft, the correction for vapor pressure will be 2.889 – 0.596 = 2.293 lb/in^2, and the correction for altitude will be 14.69 – 13.66 = 1.03 lb/in^2. The corresponding head change will be 2.31 (2.293 + 1.03)/0.9850 = 7.8 ft. For the double-suction pump the maximum suction lift would be 12.0 – 7.8 = 4.2 ft, and for the single-suction pump the positive suction head would have to be 1.0 + 7.8 = 8.8 ft.

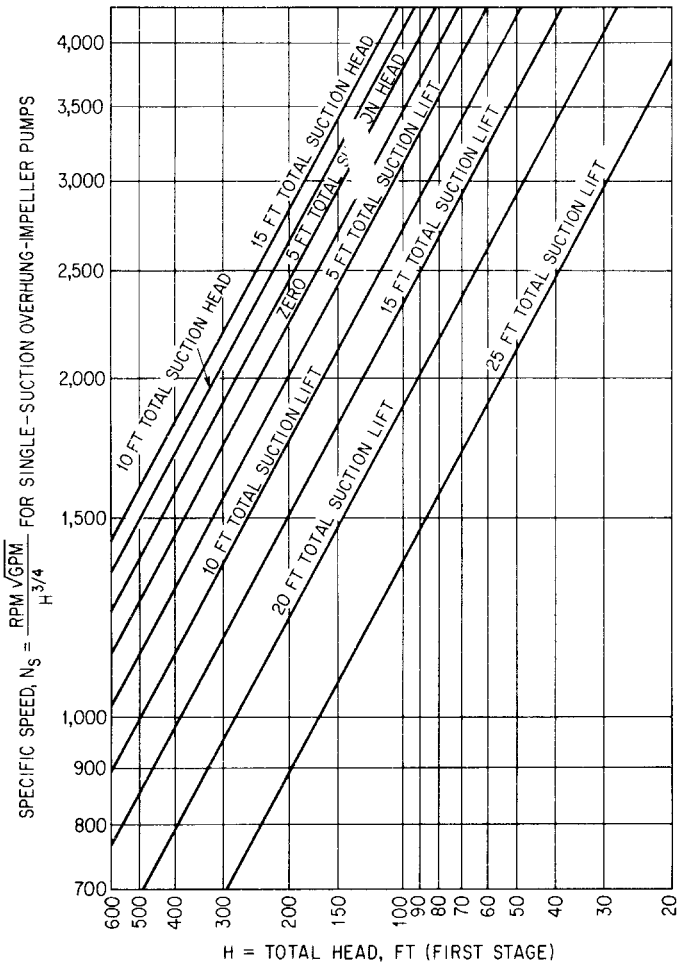


FIG. P-245 Upper limits of specific speeds: single-suction overhung impeller pumps handling clear water at 85°F at sea level. (Source: Hydraulic Institute.)

A series of diagrams (Figs. P-247 through P-249) have been prepared by the Hydraulic Institute to determine NPSH on the basis of the flow, operating speed, and discharge pressure for hot-water and condensate pumps. They may also be used to find the maximum permissible flow for a given available NPSH.

Hot water. Two curves, Figs. P-247 and P-248, have been prepared for pumps handling hot water at temperatures of 212°F and above. These curves show the recommended minimum NPSH in feet for different design capacities and speeds.

Figure P-247 applies to single-suction pumps and Fig. P-248 to double-suction pumps. These curves serve as guides in determining the NPSH for hot-water pumps and do not necessarily represent absolute minimum values.

Net positive suction head for condensate pumps. Figure P-249 indicates NPSH for condensate pumps with the shaft passing through the eye of the impeller. It applies to pumps having a maximum of three stages, the lower scale representing single-suction pumps and the upper scale double-suction pumps or pumps with a double-suction first-stage impeller.

For single-suction overhung impellers the curve may be used by dividing the specified capacity, if 400 gal/min or less, by 1.2, and if greater than 400 gal/min, by 1.15.

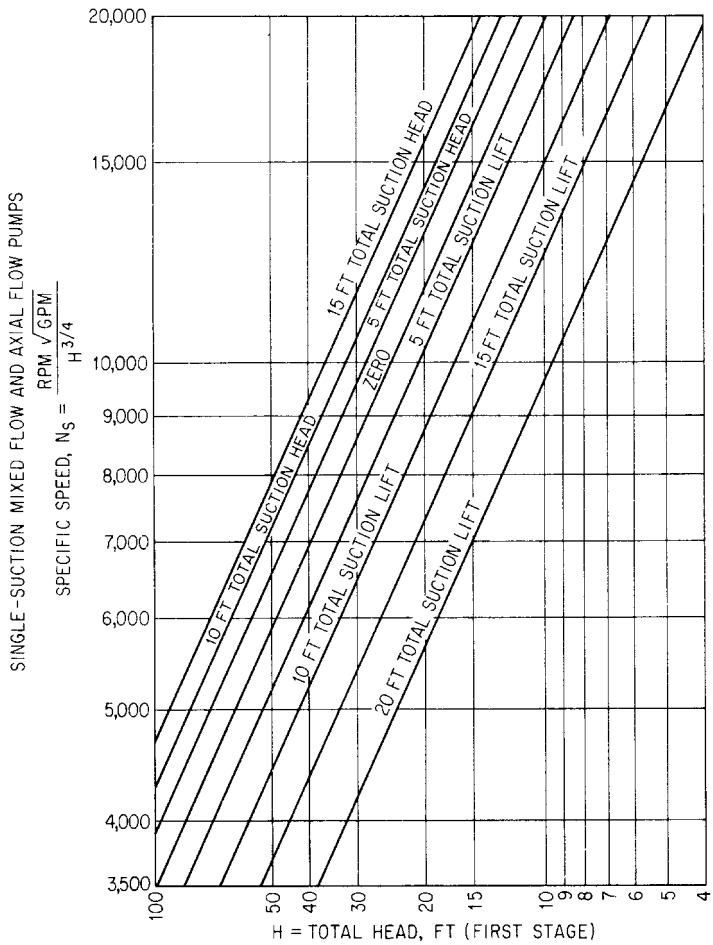


FIG. P-246 Upper limits of specific speeds: single-suction mixed- and axial-flow pumps handling clear water at 85°F at sea level. (Source: Hydraulic Institute.)

The curve may be used for capacities and speeds other than shown by the relation that, for a definite NPSH, the product of rpm $\times \sqrt{\text{gal/min}}$ remains constant.

Minimum flow-through pump. The difference between the power put into a centrifugal pump and the useful power performed by the pump, or the difference between the brake and water horsepowers, is converted into heat, most of which appears as increased temperature of the water. The bhp curve of a pump rated at 500 gal/min against a 2600-ft head and the corresponding water-horsepower curve are shown in Fig. P-250. If we neglect bearing losses, which are minor, the difference between these curves at any capacity represents the horsepower absorbed by the water in the form of heat. Multiplying these differences by 42.4 gives the Btu generated in the pump per minute. Dividing these values by the flow in pounds per minute gives the temperature rise at each capacity. This curve is plotted in Fig. P-250.

This temperature rise is generally not important in single-stage pumps, particularly if they are handling cold water, but for pumps handling hot liquids, such as boiler-feed pumps, it may become a serious matter. Then the resulting rapid temperature rise may cause the internal rotating parts to expand more rapidly than the heavier encircling parts so that severe rubbing may occur, or the impeller may even

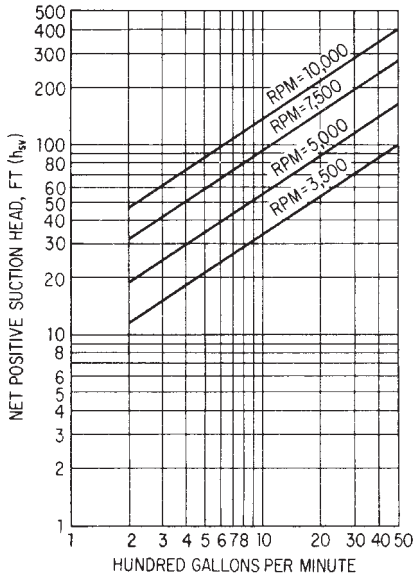


FIG. P-247 Net positive suction head for centrifugal hot-water pumps, single suction. (Source: Hydraulic Institute.)

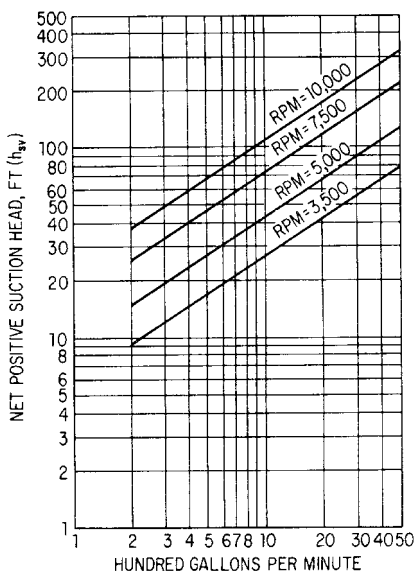


FIG. P-248 Net positive suction head for centrifugal hot-water pumps, double suction, first stage. (Source: Hydraulic Institute.)

become loose on the shaft. Also the temperature of the water may rise to a point at which the water flashes into steam, causing the pump to become vapor bound.

The temperature rise in a pump may be calculated from the formula

$$\Delta t = \frac{(1-\eta)H}{778c\eta}$$

where Δt = temperature rise, °F
η = overall pump efficiency, expressed as a decimal
c = specific heat of fluid being pumped (equals 1.0 for water)
H = total head of pump, ft

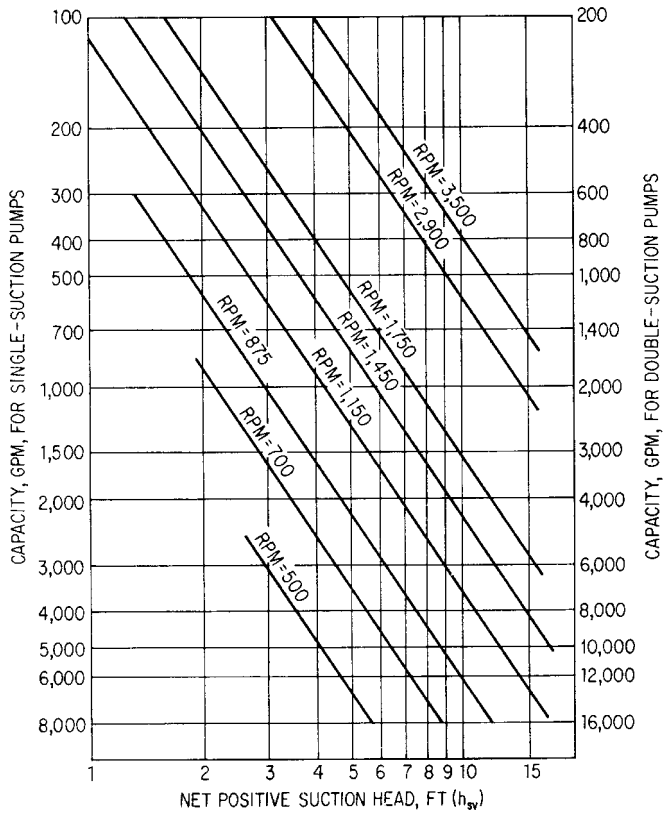


FIG. P-249 Capacity and speed limitations for condensate pumps with the shaft through the eye of the impeller. (Source: Hydraulic Institute.)

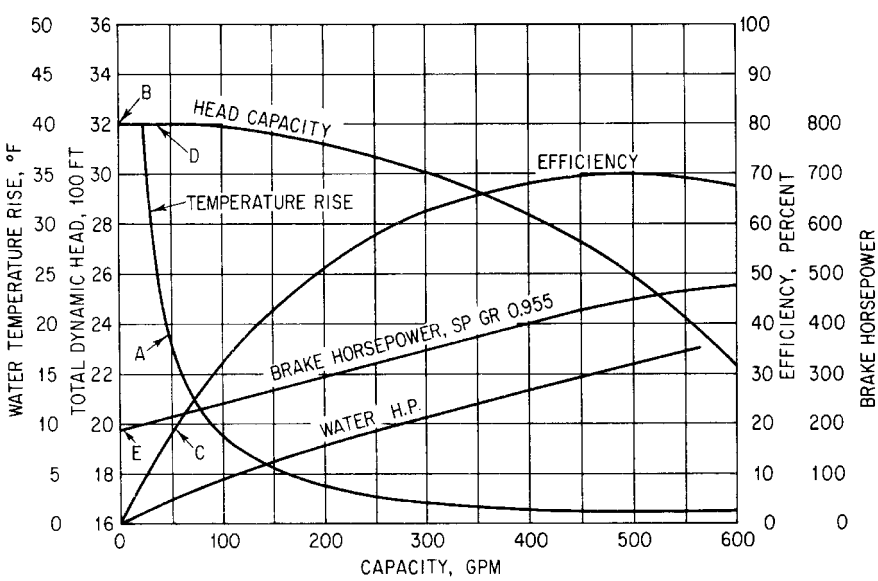


FIG. P-250 Pump characteristics. (Source: Demag Delaval.)

The allowable temperature rise of the water before it flashes into steam depends upon the suction conditions, or NPSH, of the pump. NPSH is the net head above the vapor pressure corresponding to the temperature of the liquid handled. As outlined in the subsection "Net Positive Suction Head," every installation has two types of NPSH: that available in the installation and that required by the pump. The maximum allowable vapor pressure at the pump inlet is found by converting the available NPSH into pounds per square inch and adding this to the vapor pressure corresponding to the temperature of the liquid being handled. This pressure is the vapor pressure corresponding to the temperature to which the liquid may be raised before it will flash into vapor.

After the allowable temperature rise has been established, an approximate minimum safe continuous-flow efficiency can be obtained by rewriting the previous equation in the form

$$\eta = \frac{H_{so}}{778(\Delta t)c + H_{so}}$$

where H_{so} = heat at no flow, or shutoff head
c = specific heat of liquid

The flow corresponding to this efficiency is found on the pump-performance curves.

In boiler-feed pumps having single-suction impellers, all facing in the same direction, a leak-off balancing arrangement is used to compensate hydraulic thrust. If the balancing leak-off flow is returned to the suction of the pump, flashing can occur at extremely low rates of delivery. Therefore, the balancing flow is frequently piped to an open heater in the feed system of the pump, where, by flashing, the temperature of the water will be reduced to that corresponding to the pressure and there should be no valve of any kind between the junction of the balancing connection and the heater. Many installations do pipe the balance flow to the pump suction line.

With the advent of the larger, higher-energy boiler-feed pumps of the 1960s and 1970s, it became apparent that the higher vibration and pressure pulsations occurring at partial capacities would affect the minimum-flow setting. Consequently, minimum-flow values of 25 percent of the flow at best efficiency became commonplace, with higher levels in special cases.

Recirculation connection. The minimum flow through a pump is directed through a recirculation line from a discharge-line connection to the suction source. This line has a recirculation orifice designed to pass the required minimum flow or has a modulating-pressure breakdown valve.

The recirculation connection is in the pump discharge line between the discharge nozzle and the check valve. All valves in the recirculation line must be open whenever the pump is operating under any of the following conditions:

1. Low flows
2. Starting pump
3. Stopping pump

The valves should be opened or closed either manually or by automatic controls. When automatic controls are used, they should be checked at initial starting and occasionally thereafter during starting procedures.

Figure P-251 shows a diagram of the piping arrangement.

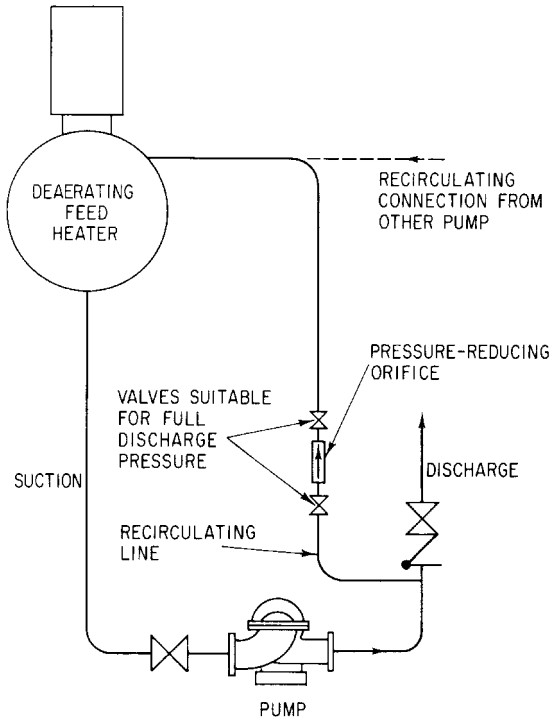


FIG. P-251 Recirculating connections for boiler-feed service. (Source: Demag Delaval.)

Materials for pumping various liquids. The materials used for pumps must be suitable for the liquids handled to prevent excessive corrosion. The *Hydraulic Institute Standards* give the materials to be used for the more common liquids and should be consulted for selecting the applicable material combination.

From the standpoint of materials, pumps may be divided into three basic types: standard fitted (combination of iron and bronze), all iron, and all bronze. Other materials, including corrosion-resisting steels, are listed in the subject standards. Table P-28 gives a summary of the various materials used for centrifugal pumps.

If the liquid to be handled is an electrolyte, the use of dissimilar metals in close combination, especially those that are widely separated in the galvanic series, should be avoided insofar as possible. The use of bronze and iron in the same pump handling seawater will greatly accelerate the corrosion of the cast iron parts.

A table of the galvanic series is given in Table P-29.

Pump application

General. As outlined in the subsection "Classification," there are on the market a multitude of centrifugal-pump types. Some of the basic types will be mentioned here.

Single-stage, double-suction pump. See Fig. P-252. This type of centrifugal pump is the most common one and is used for general service in industrial and municipal plants. In the larger sizes, the use of these pumps is almost universal for municipal-water distribution, and pumps of this type are in service in practically every major city of the United States. For heads up to 300 ft or higher, single-stage pumps are used, while for higher heads two or more units are arranged in series. Figure P-253 shows an installation of two motor-driven units arranged in series.

TABLE P-28 Summary of Material Selections and National Society Standards Designations*

ASTM†	ACI	AISI	Remarks
A48, Classes 20, 25, 30, 35, 40, and 50	...	...	Gray iron—six grades
A339, A395, and A396	...	...	Nodular cast iron—five grades
B143, 1B and 2A; B144, 3A; B145, 4A	...	...	Tin bronze—six grades (includes two grades not covered by ASTM Specifications)
A216—WCB	...	1,030	Carbon steel
A217—C5	...	501	5% chromium steel
A296—CA15	CA15	410	13% chromium steel
A296—CB30	CB30	...	20% chromium steel
A296—CC50	CC50	446	28% chromium steel
A296—CF-8	CF-8	304	18-8 austenitic steel
A296—CF-8M	CF-8M	316	18-8 molybdenum austenitic steel
A296—60T	CN-7M	...	A series of highly alloyed steels normally used where the corrosive conditions are severe
...	...	...	A series of nickel-base alloys
...	...	...	High-silicon cast iron
...	...	...	Austenitic cast iron
A439	...	...	Nodular austenitic cast iron
...	...	...	Nickel-copper alloy
...	...	...	Nickel

*Reprinted from *Hydraulic Institute Standards*, 11th ed., Hydraulic Institute, New York, 1965.
†ASTM = American Society for Testing and Materials; ACI = Alloy Casting Institute; AISI = American Iron and Steel Institute.

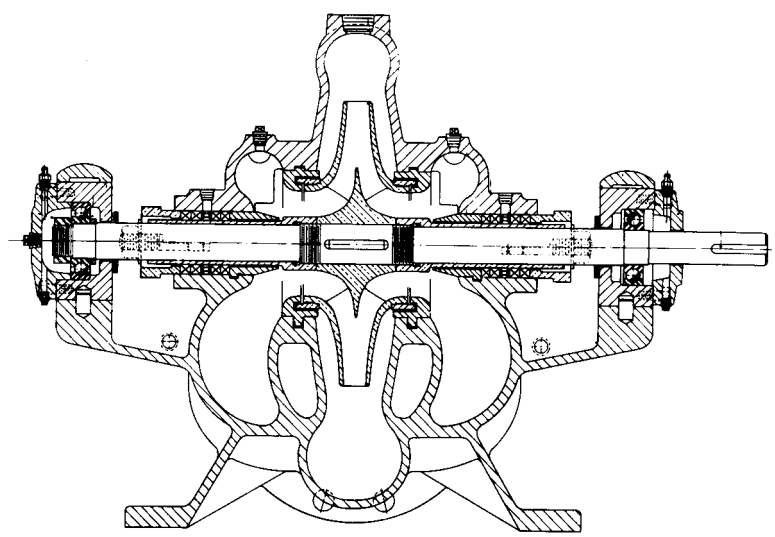


FIG. P-252 Cross-section of a double-suction pump. (Source: Demag Delaval.)

Boiler-feed pumps. For industrial use, multistage split-case pumps having two to six or more stages are employed. See Figs. P-254 and P-255.

For utility service the pressures are now generally in the range of 2000 to 5000 lb/in^2, and barrel-type multistage units are used. See Fig. P-256.

Hydraulic-pressure pumps. Pumps similar to those used in boiler-feed service are employed for this application.

TABLE P-29 Galvanic Series of Metals and Alloys*

Corroded End (Anodic, or Least Noble)

Magnesium
Magnesium alloys

Zinc

Aluminum 2S

Cadmium

Aluminum 17ST

Steel or iron
Cast iron

Chromium stainless steel, 400 series (active)

Austenitic nickel or nickel-copper cast-iron alloy

18-8 chromium-nickel stainless steel, type 304 (active)
18-8-3 chromium-nickel-molybdenum stainless steel, type 316 (active)

Lead-tin solders
Lead
Tin

Nickel (active)
Nickel-base alloy (active)
Nickel-molybdenum-chromium-iron alloy (active)

Brasses
Copper
Bronzes
Copper-nickel alloy
Nickel-copper alloy

Silver solder

Nickel (passive)
Nickel-base alloy (passive)

Chromium stainless steel, 400 series (passive)
18-8 chromium-nickel stainless steel, type 304 (passive)
18-8-3 chromium-nickel-molybdenum stainless steel, type 316 (passive)
Nickel-molybdenum-chromium-iron alloy (passive)

Silver

Graphite
Gold
Platinum

Protected End (Cathodic, or Most Noble)

* Reprinted from *Hydraulic Institute Standards*, 11th ed., Hydraulic Institute, New York, 1965.

Condensate pumps. Special pumps, which may be either single-suction or double-suction, are used for this service; they are designed to operate at low submergence. Figure P-257 shows a three-stage single-suction vertical unit of this type.

Nonclogging pumps. Pumps for this service are designed to ensure maximum freedom from clogging. They are usually of the single-suction type. For the larger sizes, mixed-flow pumps are used, as shown in Fig. P-258.

Design details. The general arrangement of several basic centrifugal-pump types has been covered under "Pump Application." See Figs. P-252 through P-259. Some special design features are discussed below.

Axial balance. The impeller of a single-suction centrifugal pump has an unbalanced hydraulic thrust directed axially toward the suction. This is due to the difference in pressure of the fluid that has passed through the impeller and of the fluid on the



FIG. P-253 Installation of two double-suction pumps arranged in series, motor driven. (Source: Demag Delaval.)

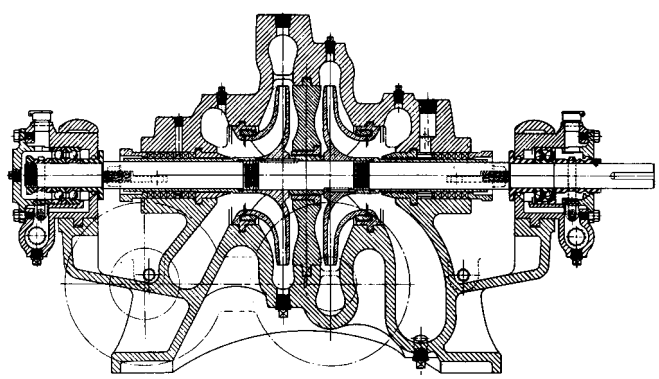


FIG. P-254 Cutaway view of a two-stage pump. (Source: Demag Delaval.)

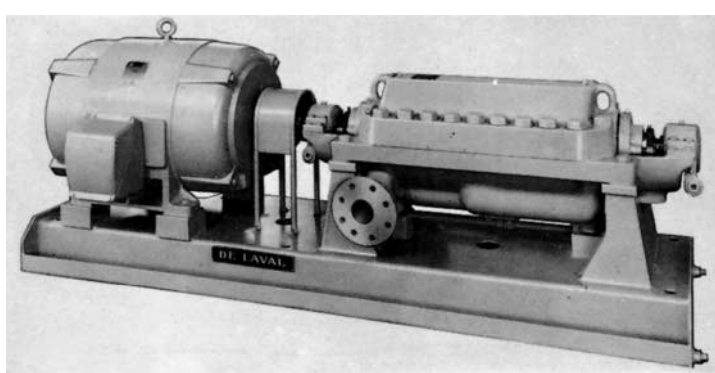


FIG. P-255 Motor-driven multistage pump. (Source: Demag Delaval.)

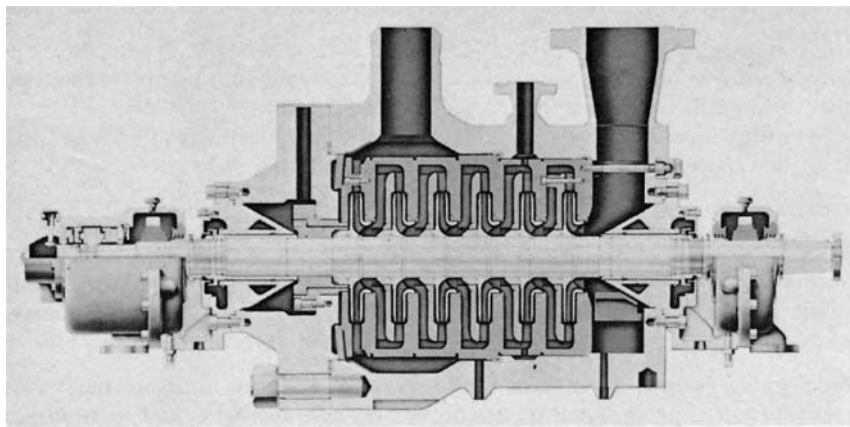


FIG. P-256 Double-case boiler-feedwater pump. (Source: Demag Delaval.)

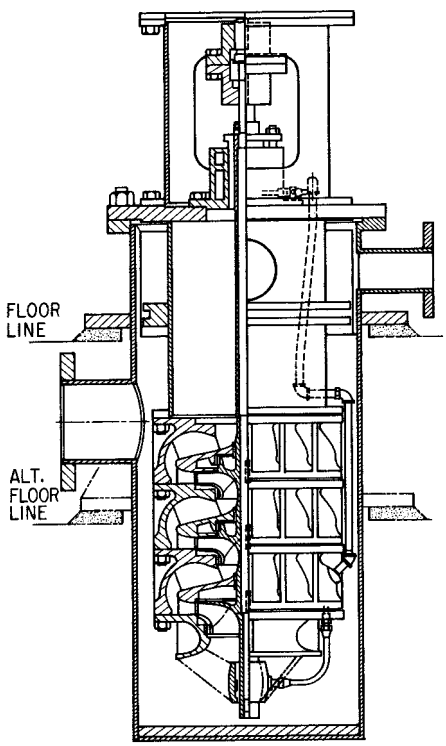


FIG. P-257 Typical assembly of a three-stage condensate pump. (Source: Demag Delaval.)

suction side. Several methods have been devised to counteract this force and avoid the use of large thrust bearings.

If a double-suction impeller (Fig. P-252) is used, the pressures are symmetrical about the centerline and no unbalanced thrust should exist.

On multistage units single-suction impellers placed back to back, as illustrated in Fig. P-254, may be employed. The axial thrust created in one impeller is thus balanced by the corresponding thrust in the other, any remaining thrust being taken by a small thrust bearing.

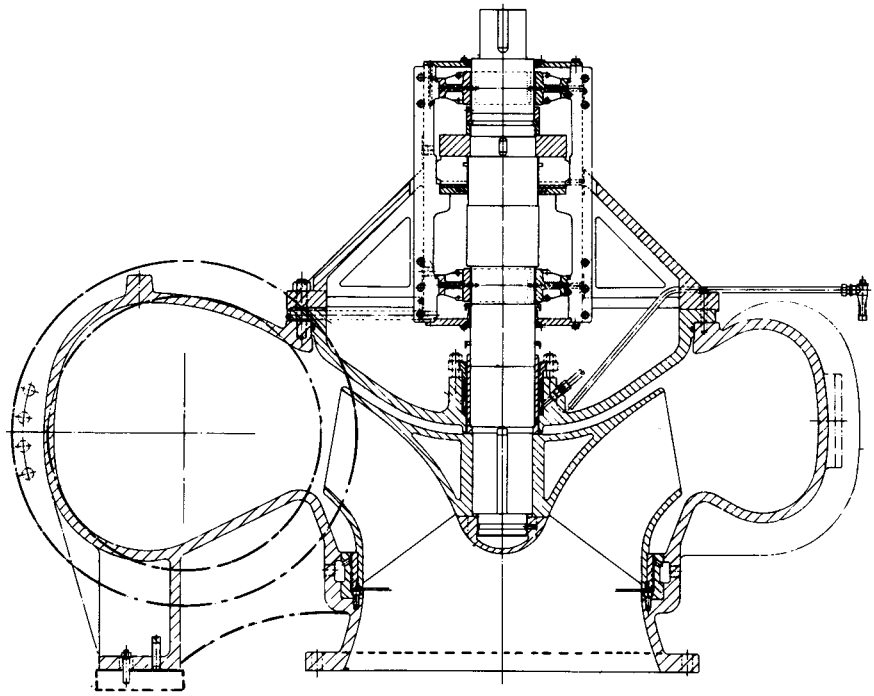


FIG. P-258 Cutaway view of a mixed-flow pump. (Source: Demag Delaval.)

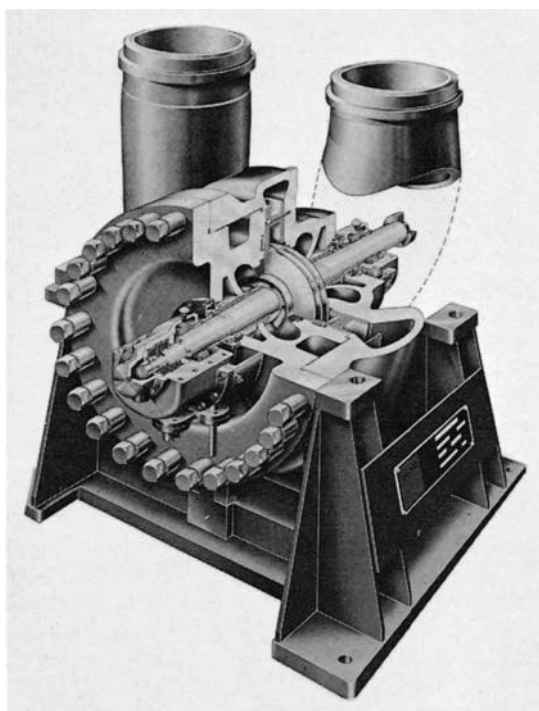


FIG. P-259 Nuclear feedwater pump. (Source: Demag Delaval.)

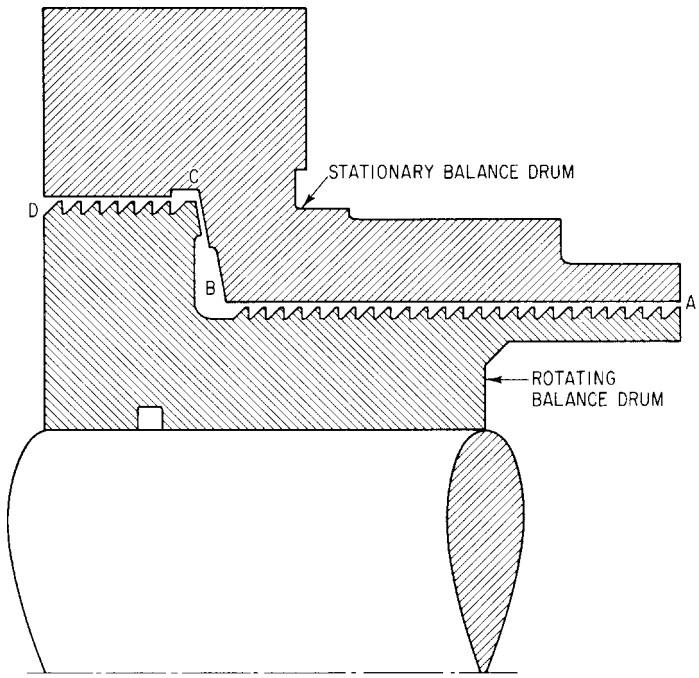


FIG. P-260 Axial balance-drum and disk combination. (Source: Demag Delaval.)

An alternative method of automatically providing a balanced thrust arrangement in boiler-feed pumps is shown in Figs. P-259 and P-260. As shown in Fig. P-260, water at essentially discharge pressure enters the clearance between the rotating and stationary drums at *A*. The water follows a path through the first fixed orifice (*A* to *B*), through the variable orifice (*B* to *C*), and then through the second fixed orifice (*C* to *D*). Chamber *D* is connected by a pipe to the suction source.

Should a condition of increased impeller thrust toward suction occur, the rotor tends to move toward suction, closing the variable orifice between the disk faces (*B* to *C*). By thus reducing the balance flow, the pressure drop between *A* and *B* decreases. The resulting greater pressure at *B* creates an increased thrust in the outboard direction, thereby providing self-compensation for the increased impeller thrust.

A similar type of self-compensating balance occurs if the impeller thrust toward suction should decrease. The rotor is free to move axially and hence permits the variable gap at the disk faces to match the requirements for hydraulic balance.

Still another method of balancing in-line multistage boiler-feed pumps is the use of a fixed-diameter balance drum similar to the *A*-to-*B* portion of Fig. P-260. This method also is reliable, but it generally requires a larger thrust bearing.

Double-volute pumps. The pressure within the casing of any pump develops radial forces when operating at capacities other than normal. The result of these radial forces in single-volute casings is shaft deflection. This is especially true if the pump operates for extended periods at other than design capacity, since imbalance from the radial forces then becomes greater. For such applications, double-volute pumps are used (Fig. P-261).

In the double-volute casing, the water leaving the impeller is collected in two similar volutes, the tongues of which are set 180° apart. The two volutes merge into a common outlet to form the discharge of the pump. Hydraulic forces (indicated by

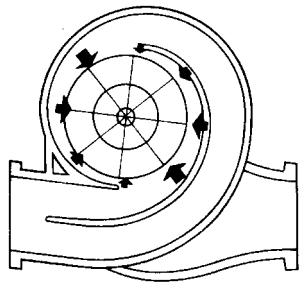


FIG. P-261 Double-volute pump. (Source: Demag Delaval.)

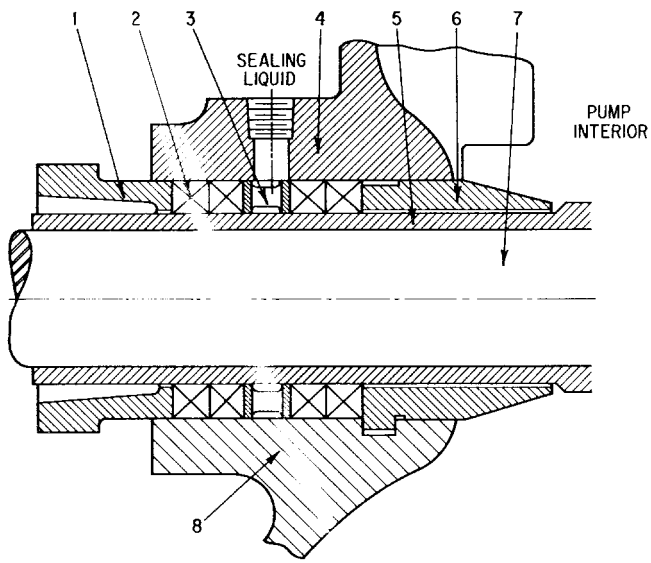


FIG. P-262 (1) Packing gland; (2) packing; (3) seal ring; (4) pump cover; (5) shaft sleeve; (6) throat bushing; (7) shaft; (8) pump case. (Source: Demag Delaval.)

arrows) produced by the pressure in one volute are balanced by equal forces produced by the pressure in the other volute. Thus, radial thrust is counterbalanced and for all practical purposes is eliminated.

The double-volute pumps provide insurance against shaft deflection and savings in repairs and shutdown time.

Centrifugal-pump stuffing boxes. Stuffing boxes are located on pumps where the rotating shaft enters the pump case. They contain packing or mechanical seals that control the leakage of fluid from within or of air from without.

Stuffing boxes. A typical stuffing box using packing, as shown in Fig. P-262, has a plain throat bushing, seal ring, and packing gland. Figure P-263 shows a water-cooled stuffing box with quench glands and breakdown bushing. Stuffing boxes may have various combinations of the features shown in Figs. P-262 and P-263, depending upon operating conditions.

The innermost packing ring is usually placed against a solid removable throat bushing. The plain bushing of Fig. P-262 is used for suction lifts and moderate pressures. Higher pressures require the breakdown bushing of Fig. P-263. It

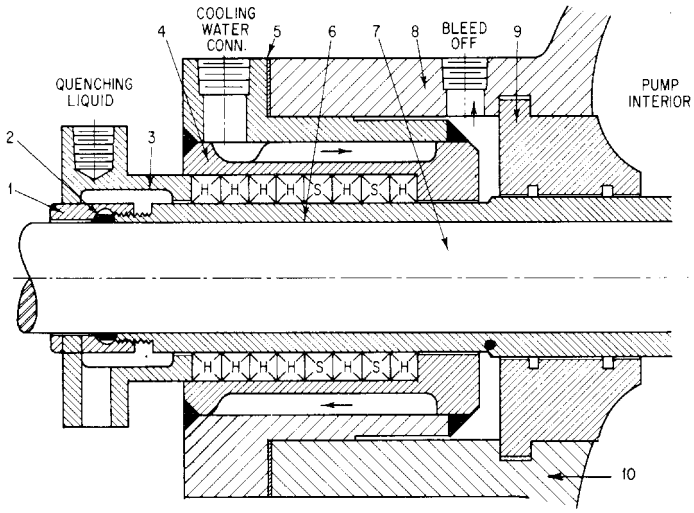


FIG. P-263 (1) Packing nut; (2) packing ring; (3) quench gland; (4) packing box; (5) gasket; (6) shaft sleeve; (7) shaft; (8) pump cover; (9) breakdown bushing; (10) pump case; *H* = metallic packing; *S* = plastic packing. (Source: Demag Delaval.)

relieves the pressure on the packing, reduces mechanical losses, and lessens wear on the shaft sleeve.

Seal rings are placed between rows of packing (see Fig. P-262). They provide a space surrounding the shaft for the sealing liquid. This forms a seal which, when a vacuum exists in the suction chamber of the pump, prevents air from entering. In addition, it ensures lubrication for the packing. The liquid comes from either an outside source or a high-pressure portion of the pump.

Packing is held in the stuffing box by packing glands. These are usually split, making it possible to remove them without taking the pump apart. Quench glands are used when the liquid being pumped exceeds a safe margin on its vapor pressure. Glands are pulled into place by gland bolts. Frequently these are swing bolts, making disassembly easier.

Renewable shaft sleeves usually protect the shaft where it passes through the stuffing boxes. Bronze is most commonly used for cold-water applications. High suction pressures, elevated temperatures, dirty water, and many liquids call for special materials.

The packing-ring cross-section is square. A good grade of braided asbestos impregnated with graphite is most commonly used, especially for water service. This often comes in a continuous coil that is cut into proper lengths for making rings. The cut should preferably be on a diagonal, with a slight gap to allow for expansion when put in place. Gaps for adjacent rings should be staggered. Metallic packing rings are required for some service, and one of the recommended combinations for high-pressure boiler-feed service is shown in Fig. P-263. These packings are usually purchased in molded sets from the packing manufacturer. Also note the special water-jacketed stuffing box in Fig. P-263.

Packing does its sealing along the shaft. It is pliable in order to form itself around the shaft. Cutting down the leakage by tightening up on the packing increases friction, resulting in more power required and increased wear on the shaft or shaft sleeve.

Packing requires a certain amount of leakage to keep it lubricated. Certain applications, such as corrosive acids and inflammable, gritty, or contaminated

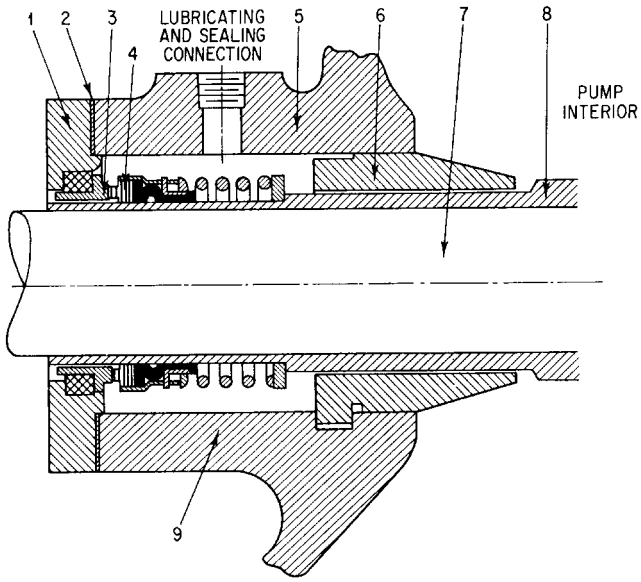


FIG. P-264 Mechanical seal: (1) packing gland; (2) gasket; (3) stationary seal; (4) rotating seal; (5) pump cover; (6) throat bushing; (7) shaft; (8) shaft sleeve; (9) pump case. (Source: Demag Delaval.)

liquids, cannot allow leakage. Mechanical seals are now being used for these applications as well as for high-pressure water seals.

Mechanical seals. Mechanical seals consist of a stationary and a rotating member with some manner of auxiliary seal. The liquid in the stuffing box is prevented from passing along the shaft by means of the auxiliary seal (an O-ring, bellows, wedge ring, or other device). The rotating member is forced against the stationary member by a spring or springs and by pressure in the stuffing box. The liquid tries to leak through the contacting surfaces and forms a liquid-pressure wedge that prevents the seal faces from actually making contact during operation.

Three general types of mechanical seals are used: single seal, double seal, and balanced seal.

A typical single-seal mechanical seal used for most applications is shown in Fig. P-264. Hydraulic forces on the sealing washer are unbalanced, as shown in Fig. P-265, limiting the use of single-seal mechanical seals to moderate pressures.

A typical *balanced* mechanical seal used for high pressures (see Figs. P-266 and P-267) shows how areas on each side of the sealing washer are adjusted to reduce unbalanced forces by stepping down the shaft. This keeps the pressure between sealing surfaces within allowable limits.

A typical double mechanical seal is shown in Fig. P-268. Simply stated, this is two single seals mounted back to back with space for an isolating liquid between them. Such seals are used for liquids that have high temperatures or are gritty, corrosive, volatile, contaminating, etc.

Seals for boiler-feed pumps. Although in the past, the rubbing velocity of the contact surfaces has been a limitation, these velocities increased in the 1970s to 230 ft/s or more, and mechanical-seal applications have increased in boiler-feed pumps, particularly in Europe.

In boiler-feed-pump applications that exceed the limitations of packing or mechanical seals, serrated bushing seals and multifloating ring seals are generally used. Examples of the latter two types are shown in Figs. P-269 and P-270,

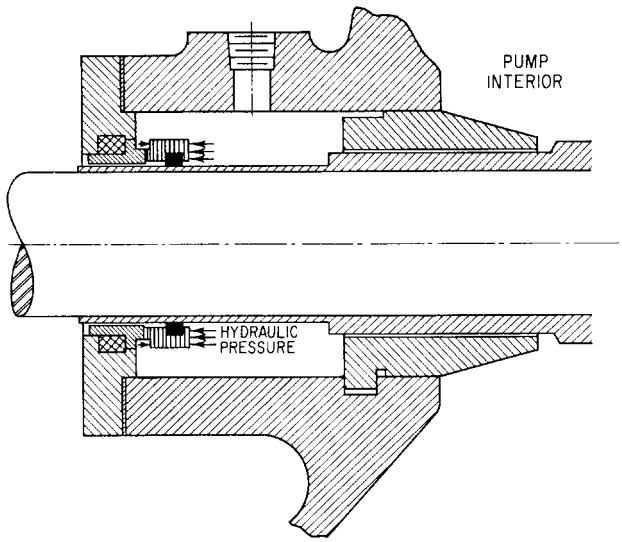


FIG. P-265 Balanced mechanical seal: (1) packing gland; (2) gasket; (3) stationary seal; (4) rotating seal; (5) pump cover; (6) shaft sleeve; (7) throat bushing; (8) shaft; (9) pump case. (Source: Demag Delaval.)

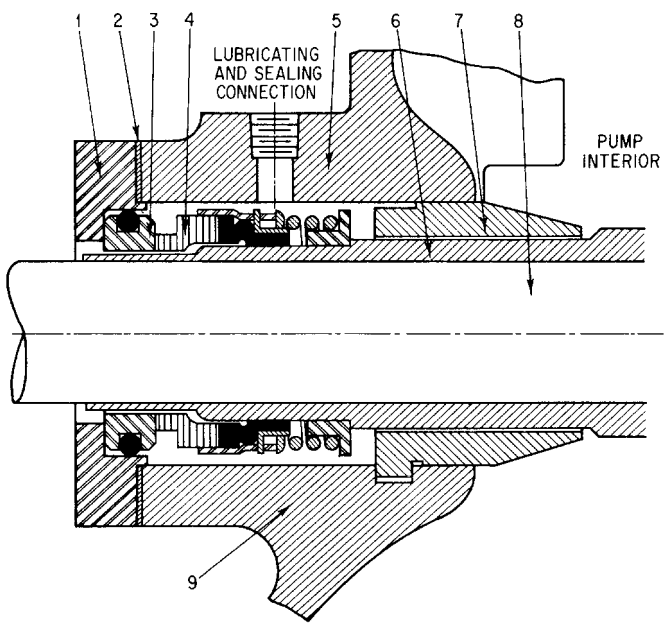


FIG. P-266 Balanced mechanical seal: (1) packing gland; (2) gasket; (3) stationary seal; (4) rotating seal; (5) pump cover; (6) shaft sleeve; (7) throat bushing; (8) shaft; (9) pump case. (Source: Demag Delaval.)

respectively. In both cases, a cool external water supply—generally condensate—is injected into the seals. The cold injection water prevents flashing in the seals that would otherwise occur if the high-temperature water in the pump were permitted to flow through the seals to atmospheric pressure.

In differential-pressure control systems, a constant differential between seal injection-water pressure and seal chamber pressure permits some water to enter

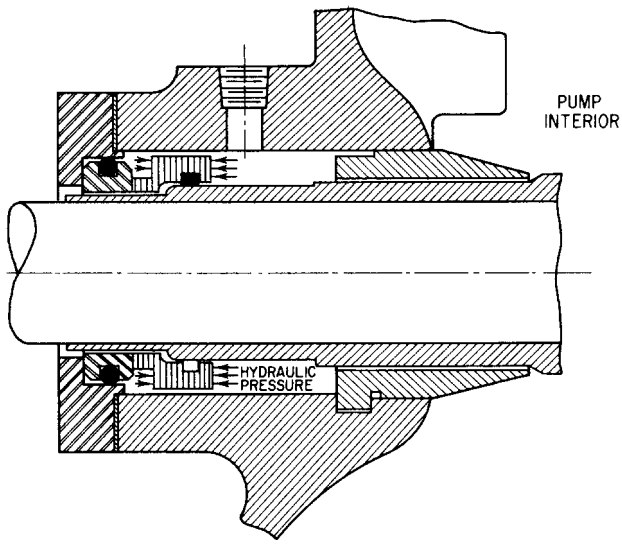


FIG. P-267 Double mechanical seal: (1) packing gland; (2) gasket; (3) stationary seal; (4) rotating seal; (5) pump cover; (6) throat bushing; (7) shaft sleeve; (8) shaft; (9) pump case. (Source: Demag Delaval.)

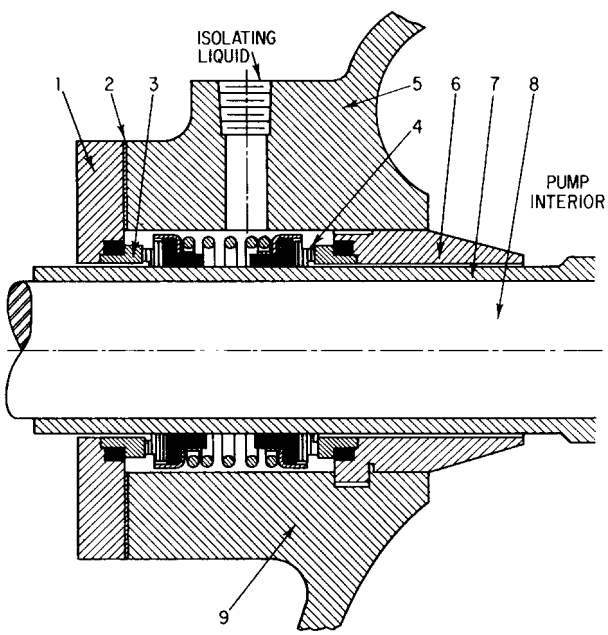


FIG. P-268 Double mechanical seal: (1) packing gland; (2) gasket; (3) stationary seal; (4) rotating seal; (5) pump cover; (6) throat bushing; (7) shaft sleeve; (8) shaft; (9) pump case. (Source: Demag Delaval.)

the seal chamber and the larger portion of the injection water to pass outboard to an atmospheric drain. This has the advantages of having no out-leakage from the pump and having only filtered water flowing from the seals.

In temperature-controlled systems, the amount of seal injection water is regulated by the temperature of the drain water from the seals. One advantage of

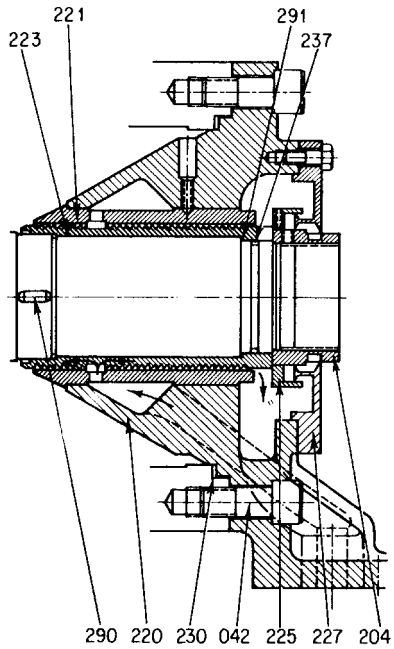


FIG. P-269 Injection-type serrated bushing seal: (042) capscrew; (204) check nut; (220) housing; (221) bushing; (223) sleeve; (225) guard; (227) cover; (230) gasket; (237) O-ring; (290) key; (291) O-ring (metallic). (Source: Demag Delaval.)

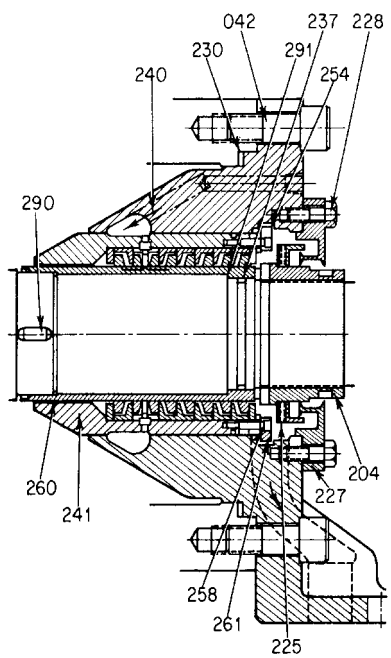


FIG. P-270 Injection-type multifloating ring seal: (042) capscrew; (204) check nut; (225) guard; (227) cover; (228) bolt; (230) gasket; (237) O-ring; (240) housing; (241) seal subassembly; (254) key; (258) capscrews; (260) sleeve; (261) ring; (290) key; (291) O-ring (metallic). (Source: Demag Delaval.)

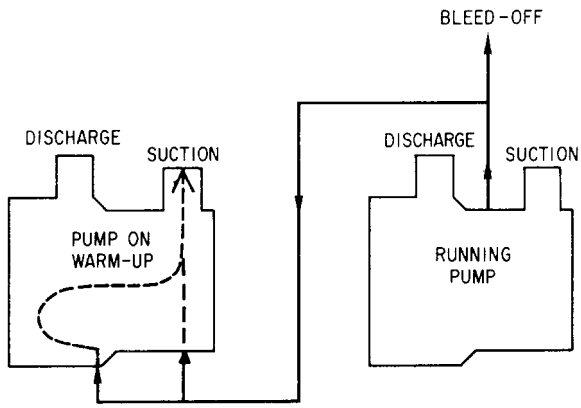


FIG. P-271 Warmup piping. (Source: Demag Delaval.)

this system is that normally no cold water enters the pump since all the seal injection water flows outward.

Although the examples in Figs. P-269 and P-270 show single cooling-water injection, other variations having double injection and/or additional bleed-off connections are used.

Warmup procedure. In starting boiler-feed pumps, the warm-up procedure plays a vital role. It is important to eliminate temperature gradients within the pump to reduce rotor distortion prior to startup. These temperature gradients are usually measured by monitoring case temperatures at locations as determined by the manufacturer. The highest temperature differential should not exceed 50°F, and the difference between the inlet temperature and the warmest part of the case should normally be not more than 50°F.

To obtain minimum temperature gradients, the warm-up water should enter the case through a warmup connection at the bottom of the case and discharge through the suction nozzle. A recommended schematic is shown in Fig. P-271, with water supplied from the bleed-off connection of the operating pump.

Alternative warmup paths may be used, but case temperatures should be the controlling factors.

Effect of operating temperature on pump efficiency at constant speed.* Variations in temperature of the fluid pumped cause changes in the specific weight and viscosity, with resultant changes in the performance of the pump.

Any reduction in specific weight, as caused by an increase in temperature, results in a directly proportional reduction in Lhp (as covered in the subsection "Determination of Power Required") and in input power; so the efficiency is not changed.

Reduced viscosity will have an influence on efficiency, and for pumps in the lower range of specific speed, such as high-pressure multistage boiler-feed pumps and large single-stage hot-water-circulating pumps, reduced viscosity will

1. Increase the internal leakage losses
2. Reduce disk friction losses
3. Reduce hydraulic skin friction or flow losses

* Courtesy of the Hydraulic Institute.

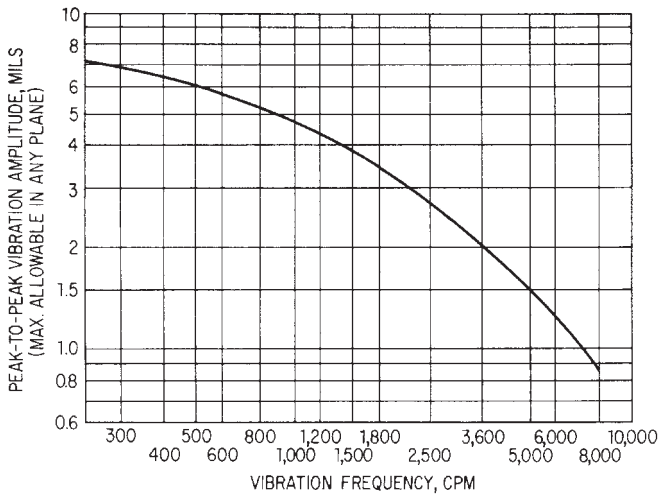


FIG. P-272 Acceptable field vibration limits for centrifugal pumps handling clean liquids. Frequency corresponds to revolutions per minute when dynamic unbalance is the cause of vibration. Vertical pumps: measure vibration at top motor bearing. Horizontal pumps: measure vibration on bearing housing. (Source: Hydraulic Institute.)

The net effect of a reduction in viscosity due to higher temperature will depend on specific speed and on the design details of the pump. When substantiating data are available and when a high degree of understanding exists between the manufacturer and the user, consideration may be given to adjusting the performance data from a cold-water test to hot-water operating conditions on the basis of the following formula:

$$\eta_0 = 1-(1-\eta_t)\left(\frac{v_0}{v_t}\right)^n$$

where η_t = efficiency at test temperature, decimal value
η_0 = efficiency at operating temperature, decimal value
v_0 = kinematic viscosity at operating temperature
v_t = kinematic viscosity at test temperature
n = exponent to be established by manufacturers' data (probably in the range of 0.05 to 0.1)

Typical example of adjustment of efficiency for increased temperature. A test on water at 100°F resulted in an efficiency of 80 percent. What will be the probable efficiency at 350°F?

$$\eta_0 = 1-(1-\eta_t)\left(\frac{v_0}{v_t}\right)^n$$

$$\eta_0 = 1-(1-0.80)\left(\frac{0.00000185}{0.0000076}\right)^{0.1}$$

$$\eta_0 = 1-(0.2)0.868$$

$$\eta_0 = 0.826 = 82.6 \text{ percent}$$

Vibration limits of centrifugal pumps

General. Recommendations for upper limits of vibration of centrifugal-pump units under field-operating conditions are shown in the curves in Figs. P-272 and P-273. Figure P-272 is for centrifugal pumps handling clean liquids, and Fig. P-273 is for vertical or horizontal centrifugal nonclog dry-pit pumps.

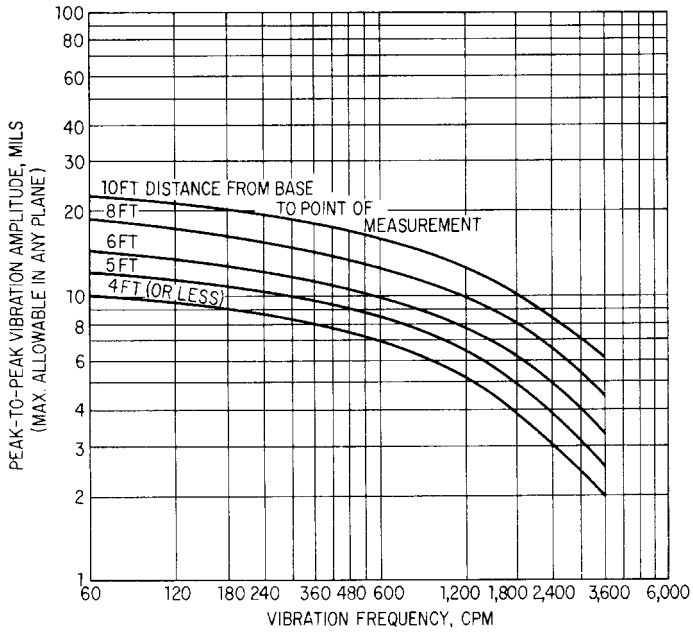


FIG. P-273 Acceptable field vibration limits for vertical or horizontal centrifugal nonclog pumps. Frequency corresponds to revolutions per minute when dynamic unbalance is the cause of vibration. Vertical pumps: measure vibration at top motor bearing. Horizontal pumps: measure vibration on bearing housing. (Source: Hydraulic Institute.)

In using these curves, the following conditions apply:

1. Curves are applicable when the pump is operating at rated speed within ±10 percent of rated capacity.
2. Measurements should be made as indicated on the appropriate curve sheet.
3. Piping should be connected so as to avoid strains on the pump.

The curves should be used as a general guide, with recommendations that vibrations in excess of the curve may require investigation and correction. Often more important than the actual vibration itself is the change of vibration over a period of time. Vibrations in excess of the curves may often be tolerated if they show no increase over considerable periods of time and if there is no other indication of damage, such as an increase in bearing clearance.

Factors affecting vibration. A number of factors besides physical unbalance of the rotating parts may cause vibration. Among these are:

1. Resonance between the unit and its foundation or piping or resonance within the unit due to natural frequency of the pump, the motor frame, the motor-supporting pedestal, or the foundation. Resonant vibrations may also be caused by other equipment in operation in the area.
2. Operation at or near a critical speed. The amount of vibration observed will depend on the degree of unbalance and damping present. Normal design practice is to avoid a critical speed by approximately 25 percent.
3. Vibrations due to hydraulic disturbances caused by improper design of the suction piping or sump. Disturbances may also be caused by improperly designed valves, piping supports, piping, and other components exterior to the pump. Such vibrations are usually at random frequencies.

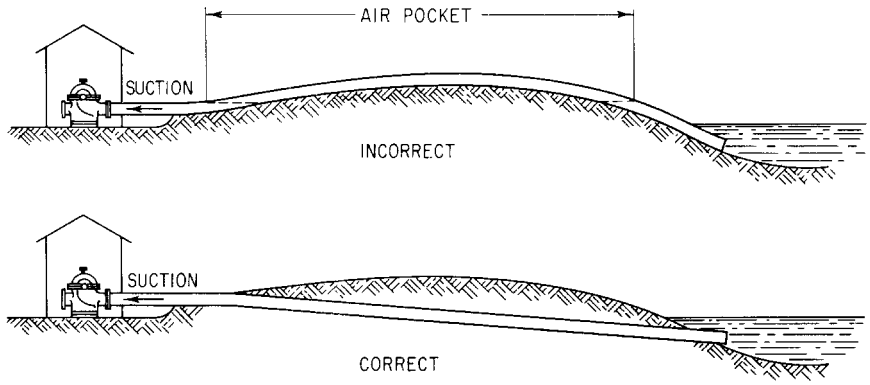


FIG. P-274 Slope of suction pipe. (Source: Hydraulic Institute.)

4. For the nonclog pumps, sudden increases in the vibration levels may be due to the passage of large solids through the pump. If the vibration condition persists, solids may be lodged in the impeller, and remedial measures should be taken to clear it.

Pump installation and operation. In addition to consideration of the correct suction head on the pump (NPSH), other precautions must be followed to ensure satisfactory operation. A few of these factors are:

1. *Accessibility.* The unit should be located where it may be easily inspected and repaired.
2. *Foundation.* The foundation should be heavy and rigid to avoid misalignment and vibration.
3. *Alignment.* The pump and driver should be correctly aligned to avoid excessive wear of the coupling, packing, and bearings. Pumps handling hot liquids should be aligned at their operating temperature.
4. *Piping.* Both suction and discharge lines should be independently supported near the pump to avoid strains on the casing. Piping should have as few bends as possible and be larger than the pump nozzles to reduce head losses.

If an expansion joint is installed in the piping between the pump and the nearest point of anchor in the piping, it should be noted that a force equal to the area of the expansion joint (which may be considerably larger than the normal pipe size) times the pressure in the pipe will be transmitted to the pump proper. Some slip-type couplings have the same effect. This force may exceed the allowable pump loading. If an expansion joint or slip-type coupling must be used, it is recommended that either an anchor be installed between it and the pump or that the joint be restrained or otherwise designed so as to prevent this force from being transmitted to the pump. If properly installed, this will eliminate the objectionable forces mentioned above.

The suction pipe should slope upward to the pump nozzle to avoid pockets in which dissolved air may be liberated (see Fig. P-274). The reducer at the pump suction nozzle should be eccentric rather than a straight taper for the same reason. Any bends should have a long radius, and on the suction side they should be as far from the pump as possible.

5. *Valves.* A check valve and a gate valve should be placed in the discharge line. The former is placed next to the pump to prevent water from running back through the pump if the driver should fail and to protect the pump from excessive line pressure. The latter is used to regulate the flow and in priming.

Priming. Before a centrifugal pump can deliver fluid, it is necessary that the suction side of the impeller be submerged and the suction line filled. The pump should never be run unless the impeller is filled with liquid since the wearing rings may rub and seize; also the packing must be lubricated by the liquid leaking past it. If air should leak into the suction line or casing, the unit may become air-bound and lose its prime, i.e., cease delivery. It is then necessary to stop the pump and prime it before starting it up again.

If the impeller is submerged below the water level, it is only necessary to open a petcock leading from the volute to release any entrapped air and ensure that the pump is primed. If the pump is above the water level, it may be primed by using either a water or a steam-jet exhauster or a small priming pump to remove the air from the pump casing. In such cases there must be a shutoff valve in the discharge line. Water from an outside source, such as a reservoir, or from a filled discharge line may be used to fill the suction line and pump. A foot valve or a check valve must then be placed below the water level in the suction line.

The foot valve is installed in the suction line for priming with low- or medium-suction lifts. Foot valves should not be used for high lifts since failure of the driver would cause the water to rush back suddenly and cause water hammer. A screen is placed before the foot valve to prevent foreign matter from lodging in it.

For large units or those located in remote localities, automatic priming devices that maintain the water level in the pump at a safe level continuously are employed.

Operating difficulties. The following outline, taken from the *Hydraulic Institute Standards*, gives the causes of common operating difficulties:

1. No water delivered
   - *a*. Pump not primed
   - *b*. Speed too low*
   - *c*. Discharge head too high
   - *d*. Suction life higher than for which pump is designed
   - *e*. Impeller completely plugged up
   - *f*. Wrong direction of rotation
2. Not enough water delivered
   - *a*. Air leaks in suction or stuffing boxes
   - *b*. Speed too low*
   - *c*. Discharge head higher than anticipated
   - *d*. Suction life too high (Check with gauges. Check for clogged suction line or screen.)
   - *e*. Impeller partially plugged up
   - *f*. Not enough suction head for hot water
   - *g*. Mechanical defects:
     - (1) Wearing rings worn
     - (2) Impeller damaged
     - (3) Casing packing defective
   - *h*. Foot valve too small
   - *i*. Foot valve or suction opening not submerged deep enough
3. Not enough pressure
   - *a*. Speed too low*
   - *b*. Air in water

* When the pump is direct-connected to an electric motor, check whether the motor is across the line and receives full voltage. When the pump is direct-connected to a steam turbine, make sure that the turbine receives full steam pressure.

c. Mechanical defects:
  (1) Wearing rings worn
  (2) Impeller damaged
  (3) Casing packing defective
d. Impeller diameter too small

4. Pump working for a while and then losing suction
   a. Leaky suction line
   b. Water seal plugged
   c. Suction lift too high
   d. Air or gases in liquid
5. Pump taking too much power
   a. Speed too high
   b. Head lower than rating; too much water pumped
   c. Specific gravity or viscosity too high
   d. Mechanical defects:
     (1) Shaft bent
     (2) Rotating element binding
     (3) Stuffing boxes too tight
     (4) Wearing rings worn
     (5) Casing packing defective

Rotary pumps

The rotary pump is one of the most versatile and widely used types of pump serving industry today. It is the vital heart of the fluid-power systems providing the muscle for most of the equipment involved in the aerospace age. It is the workhorse of the rapidly expanding fluid-power industry, which is providing much of the energy-transfer systems for today's highly sophisticated machines and tools. It is finding ever-widening use in diversified fields of application such as Navy and marine fuel-oil service, marine cargo, oil burners, crude oil, chemical processing, and lubricating service. Its broadest field of application is in the handling of fluids having some lubricating value and sufficient viscosity to prevent excessive leakage at required pressure.

The rotary pump is built in capacities from a fraction of a gallon to more than 5000 gal/min, with pressures ranging up through 10,000 lb/in^2 and handling viscosities from less than 1 cSt to more than 1,000,000 SSU.

The rotary pump is quite often defined as a positive-displacement type by most authoritative engineering references because of the general employment of characteristic close-running clearances that substantially limit internal leakage. It might be more logical and technically correct to drop the *positive* term and refer to this type simply as a *displacement pump*. In the rotary pump, mechanical displacement of the fluid from inlet to outlet is produced by trapping a slug of fluid between one or more moving elements such as gears, cams, screws, vanes, lobes, plungers, or similar devices within a stationary housing or casing. The rotary motion of the centrifugal pump is combined with the positive-pressure characteristic of the reciprocating pump, resulting in a displacement device that delivers a given quantity of fluid with each revolution of the input shaft.

Unlike the centrifugal pump, it is generally self-priming and produces a delivery not severely affected by pressure variations. Speeds of operation are much higher than normally found in a reciprocating pump, with the result that in many instances direct-connected drivers can be used.

Rotaries are available for pumping practically any fluid that will flow, although their greatest specialty is the handling of very viscous fluids. Rotaries are generally simple, compact, and lightweight.

c. Mechanical defects:
  (1) Wearing rings worn
  (2) Impeller damaged
  (3) Casing packing defective
d. Impeller diameter too small

4. Pump working for a while and then losing suction
   a. Leaky suction line
   b. Water seal plugged
   c. Suction lift too high
   d. Air or gases in liquid
5. Pump taking too much power
   a. Speed too high
   b. Head lower than rating; too much water pumped
   c. Specific gravity or viscosity too high
   d. Mechanical defects:
     (1) Shaft bent
     (2) Rotating element binding
     (3) Stuffing boxes too tight
     (4) Wearing rings worn
     (5) Casing packing defective

Rotary pumps

The rotary pump is one of the most versatile and widely used types of pump serving industry today. It is the vital heart of the fluid-power systems providing the muscle for most of the equipment involved in the aerospace age. It is the workhorse of the rapidly expanding fluid-power industry, which is providing much of the energy-transfer systems for today's highly sophisticated machines and tools. It is finding ever-widening use in diversified fields of application such as Navy and marine fuel-oil service, marine cargo, oil burners, crude oil, chemical processing, and lubricating service. Its broadest field of application is in the handling of fluids having some lubricating value and sufficient viscosity to prevent excessive leakage at required pressure.

The rotary pump is built in capacities from a fraction of a gallon to more than 5000 gal/min, with pressures ranging up through 10,000 lb/in^2 and handling viscosities from less than 1 cSt to more than 1,000,000 SSU.

The rotary pump is quite often defined as a positive-displacement type by most authoritative engineering references because of the general employment of characteristic close-running clearances that substantially limit internal leakage. It might be more logical and technically correct to drop the *positive* term and refer to this type simply as a *displacement pump*. In the rotary pump, mechanical displacement of the fluid from inlet to outlet is produced by trapping a slug of fluid between one or more moving elements such as gears, cams, screws, vanes, lobes, plungers, or similar devices within a stationary housing or casing. The rotary motion of the centrifugal pump is combined with the positive-pressure characteristic of the reciprocating pump, resulting in a displacement device that delivers a given quantity of fluid with each revolution of the input shaft.

Unlike the centrifugal pump, it is generally self-priming and produces a delivery not severely affected by pressure variations. Speeds of operation are much higher than normally found in a reciprocating pump, with the result that in many instances direct-connected drivers can be used.

Rotaries are available for pumping practically any fluid that will flow, although their greatest specialty is the handling of very viscous fluids. Rotaries are generally simple, compact, and lightweight.

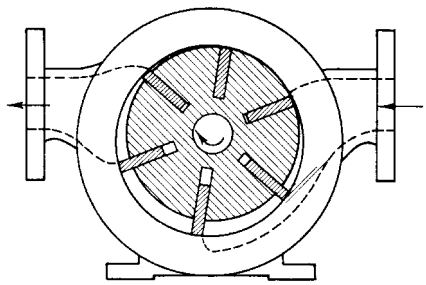


FIG. P-275 Sliding-vane pump. (Source: Demag Delaval.)

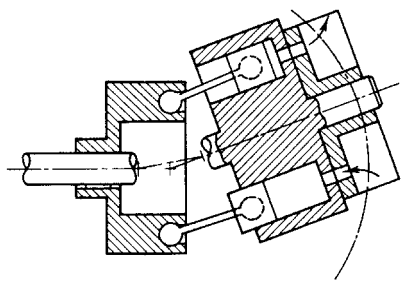


FIG. P-276 Axial piston pump. (Source: Demag Delaval.)

Classification. There are many different types of pumps that fall into the general category of *rotaries*. It is recommended that the reader refer to the *Hydraulic Institute Standards*, "Rotary Pumps," for a detailed description of the various types. A number of the major types are listed and described briefly here.

1. *Vane (sliding)* (Fig. P-275). Vanes, blades, or rollers are located in the periphery of a rotor surrounded by a stator to form cavities between two successive vanes that carry fluid from inlet to outlet.
2. *Piston (axial)* (Fig. P-276). A number of pistons reciprocate within cylinders arranged axially around the periphery of a rotor moving past inlet and outlet ports.
3. *Gear (external)* (Fig. P-277). Fluid is carried between teeth of two external gears and displaced as they mesh.
4. *Gear (internal)* (Fig. P-278). Fluid is carried between teeth of one internal and one external gear and displaced as they mesh.
5. *Lobe* (Fig. P-279). Fluid is carried between one or more lobes on each of two rotors that are timed by separate means.
6. *Screw.* Fluid is carried between screw threads on two or more engaged rotors and is displaced axially as they mesh.
   a. *Timed* (Fig. P-280). Separate timing gears, located either internally or externally, are required to maintain the proper meshed relationship of the screw threads, and rotors are also generally supported by separate sets of bearings.
   b. *Untimed* (Fig. P-281). Rotors incorporate the use of generated thread forms that provide a synchronized gearing action, making separate timing unnecessary.

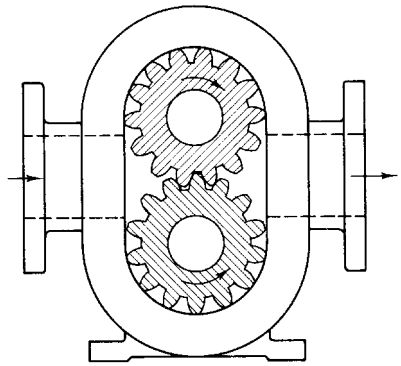


FIG. P-277 External-gear pump. (Source: Demag Delaval.)

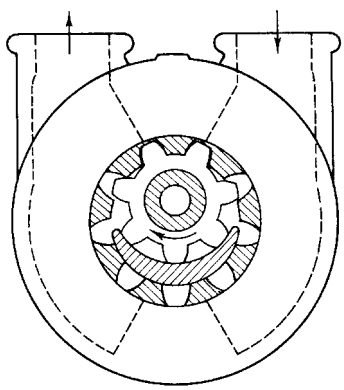


FIG. P-278 Internal-gear pump. (Source: Demag Delaval.)

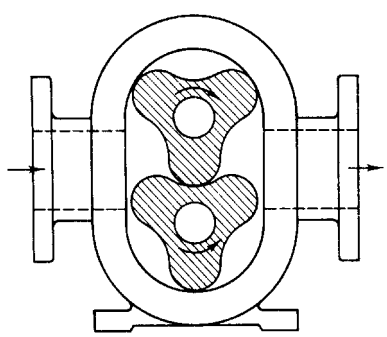


FIG. P-279 Three-lobe pump. (Source: Demag Delaval.)

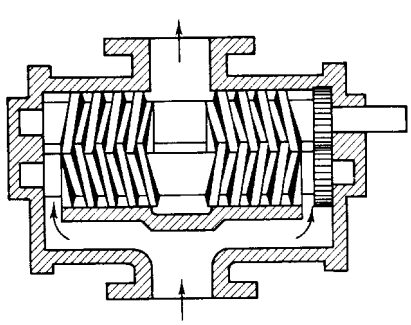


FIG. P-280 Timed-screw pump. (Source: Demag Delaval.)

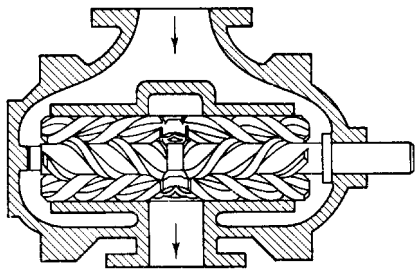


FIG. P-281 Untimed-screw pump. (Source: Demag Delaval.)

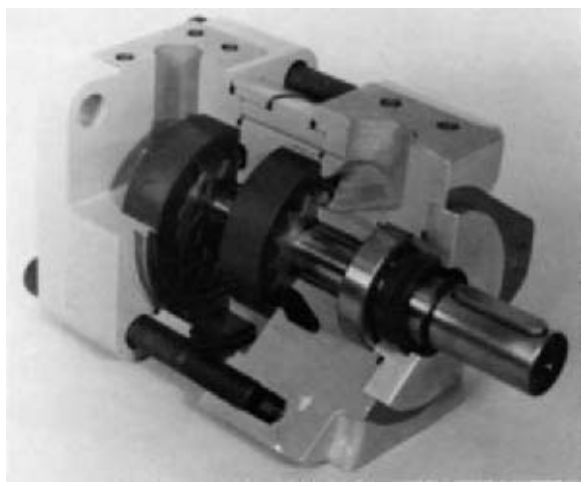


FIG. P-282 Cutaway view of two-stage pump. (Source: Demag Delaval.)

The Transamerica Delaval™ CIG pump (see Fig. P-282) is of the internal-gear type (see Fig. P-278). In this type of pump, fluid is carried from the inlet to the discharge by a pair of gears consisting of one internal and one external gear. The gears are placed eccentrically to each other and are separated by a crescent-shaped divider that provides a sealing path for the internal and external flow paths.

The internal-gear design is generally known for its quiet operation. Modification of the gear profile also provides for reduction of the trapped oil, eliminating any pressure pulsations and thus reducing the noise level. The design is extremely simple and allows gear sets to be stacked into a multistage arrangement for increased pressure rating. With this arrangement, the pressure loads are distributed to reduce stress on the pump components, thus lengthening pump life. The design also has the inherent feature of providing for a hydrodynamic film buildup on the bearings and external gear ring that eliminates metal contact between the working parts, also adding to pump life. The design also provides for double pump configurations consisting of two independent pumps arranged on a common shaft, each pump having a separate discharge and sharing a common suction.

The Transamerica™ Delaval GTS pump is of the externally timed-screw type (see Fig. P-283). The construction of this type of pump is conducive to operation on nonviscous liquids such as water that exclude the use of the IMO design.

This design relies on timing gears for phasing the mesh of the threads and support bearings at each end of the rotors to absorb the reaction forces. With this

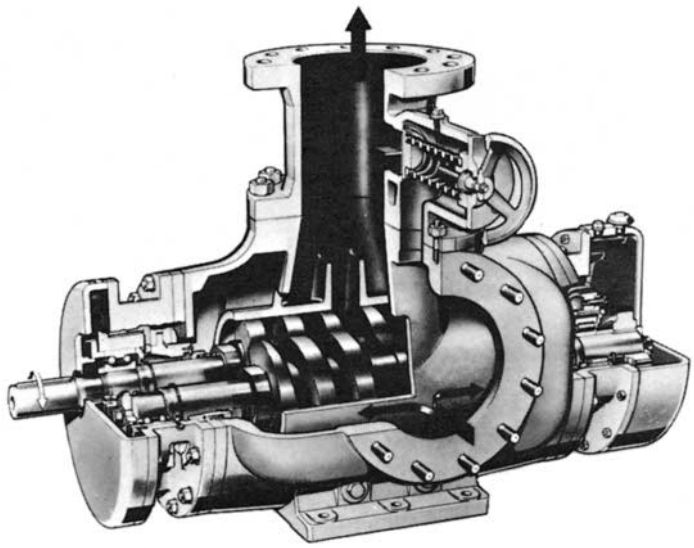


FIG. P-283 Cutaway view of externally timed-screw pump. (Source: Demag Delaval.)

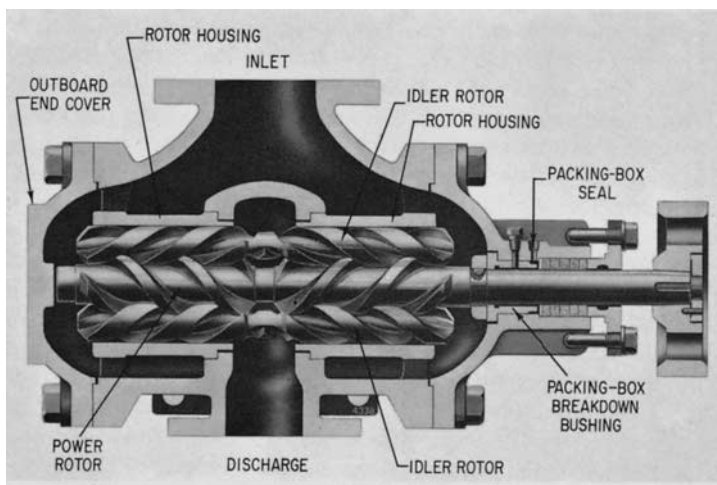


FIG. P-284 Cutaway view of double-end IMO pump. (Source: Demag Delaval.)

arrangement, the threads do not come into contact with each other or with the housing bores in which they rotate. This feature, combined with the external location of the timing gears and bearings, which are oil-bath- or grease-lubricated, makes the pump suitable for handling nonviscous, corrosive, or abrasive fluids.

To provide for operation with corrosive or abrasive fluids, the pump housing can be supplied in a variety of materials including cast iron, ductile iron, cast steel, stainless steel, and bronze. Moreover, the rotor bores can be lined with industrial hard chromium for additional abrasive resistance. The rotors also may be supplied in a variety of materials including cast iron, heat-treated alloy steel, stainless steel, Monel, and Nitralloy. The outside diameter of the rotors can be furnished with hard coatings including tungsten carbide, chromium oxide, and ceramics.

The IMO pump (see Fig. P-284) falls into the untimed-screw category, and it will serve as a base for all further discussion of rotary pumps in general. Because the

fundamental characteristics of all rotaries are similar, many IMO pump features can be related to other types of rotaries without comment; however, when characteristics unique to the IMO pump are mentioned, they will be so identified.

Characteristics. The IMO pump normally is offered as a three-screw type having no need for timing gears or conventional support bearings. It is simple and rugged and has no valves or reciprocating parts to foul. It can run at high speeds, is quiet-operating, and produces a steady pulsation-free flow of fluid.

Properly applied, the IMO pump can handle a wide range of fluids from molasses to gasoline, including modern fire-resistant types, even to 5 percent soluble oil in water. It can be made with hardened wear-resistant rotors to handle some types of contamination and abrasives. Wide ranges of flow and pressure are available.

In the IMO pump, as in most screw pumps, it is the intermeshing of the threads on the rotors and the close fit of the surrounding housing that create one or more sets of moving seals between pump inlet and outlet. These sets of seals act as a labyrinth and provide the screw pump with its positive-pressure capability. Between successive sets of moving seals or threads are voids that move continuously from inlet to outlet. These moving voids, when filled with fluid, carry the fluid along and provide a smooth flow to the outlet, which is essentially pulsationless. Increasing the number of threads or seals between inlet and outlet increases the pressure capability of the pump, the seals again acting similarly to classic labyrinth seals.

The flow of fluid through the screw pump is parallel to the axis of the screws as opposed to the travel around the periphery of centrifugal, vane, and gear-type pumps. This axial flow gives the screw pump ability to handle fluids at low relative velocities for a given input speed, and it is therefore suitable for running at higher speeds, with 1750 and 3500 rpm common for IMO pumps.

The fundamental difference between the IMO pump and other types of screw pumps lies in the method of engaging or meshing the rotors and maintaining the running clearances between them. Timed-screw pumps require separate timing gears between the rotors to provide proper phasing or meshing of the threads. Some sort of support bearing also is required at the ends of each rotor to maintain proper clearances and proper positioning of the timing gears themselves.

The IMO pump rotors are precision-made gearing in themselves, having mating generated thread forms such that any necessary driving force can be transmitted smoothly and continuously between the rotors without need for additional timing gears. The center or driven rotor, called the *power rotor*, is in mesh with two or three close-fitting unsupported *sealing*, or *idler*, *rotors* symmetrically positioned about the central axis by the close-fitting rotor housing. This close-fitting housing and the idlers provide the only transverse bearing support for the power rotor. Conversely, the idlers are transversely supported only by the housing and the power rotor.

The real key to all IMO pump operation is the means employed for absorbing the transverse idler-rotor-bearing loads that are developed as a result of the hydraulic forces built up within the pump to move the fluid against pressure. These rotors and the related housing bores are, in effect, partial journal bearings with a hydrodynamic fluid film being generated to prevent metal-to-metal contact. This phenomenon is most often referred to as the *journal-bearing theory*, and IMO pump behavior is closely related to the applied principles of this theory. The three key parameters of speed, fluid viscosity, and bearing pressure are related exactly as in a journal bearing. If viscosity is reduced, speed must be increased or bearing pressure reduced in order not to exceed acceptable operating limits. For a constant viscosity, however, the bearing-pressure capability can be increased by increasing

TABLE P-30 Safe Axial-Velocity Limits for Various Fluids and Pumping Viscosities

Fluid*	Viscosity, SSU	Velocity, ft/s
Diesel oil	32	30
Lubricating oil	1,000	12
No. C fuel oil	7,000	7
Castor oil	20,000	2
Cellulose	60,000	$^1/_2$

*For characteristics of fuel oils see Table P-31.

the speed. It is this phenomenon that gives the IMO its high-speed capability; in fact, with proper inlet conditions, the higher the IMO pump speed the better the performance and the better the life. This is directly opposite to most rotary-pump behavior.

Since the IMO pump is a displacement device, like all rotaries, it will deliver a definite quantity of fluid with every revolution of the power rotor. If no internal clearances exist, this quantity, called *theoretical capacity* Q_t, would depend only upon the physical dimensions of the rotor set and the speed. Clearances, however, do exist, with the result that whenever a pressure differential occurs, there always will be internal leakage from outlet to inlet. This leakage, commonly called slip S, varies with the pump type or model, amount of clearance, fluid viscosity at pumping conditions, and differential pressure. For any given set of conditions, it is usually unaffected by speed. The delivered capacity, or net capacity Q, therefore, is the theoretical capacity less slip.

The theoretical capacity of any pump can readily be calculated with all essential dimensions known. Basically, IMO pump theoretical capacity varies directly as the cube of the power rotor's outside diameter, which is generally used as the pump-size designator. Thus a relatively small increase in pump size can give a large increase in capacity. Slip can also be calculated but usually is based upon empirical values developed by extensive testing.

Performance

Inlet conditions. The key to obtaining good performance from an IMO pump, as with all other rotaries, lies in a complete understanding and control of inlet conditions and the closely related parameters of speed and viscosity. To ensure quiet, efficient operation, it is necessary to completely fill with fluid the moving voids between the rotor threads as they open to the inlet, and this becomes more difficult as viscosity, speed, or suction lift increases. Basically, if the fluid can properly enter into the rotor elements, the pump will perform satisfactorily.

Remember that a pump does not pull or lift fluid into itself. Some external force must be present to push the fluid into the voids. Normally, atmospheric pressure is the only force present, but in some applications a positive inlet pressure is available.

Naturally the more viscous the fluid, the greater the resistance to flow and, therefore, the lower the rate of filling the moving voids of the threads in the inlet. Conversely, light-viscosity fluids will flow quite rapidly and will quickly fill the moving voids. It is obvious that if the rotor elements are moving too fast, the fill will be incomplete and a reduction in output will result. The rate of fluid flow must always be greater than the rate of void travel or closing to obtain complete filling. Safe rates of flow through the pump for complete filling have been found from experience when atmospheric pressure is relied upon to force the fluid into the rotors. Table P-30 (see also Table P-31) gives these safe axial-velocity limits for various fluids and pumping viscosities.

TABLE P-31 Detailed Requirements for Fuel Oils[a]

Grade of Fuel Oil[b]		Flash Point, °F	Pour Point, °F	Water and Sediment, %	Carbon Residue on 10% Residuum, %	Ash, %	Distillation Tem., °F			Viscosity								Gravity, API
										Saybolt				Kinematic Centistokes at				
										Universal at 100°F		Furol at 122°F		100°F		122°F		
No.	Description	Min	Max	Max	Max	Max	10% Point Max	90% Point Max	End Point Max	Max	Min	Max	Min	Max	Min	Max	Min	Min
1	Distillate oil intended for vaporizing pot-type burners and other burners requiring this grade[c]	100 or legal	0	Trace	0.15	...	420	...	625	...	...	...	...	2.2	1.4	...	...	35
2	Distillate oil for general-purpose domestic heating for use in burners not requiring No. 1	100 or legal	20[d]	0.10	0.35	...	[e]	675	...	40	...	...	...	(4.3)	...	...	...	26
4	Oil for burner installations not equipped with preheating facilities	130 or legal	20	0.50	...	0.10	...	...	...	125	45	...	...	(26.4)	(5.8)	...		
5	Residual-type oil for burner installations equipped with preheating facilities	130 or legal	...	1.00	...	0.10	...	...	...	...	150	40	...	...	(32.1)	(81)		
6	Oil for use in burners equipped with preheaters permitting a high-viscosity fuel	150 or legal	...	2.00[f]	...	...	...	...	...	...	...	300	45	...	...	(638)	(92)	

Reprinted by permission from Commercial Standard CS 12–48 on Fuel Oils of U.S. Department of Commerce.

[a] Recognizing the necessity for low-sulfur fuel oils used in connection with heat treatment, nonferrous metal, glass, and ceramic furnaces, and other special uses, a sulfur requirement may be specified in accordance with the following table:

Grade of fuel oil	Sulfur, maximum %
No. 1	0.5
No. 2	1.0
Nos. 4, 5, and 6	No limit

Other sulfur limits may be specified only by mutual agreement between the buyer and seller.

[b] It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

[c] No. 1 oil shall be tested for corrosion for 3 h at 122°F. The exposed copper strip shall show no gray or black deposit.

[d] Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0°F under any conditions.

[e] The 10 percent point may be specified at 440°F; maximum for use in other than atomizing burners.

[f] The amount of water by distillation plus the sediment by extraction shall not exceed 2.00 percent. The amount of sediment by extraction shall not exceed 0.50 percent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 percent.

It is thus quite apparent that pump speed must be selected to satisfy the viscosity of the fluid to be pumped. The pump manufacturer generally must supply the determination of the axial velocity through a screw pump, although the calculation is quite simple when the driving-rotor speed and screw-thread lead are known. The lead is the advancement made along the thread during a complete revolution of the rotor as measured along the axis. In other words, it is the travel of the fluid slug in one complete revolution.

In this handbook, the more general term *fluid* is used to describe the fluids handled by rotaries that may contain or be mixed with matter in other than the liquid phase. The word *liquid* is used only to describe true liquids that are free of vapors and solids. Most of the fluids handled by rotary pumps, especially petroleum oils, because of their complex nature contain certain amounts of entrained and dissolved air or gas that is released as vapor when the fluid is subjected to pressures below atmospheric. If the pressure drop required to overcome entrance losses to push such a fluid into the rotor voids is sufficient to reduce the pressure so that vapors are released in the rotor voids, cavitation results. This leads to noisy operation and an attendant reduction in output. It is therefore very important to be aware of the characteristics of the entrained air and gas of the fluids to be handled. In fact, it is so important that a more detailed study of this relatively complex subject is included below in the subsection "Effect of Entrained or Dissolved Gas on Performance."

Speed. The speed N of a rotary pump is the number of revolutions per minute of the driving rotor. In most instances this is the input shaft speed; however, in some geared-head units the driving-rotor speed can differ from the input shaft speed.

Capacity. The actual delivered capacity of any rotary pump, as stated earlier, is theoretical capacity less internal leakage or slip when handling vapor-free fluids. For a particular speed, this may be written $Q = Q_t - S$, where the standard unit of Q and S is the U.S. gallon per minute. Again, if the differential pressure is assumed to be zero, the slip may be neglected and $Q = Q_t$.

The term *displacement D* is of some general interest, although it is no longer used in rotary-pump calculations. It is the theoretical volume displaced per revolution of the driving rotor and is related to theoretical capacity by speed. The standard unit of displacement is cubic inches per revolution; thus $Q_t = DN \div 231$. The terms *actual displacement* and *liquid displacement* are also less frequently used for rotary-pump calculations but continue to be used for some theoretical studies. Actual displacement is related to delivered capacity by speed.

The actual delivered capacity of any specific rotary pump is reduced by

1. Decreasing speed
2. Decreased viscosities
3. Increased differential pressure

The actual speed must always be known and most often differs somewhat from the rated or nameplate specification. This is the first item to be checked and verified in analyzing any pump's operating performance. It is surprising how often the speed is incorrectly assumed and later found to be in error.

Because of the internal clearances between rotors and the housing of a rotary pump, lower viscosities and higher pressure increase slip, which results in a reduced delivered capacity for a given speed. The impact of these characteristics can vary widely for the various types of rotary pumps encountered. The slip, however, is not measurably affected by changes in speed and thus becomes a smaller percentage of the total flow with the use of higher speeds. This is a very significant

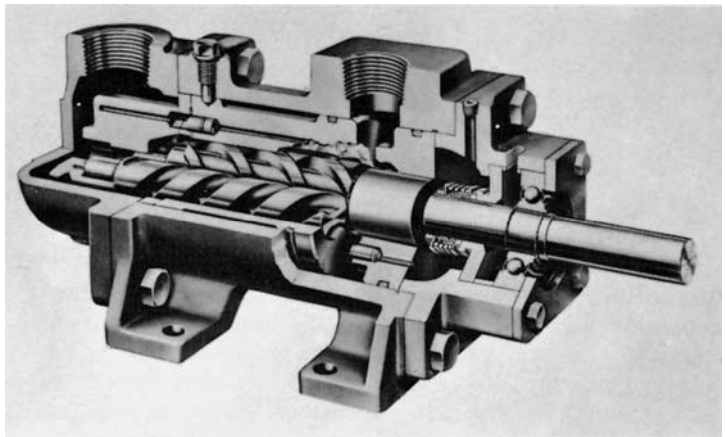


FIG. P-285 Cutaway view of single-end IMO pump; two closures. (Source: Demag Delaval.)

factor in dealing with the handling of light viscosities at higher pressures, particularly in the case of devices, such as the IMO pump, that favor high speed. Always run at the highest speed possible for best results and best volumetric efficiency with the IMO pump. This will not generally be the case with rotaries having support-bearing speed limits.

Pump volumetric efficiency V_y is calculated as $V_y = Q/Q_t = (Q_t - S)/Q_t$, with Q_t varying directly with speed.

As stated previously, theoretical capacity of an IMO pump is a function that varies directly as the cube of the power rotor's outside diameter for a standard three-rotor pump configuration. For a constant speed, a 2-in rotor will have a theoretical capacity 8 times that of a 1-in rotor size. However, for a given model, slip varies directly as the square of the rotor size; therefore, the slip of the 2-in rotor is 4 times that of a 1-in rotor with all fluid variables held constant.

On the other hand, viscosity change affects the slip inversely to some power which has been determined empirically. An acceptable approximation for 100 to 10,000 SSU is obtained by using the one-half power. Slip varies directly with approximately the square root of differential pressure, and a change from 400 to 100 SSU will double the slip just as a differential-pressure change from 100 to 400 lb/in^2.

Pressure. The pressure capability of different types of rotary pumps varies widely. Some of the gear and lobe types are fairly well limited to 100 lb/in^2, normally considered low pressure. Other gear and vane types perform very well in the moderate-pressure range (100 to 500 lb/in^2) and beyond. Some types can operate well in the high-pressure range, while others such as axial piston pumps can work at 5000 lb/in^2 and above. The slip characteristic of a particular pump is one of the key factors that determine the acceptable operating range, which generally is well defined by the pump manufacturer; however, all applications for high pressure should be approached with some caution, and the manufacturer or the manufacturer's representative should be consulted.

The IMO pump is suitable for a wide range of pressures from 50 to 5000 lb/in^2, dependent upon the selection of the right model. Internal leakage can be restricted for high-pressure applications by introducing increased numbers of moving seals or threads between inlet and outlet (see Figs. P-285 through P-287). The number of seals between inlet and outlet normally is specified for a particular model in terms of number of closures. The number of closures is increased to obtain higher-pressure capability, which also results in increased pump length for a given rotor size.

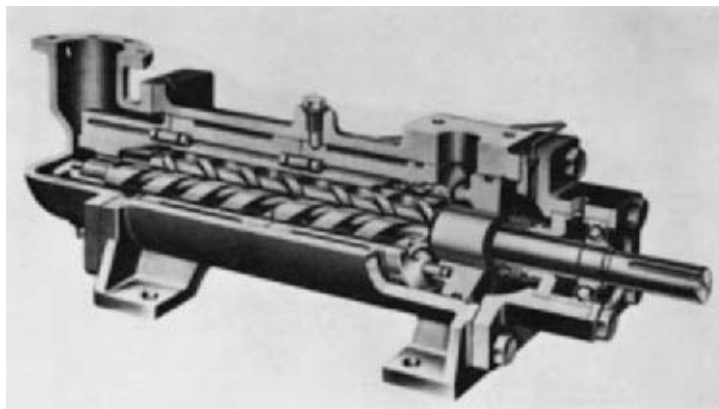


FIG. P-286 Cutaway view of single-end IMO pump; five closures. (Source: Demag Delaval.)

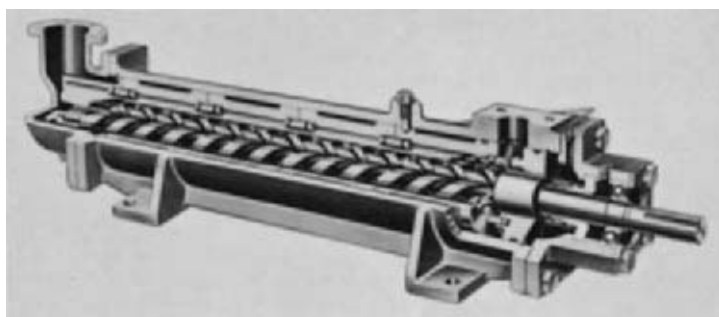


FIG. P-287 Cutaway view of single-end IMO pump; 11 closures. (Source: Demag Delaval.)

TABLE P-32 IMO Pumps

Number of Closures	Maximum Pressure, lb/in^2
1	100
2	500
3	1500
5	3000
11	5000

IMO pumps generally are available with predetermined numbers of closures versus maximum pressure rating when rated at 150 SSU and 3500 rpm in the 10- to 100-gal/min range (see Table P-32).

Horsepower. The brake horsepower (bhp) required to drive a rotary pump is the sum of the theoretical liquid horsepower and the internal power losses. The theoretical liquid horsepower is the actual work done in moving the fluid from its inlet-pressure condition to the outlet at discharge pressure.

Note: This work is done on all the fluid of theoretical capacity, not just delivered capacity, because slip does not exist until a pressure differential occurs. Rotary-pump power ratings are expressed in terms of horsepower (550 ft·lb/s), and

theoretical liquid horsepower can be calculated: tLhp = $Q_t \Delta P \div 1{,}714$. It should be noted that the theoretical liquid horsepower is independent of viscosity and is concerned only with the physical dimensions of the pumping elements, the rotative speed, and the differential pressure.

Internal power losses are of two types: mechanical and viscous. Mechanical losses include all the power necessary to overcome the mechanical friction drag of all the moving parts within the pump, including rotors, bearings, gears, mechanical seals, etc. Viscous losses include all the power lost from the fluid viscous-drag effects against all the parts within the pump as well as from the shearing action of the fluid itself. It is probable that the mechanical loss is the major component when operating at low viscosities and high speeds while the viscous loss is the larger at high viscosity and slow-speed conditions.

No direct comparison can easily be made between various types of rotary pumps for internal power loss, as this falls into the category of closely guarded trade secrets. Most manufacturers have established their own data on the basis of tests made under closely controlled operating conditions, and they are very reluctant to divulge their findings. In general, the losses for a given type and size of pump vary with viscosity and rotative speed and may or may not be affected by pressure, depending upon the type and model of pump under consideration. These losses, however, must always be based upon the maximum viscosity to be handled since they will be highest at this point.

The actual pump power output (whp), or delivered liquid horsepower, is the power imparted to the fluid by the pump at the outlet. It is computed in the same way as theoretical liquid horsepower, using Q in place of Q_t; hence the value will always be less.

Pump efficiency is the ratio of whp to bhp.

Application and selection. In the application of rotary pumps certain basic factors must be considered to ensure a successful installation. These factors are fundamentally the same regardless of the fluids to be handled or the pumping conditions.

The pump selection for a specific application is not difficult if all the operating conditions are known. It is often quite difficult, however, to obtain accurate information as to these conditions. This is particularly true of inlet conditions and viscosity, since it is a common feeling that inasmuch as the rotary pump is a positive-displacement device, these items are unimportant.

In any rotary-pump application, regardless of the design, suction lift, viscosity, and speed are inseparable. Speed of operation, therefore, is dependent upon viscosity and suction lift. If a true picture of these two items can be obtained, the problem of making a proper pump selection becomes simpler, and it is probable that the selection will result in a more efficient unit.

Viscosity. It is not very often that a rotary pump is called upon to handle fluids having a constant viscosity. Normally, because of temperature variations, it is expected that a range of viscosity will be encountered, and this range can be quite wide; for instance, it is not unusual that a pump is required to handle a viscosity range of 150 to 20,000 SSU, the higher viscosity usually being due to cold-starting conditions. This is a perfectly satisfactory range insofar as a rotary pump is concerned; but if information can be obtained concerning such things as the amount of time during which the pump is required to operate at the higher viscosity and whether or not the motor can be overloaded temporarily, a multispeed motor can be used, or the discharge pressure can be reduced during this period, a better selection can often be made.

Quite often no viscosity but only the type of fluid is given. In such cases assumptions can sometimes be made if sufficient information is available concerning the fluid in question. For instance, Bunker C, or Number 6, fuel oil is known to have a wide latitude as to viscosity and usually must be handled over a considerable temperature range. The normal procedure in a case of this type is to assume an operating viscosity range of 20 to 700 SSF. The maximum viscosity, however, may very easily exceed the higher value if extra heavy oil is used or exceptionally low temperatures are encountered. If either should occur, the result may be improper filling of the pumping elements, noisy operation, vibration, and overloading of the motor.

Although it is the maximum viscosity and the expected suction lift that determine the size of the pump and set the speed, it is the minimum viscosity that determines the capacity. Rotary pumps must always be selected to give the specified capacity when handling the expected minimum viscosity, since this is the point at which maximum slip, hence minimum capacity, occurs. If this rule is not followed, the pump will not meet the requirements of the system unless a considerable margin has been allowed initially in specifying capacity or there is overcapacity available in the pump. The latter is often the case, since practically all rotary pumps are made in certain stock sizes and it is standard practice to apply the next larger pump when a capacity that falls between sizes is specified.

It should also be noted that the minimum viscosity often sets the model of the pump selected since it is more or less standard policy of most manufacturers to downrate their pumps, insofar as allowable pressure is concerned, when handling liquids having a viscosity of less than 100 SSU. This is done for two reasons: first, to avoid the poorer volumetric efficiency as a result of increased slip under these conditions; and second, because a film of the liquid must be maintained between the closely fitted parts that is likely to break down if a combination of low viscosity and high pressure should occur. Although viscosity is not necessarily a definite criterion of film strength, it is generally so used by pump manufacturers.

Entrained air. As mentioned previously, a factor that must also be given careful consideration is the possibility of entrained air or gas in the fluid to be pumped. This is particularly true of installations in which recirculation occurs and the fluid is exposed to air through either mechanical agitation, leaks, or improperly located drain lines.

Likewise, most fluids will also dissolve air or gas, retaining it in solution, the amount depending upon the liquid itself and the pressure to which it is subjected. It is known, for instance, that lubricating oils under conditions of atmospheric temperature and pressure will dissolve up to 10 percent air by volume and gasoline up to 20 percent.

When pressures below atmospheric exist at the pump inlet, dissolved air will come out of solution, and both this and entrained air will expand in proportion to the existent absolute pressure. This expanded air will accordingly take up a proportionate part of the available volume of the moving voids with a consequent reduction in delivered capacity. (See subsection “Effect of Entrained or Dissolved Gas on Performance.”)

One of the apparent effects of handling fluids containing entrained or dissolved air or gas is noisy pump operation. When such a condition occurs, it is usually dismissed as *cavitation*; then, too, many operators never expect anything but noisy operation from rotary pumps. This should not be the case. With properly designed systems of pumps, quiet, vibration-free operation can be produced and should be expected. Noisy operation is inefficient, and steps should be taken to make corrections until the objectionable conditions are overcome. It is true, of course, that

some types of pumps are more critical to the handling of air than others; this is usually due to the high inlet losses inherent in these types, but proper design and speed selection can go a great way toward overcoming the problem.

It should be pointed out that if a pump will be called on to handle fluids containing entrained air, this fact should be included in any specifications that may be written and the percentage specified.

Nonnewtonian fluids. The viscosity of most liquids, as, for example, water and mineral oil, is unaffected by any agitation to which they may be subjected as long as the temperature remains constant; these liquids are accordingly known as *true*, or *newtonian*, fluids. There is another class of liquids, however, such as cellulose compounds, glues, greases, paints, starches, slurries, and candy compounds, which change in viscosity as agitation is varied at constant temperature. The viscosity of these fluids will depend upon the shear rate at which it is measured, and these fluids are termed *nonnewtonian*.

If a fluid is known to be nonnewtonian, the expected viscosity under actual pumping conditions should be determined, since it can vary quite widely from the viscosity under static conditions. One instance concerned the handling of a cellulose product for which the viscosity was given as 20,000 SSU, which was its actual static, or apparent, viscosity. It developed that under actual pumping conditions the viscosity was approximately 500 SSU. No serious harm was done, but a large low-speed pump was installed when a smaller, cheaper, higher-speed unit could have been used.

Since a nonnewtonian fluid can have an unlimited number of viscosity values (as the shear rate is varied), the term *apparent viscosity* is used to describe its viscous properties. Apparent viscosity is expressed in absolute units and is a measure of the resistance to flow at a given rate of shear. It has meaning only if the shear rate used in the measurement is also given.

The grease-manufacturing industry is very familiar with the nonnewtonian properties of its products, as evidenced by the numerous curves wherein *apparent viscosity* is plotted against *rate of shear* that have been published. The occasion is rare, however, when one is able to obtain accurate information as to viscosity if it is necessary to select a pump for handling this fluid.

It is understood that it is practically impossible, in most instances, to give the viscosity of grease in the terms most familiar to the pump manufacturer, i.e., Saybolt Seconds Universal or Saybolt Seconds Furol; but if only a rough approximation could be given, it would be of great help.

For applications of this type, data taken from similar installations are most valuable. Such information should consist of type, size, capacity, and speed of already installed pumps, suction pressure, and temperature at the pump-inlet flange, total working suction head, and above all the pressure drop in a specified length of piping. From the latter, an excellent approximation of viscosity under actual operating conditions can be obtained.

Suction conditions. Suction lift occurs when the total suction head at the pump inlet is below atmospheric pressure. It is normally the result of a static lift and pipe friction. Although rotary pumps are capable of producing a high vacuum, it is not this vacuum that forces the fluid to flow. As previously explained, it is atmospheric pressure that forces the fluid into the pump. Since atmospheric pressure at sea level corresponds to 14.7 psia, or 30 inHg, this is the maximum amount of pressure available for moving the fluid, and suction lift cannot exceed these figures. Actually, it must be somewhat less since there are always pump-inlet losses that must be taken into account. It is considered the best practice to keep suction lifts just as low as possible.

The majority of rotary pumps operate with suction lifts of approximately 5 to 15 inHg. Lifts corresponding to 24 to 25 inHg are not uncommon, and there are numerous installations operating continuously and satisfactorily in which the absolute suction pressure is within $^1/_2$ in of the barometer. In the latter cases, however, the pumps are usually taking the fluid from tanks under vacuum and no entrained or dissolved air or gases are present. Great care must be taken in selecting pumps for these applications, since the inlet losses can very easily exceed the net suction head available for moving the fluid into the pumping elements.

There are many known instances of successful installations in which pumps were properly selected for high-suction conditions. There are also, unfortunately, many other installations with equally high suction lifts which are not so satisfactory. This is because proper consideration was not given, at the time when the installations were made, to the actual suction conditions at the pump inlet. Frequently, suction conditions are given as "flooded" simply because the source feeding the pump is above the inlet flange. Absolutely no consideration is given to outlet losses from the tank or pipe friction, and these can be exceptionally high when dealing with extremely viscous fluids.

When it is desired to pump extremely viscous fluids such as grease, chilled shortening, and cellulose preparations, care should be taken to use the largest possible size of suction piping, eliminate all unnecessary fittings and valves, and place the pump just as closely as possible to the source of supply. In addition, it may be found necessary to supply the fluid to the pump under some pressure, which may be supplied by elevation, air pressure, or mechanical means.

Speed. It was previously stated that viscosity and speed are closely linked and that it is impossible to consider one without the other. Although rotative speed is the ultimate outcome, the basic speed which the manufacturer must consider is the velocity of the fluid going through the pump; this is a function of pump type and design. Certain types, such as gear and vane pumps, carry the fluid around the periphery of the pumping elements, and as a result, the velocity of the fluid through the pump can become quite high unless relatively low rotative speeds are used. On the other hand, in screw-type pumps the flow is axial and fluid speeds are relatively lower, with the result that higher rotative speeds can be used. On the basis of handling light fluids, say, 100 to 500 SSU, gear- or vane-type pumps rarely exceed a rotative speed of 1200 rpm except in the case of a very small unit or special designs for a particular use such as for aircraft purposes. Screw pumps, however, for which timing gears are not required, commonly operate without difficulty at speeds up to 5000 rpm, and IMO pumps have been run in the field to 24,000 rpm.

Although rotative speeds are relative and depend upon the pump type, they usually should be reduced when handling fluids of high viscosity. This is due not only to the difficulty of filling the pumping elements but also to the mechanical losses that result from the shearing action of these parts on the fluid handled. The reduction of these losses is frequently of more importance than relatively high speeds, even though the latter might be possible because of positive inlet conditions.

Rotary pumps do not in themselves create pressure; they simply transfer a quantity of fluid from the inlet to the outlet side. The pressure developed on the outlet side is solely the result of resistance to flow in the discharge line. If, for example, a pump were to be set up and run without a discharge line, a gauge placed at the pump outlet flange would register zero no matter how fast or how long the pump was run.

Pipe size. Resistance usually consists of differences of elevation, fixed resistances such as orifices, and pipe friction. Nothing much can be done about the first two,

since these are the basic reasons for using a pump. Something, however, can be done about pipe friction. Millions of dollars are thrown away annually because of the use of piping that is too small for the job. To be sure, all pipe friction cannot be eliminated as long as fluids must be handled in this manner, but every effort should be made to use the largest pipe that is economically feasible. There are numerous tables from which friction losses in any combination of piping may be calculated, among the most recent of which are those published by the Hydraulic Institute.

Before any new installation is made, the cost of larger-size piping that will result in lower pump pressures should be carefully balanced against the cost of a less expensive pump, a smaller motor, and a saving in horsepower over the expected life of the system. The large piping may cost a little more in the beginning, but the ultimate saving in power will often offset the original cost many times. These facts are particularly true of the handling of extremely viscous fluids, and although most engineers dealing with fluids of this type are conscious of what can be done, it is surprising how many installations are encountered in which considerable savings could have been effected if a little more study had been made initially.

Abrasives. There is one other point that we have not as yet touched, and that is the handling of fluids containing abrasives. Because rotary pumps depend upon close clearances for proper pumping action, the handling of abrasive fluids will usually cause rapid wear. Much progress has been made in the use of harder and more abrasive-resistant materials for the pumping elements, so that a good job can be done in some instances. It cannot be said, however, that performance is always satisfactory when handling fluid laden excessively with abrasive materials. On the whole, rotary pumps should not be used for handling fluids of this character unless shortened pump life and an increased frequency of replacement are acceptable.

Design details. It is virtually impossible to include a discussion of the design details for the many varieties of rotary pumps within the framework of this handbook; therefore, this subsection will be limited to a brief discussion of IMO pump design with some reference to other types when applicable.

Basic construction. The IMO pump, as well as other types and makes of rotary screw pumps, is available in two basic configurations: single- and double-end construction. The double-end construction (see Fig. P-284) is probably the best-known version, as it was by far the most widely used, by many years, because of the relative simplicity and compactness of its design. As pressure requirements were raised, however, the single-end version developed increased usage, until today it accounts for by far the largest portion of total IMO pump annual production (see Fig. P-288).

The general double-end screw-pump construction usually is limited to low- and medium-pressure applications, with 400 lb/in^2 a good practical limit for planning purposes. However, with special design features incorporated, applications up to 800 lb/in^2 can be handled. Double-end pumps generally are employed when large flows are required or very viscous fluids are handled.

There is in use one other principal variation of IMO pump construction that must be mentioned briefly, that is, the four-rotor design having three idlers, which is sometimes used for low-pressure lubricating service. The introduction of the third idler, in effect, makes the pump nonpositive, which gives it additional capability for handling heavily air-laden lubricating oil without cavitation or the related heavy vibration. This design, however, is restricted to very low pressure use because of the resulting increased slip characteristic.

The single-end screw-pump construction (see Fig. P-285) is most often employed for handling low-viscosity fluids at high pressure or hydraulic-type fluids at very high pressure. It is most practical to provide the additional number of moving seals

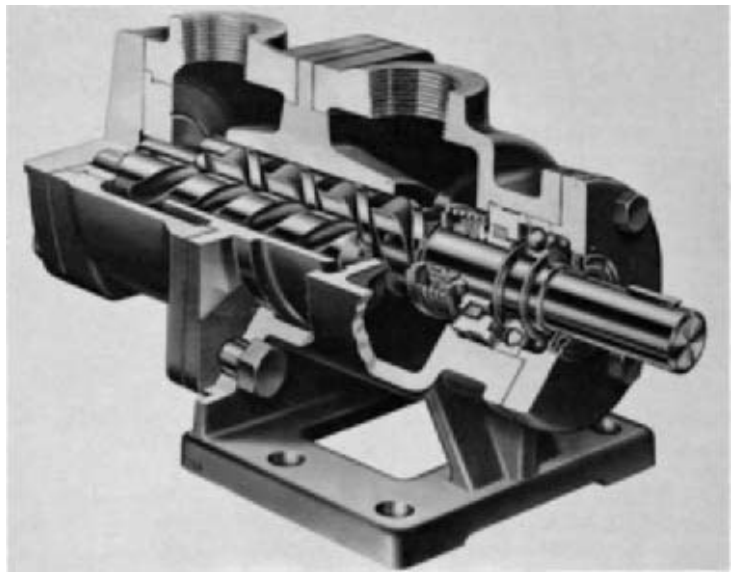


FIG. P-288 Cutaway view of IMO pump. (Source: Demag Delaval.)

or closures between inlet and outlet necessary to handle high pressure in the single-end construction. This is accomplished in IMO pumps by literally stacking a number of medium-pressure single-end pumping elements in series within one pump casing. The single-end construction also offers the best design arrangement for high-production manufacture even though the design itself is more complex than the relatively simple double-end construction.

The double-end type (see Fig. P-284) is basically two opposed single-end pumps or pump elements of the same size with a common power rotor of double-helix design within one casing. As can be seen from the illustration, the fluid normally enters a common inlet, with a split flow going to the outboard ends of the two pumping elements, and is discharged from the middle or center of the pump elements. The two pump elements are, in effect, pumps connected in parallel. The design can also be provided with a reversed flow for low-pressure applications.

Axial balance. Whichever design is employed, means must always be provided to absorb the mechanical and hydraulic axial thrust on the rotors of a screw pump. The double-end design provides the simplest arrangement for accomplishing this, as both the power and the idler types of rotor (see Fig. P-284) are constructed with opposed-thread helices on the same shaft, which provides true axial balancing, both mechanically and hydraulically, since all thrust forces between the opposed pump elements are canceled out.

In single-end designs, special axial balancing arrangements must be employed for both the power and the idler rotors, and in this respect they are thus more complicated than the double-end construction. Mechanical thrust-bearing arrangements (see Fig. P-288) are used for the idlers for 150-lb/in^2 differential pressures and below, while a hydraulic-balance arrangement (see Fig. P-285) is used for pressures above 150 lb/in^2. Here hydraulic balance is accomplished by directing discharge pressure to a bearing area on the inlet end of the idler that is equal and opposite to the area exposed to discharge pressure on the outlet end of the same idler.

Hydraulic balance is provided for the power rotor through the balance piston (see Fig. P-285) mounted on the power rotor between the outlet and seal chambers. This

piston is exposed to discharge pressure on the outlet side and is equal and opposite in area to the exposed area of the power-rotor thread; thus the discharge-pressure hydraulic forces on the rotor threads are canceled out.

Seals. The IMO pump, like most other modern equipment, makes extensive use of mechanical-face seals for shaft sealing. Packing is now used only when absolutely necessary as dictated by the fluid handled. Seal technology has advanced rapidly, with many new materials such as Buna N, neoprene, Viton, and Teflon introduced for elastomers. Ni-Resist, carbon, carbide, and ceramics are now available in addition to the original standby pearlitic cast iron for use in the sealing faces. All this has made the use of packing virtually obsolete.

In all but some small low-pressure series, the IMO pump always has the seal located in a chamber connected to the suction side. To accomplish this in the single-end design in which the outlet is at the shaft end, the aforementioned power-rotor balance piston also serves as a breakdown bushing or flow restrictor between outlet and seal chambers to limit the pressure in the seal chamber. This seal chamber, in turn, is connected to the suction side of the pump through a small internal drilled conduit or through external tubing (see Fig. P-285).

In most cases in which a mechanical seal is used in an IMO pump, an external grease-sealed ball bearing is employed on the power-rotor drive shaft to maintain precise shaft positioning. This ensures long mechanical seal life. This bearing also serves to minimize flexible-coupling-misalignment conditions that can adversely affect the performance of high-speed equipment such as the IMO pump. The use of the ball bearing also provides a means for taking overhung loads, such as from belt drives, on certain models.

Inlet pressures. Standard IMO pumps are normally designed to handle positive inlet pressures up to 40 psig. This limitation of pressure concerns the resulting thrust on the rotors, and design modifications can be made for much higher inlet pressures as required. The double-end design is ideal for adapting to high-inlet-pressure applications because the idlers are in thrust balance at all times and the power rotor can be thrust-balanced by making it double-ended so that both ends are exposed to the inlet pressure identically. The drawback is the need for two seals, but this is not very significant if the high inlet pressure really is important to obtain.

The above double-shaft arrangement also is used when two or more pumps are to be driven in tandem, which is quite advantageous in some applications. Shaft tapers are always used on larger IMO pumps for locating the coupling. The use of this taper helps to protect the mechanical seals and bearings from shock damage that can arise when installing a large coupling on a straight shaft.

Casings. Standard IMO pumps normally are provided with high-grade cast iron for the casing of low- and medium-pressure models. Standard high-pressure pumps employ ductile iron or cast steel for the casings with fabricated steel used for special orders when necessary. Casings also are made suitable for steam jacketing when absolutely necessary for high-viscosity applications; however, the use of heat tracing with either steam coils or electric tape covered with a good insulation blanket is the recommended preference.

IMO pumps can normally be mounted in virtually any position including vertical as well as all horizontal rotations. Double-end designs usually are arranged with opposed side inlet and outlet positions parallel to the foot mounting. Side inlet and top discharge can also be furnished if necessary.

Rotor materials. The rotors and housings of the IMO pump can be made of various types and grades of hardened materials for use in handling corrosive-type fluids as well as those containing some abrasives. One of the popular material combinations

in use in many of the medium- and high-pressure models is nitrided-steel power rotors and induction-hardened ductile-iron idlers with pearlitic gray iron housings.

Most of the rotors of IMO pumps are finish-machined after hardening by thread grinding in order to obtain a high degree of accuracy. Very small and very large rotor sets at the extremes of the size range are finish-thread-milled. Thread forms are controlled very accurately to obtain the proper mating action of all rotor sets as well as to maintain running clearances between rotors to a minimum for limited internal leakage.

Installation and operation. Rotary-pump performance can be improved by following the recommendations on installation and operation given next.

The pump should be placed on a smooth, solid foundation readily accessible for inspection and repair. It is essential that the power shaft and drive shafts be in perfect alignment. IMO practice normally requires a concentricity and parallelism of 0.003 FIR.

The suction pipe should be as short and straight as possible with all joints airtight. There should be no points at which air or entrapped gases may collect. If it is not possible to have the fluid flow to the pump, a foot or check valve should be installed at the end of the suction line or as far from the pump as possible. All piping should be independently supported to avoid strains on the pump casing.

A priming connection should be provided on the suction side and a relief valve set from 5 to 10 percent above the maximum working pressure on the discharge side.

Starting the unit may involve simply opening the pump suction and discharge valves and starting the motor, but it is always better to prime the unit on initial starting. On new installations, the system is full of air that must be removed. If it is not removed, the performance of the unit will be erratic, and in some cases air in the system can prevent the unit from pumping. Priming the pump should preferably consist of filling not only the pump with fluid but as much of the suction line as possible.

The discharge side of the pump should be vented on the initial starting. Venting is especially essential when the suction line is long or the pump is discharging against system pressure upon starting.

If the pump does not discharge after being started, the unit should be shut down immediately. The pump should then be primed and tried again. If it still does not pick up fluid promptly, there may be a leak in the suction pipe or the trouble may be traceable to excessive suction lift from an obstruction, throttled valve, or other causes. Attaching a gauge to the suction pipe at the pump will help find the trouble.

Once the pump is in service, it should continue to operate satisfactorily with practically no attention other than an occasional inspection of the mechanical seal or packing for excessive leakage and a periodic check to be certain alignment is maintained within reasonable limits for prolonged periods.

(*Note:* Although mechanical seals are becoming more widely used, there are some applications in which packing will continue to be preferred, and it is therefore necessary to make some brief comment concerning the proper installation and care of packing. The packing gland should never be set up too tightly. Packing properly used will require some leakage to maintain correct lubrication. The recommended leakage rate is somewhat dependent upon the type of fluid being handled but should never be less than several drops per minute. Excessive gland pressure on the packing causes scoring of shaft and rapid deterioration of the packing itself. The best practice is to keep the gland stud nuts about finger-tight.)

Should the pump develop a noise after satisfactory operation, this usually indicates either excessive suction lift due to cold fluid, air in the fluid, misalignment of the coupling, or, in the case of an old pump, excessive wear.

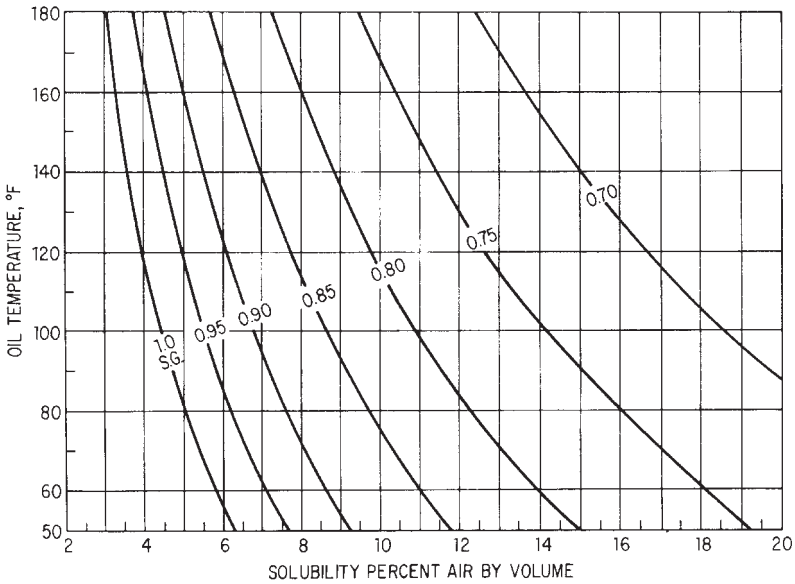


FIG. P-289 Solubility of air in oil. (Source: *Journal of the American Society of Naval Engineers*.)

Whenever the unit is shut down, if the operation of the system permits, both suction and discharge valves should be closed. This is particularly important if the shutdown is to be for an extended period because leakage in the foot valve, if the main supply is below the pump elevation, could drain the fluid from the unit and necessitate repriming as in the initial starting of the system.

Effect of entrained or dissolved gas on performance. A very important factor in rotary-pump applications is the amount of entrained and dissolved air or gas in the fluid handled. This is especially true if the suction pressure is below atmospheric. Such air or gas is generally neglected since rotary pumps are of the displacement type and hence are self-priming. If the entrained or dissolved air and gases are a large percentage of the volume handled and if their effect is neglected, there may be noise and vibration, loss of liquid capacity, and pressure pulsations.

The amount of entrained air or gas is extremely variable, depending upon the viscosity, the type of liquid, and the time and manner of agitation that it may have received.

There is little information available covering the solubility of air and other gases in liquids, especially all those handled by rotary pumps. About 1930, Dr. C. S. Cargoe of the National Bureau of Standards developed the following formula on the basis of available literature data to show the solubility of air at atmospheric pressure in oils, both crude and refined, and in other organic liquids:

$$\log_{10} A = \frac{792}{t + 460} - 4 \log_{10} \text{sg} - 0.4$$

where A = dissolved air, in^3/gal
t = temperature, °F
sg = specific gravity of the liquid

This equation is plotted as Fig. P-289, as taken from a paper on rotary pumps by R. J. Sweeney in the February 1943 issue of the *Journal of the American Society of Naval Engineers*. The equation and curve should be considered as approximate only,

since some liquids have a higher affinity for air and gases. For example, gasoline at atmospheric pressure will dissolve as much as 20 percent of air by volume.

This actual displacement is measured in terms of volume of fluid pumped and will be the same whether it is a liquid, a gas, or a mixture of both as long as the fluid can get to and fill the pump moving voids.

If the fluid contains 5 percent entrained gas by volume and no dissolved gas and the suction pressure is atmospheric, the mixture is then 95 percent liquid and 5 percent gas. This mixture fills up the moving voids on the inlet side, but 5 percent of the space is filled with gas and the remainder with liquid. Therefore, in terms of the amount of liquid handled, the output is reduced directly by the amount of gas present, or 5 percent. The liquid displacement as a function of the theoretical displacement when the suction pressure is atmospheric then becomes

$$D' = D(1 - E)$$

where D = theoretical displacement
D' = liquid displacement
E = percent entrained gas by volume at atmospheric pressure, divided by 100

Assume that the fluid handled is a liquid mixture containing 5 percent entrained gas by volume at atmospheric pressure and no dissolved gas, but with the inlet pressure at the pump p_i in psia that is below atmospheric. The entrained gas will increase in volume as it reaches the pump in direct ratio to the absolute pressures. The new mixture will have a greater percentage of gas present, and the portion of theoretical displacement available to handle liquid becomes

$$D' = \frac{D(1 - E)}{(1 - E) + Ep/p_i}$$

when p = atmospheric pressure, psia
p_i = inlet pressure, psia

Note that p_i depends upon the vapor pressure of the liquid, the static lift, and the friction and entrance losses to the pump.

In the above equation, if the atmospheric pressure is 14.7 psia, the pump-inlet pressure 5 psia, and the vapor pressure very low, the liquid displacement is 86.6 percent of the theoretical.

If dissolved gases in liquids are considered, the effect on the liquid-displacement reduction is the same as that due to entrained gases, since in the latter case the dissolved gases come out of solution when the pressure is lowered. For example, assume a liquid free of entrained gas but containing gas in solution at atmospheric pressure and the pumping temperature. As long as the inlet pressure at the pump does not go below atmospheric pressure and the temperature does not rise, gas will not come out of solution. If pressure below atmospheric does exist at the pump inlet, gas will evolve and expand to the pressure existing. This will have the same effect as entrained gas taking up available displacement capacity and will reduce the liquid displacement accordingly. The liquid displacement then will be

$$D' = \frac{D}{1 + y(p - p_i)/p_i}$$

where the symbols have the meanings given previously and y is the percentage of dissolved gas by volume at pressure p divided by 100. If the operating conditions are 9 percent of dissolved gas at 14.7 psia with a pump-inlet pressure of 5 psia, the liquid displacement will be 85.2 percent of the theoretical displacement.

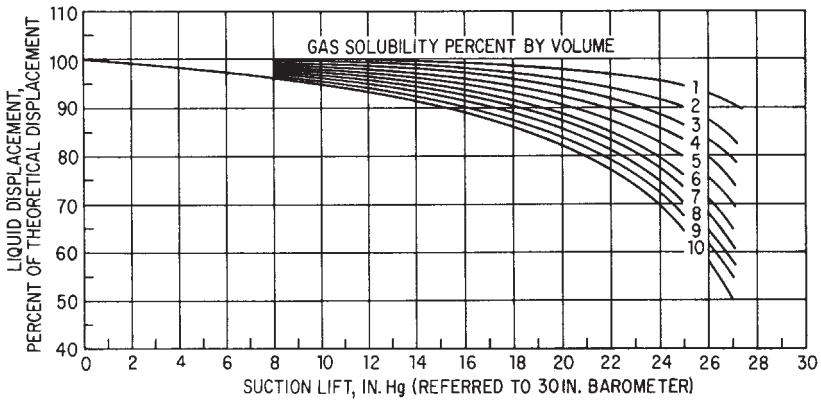


FIG. P-290 Effect of dissolved gas on liquid displacement. (Source: Demag Delaval.)

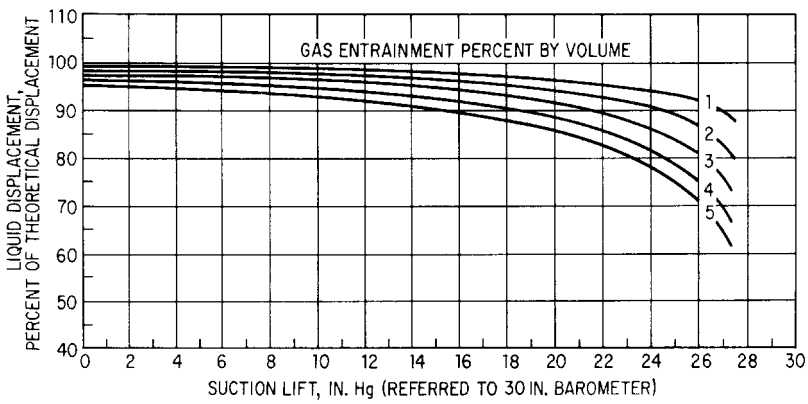


FIG. P-291 Effect on entrained gas on liquid displacement. (Source: Demag Delaval.)

If both entrained and dissolved gases are considered as existing in the material to be pumped, the liquid displacement becomes

$$D' = \frac{Dp_i(1-E)}{(1-E)[p_i + y(p-p_i)] + Ep}$$

where the symbols have the meanings given above. For operating conditions of 5 percent entrained gas, 9 percent dissolved gas at 14.7 psia, and a pump-inlet pressure of 5 psia, the liquid displacement is 75.2 percent of the theoretical. Figure P-290 shows graphically the reduction in liquid displacement as a function of pump-inlet pressure, expressed in terms of suction lift, for different amounts of dissolved gas, neglecting slip.

Figure P-291 shows the reduction in liquid displacement as a function of pump-inlet pressure, expressed as suction lift, for different amounts of entrained air only, neglecting slip. From this figure it may be noted that a very small air leak can cause a large reduction in liquid displacement, especially if the suction lift is high.

From these few examples and curves it would appear that the problem of entrained and dissolved gases could be cared for by providing ample margins in pump capacity. Unfortunately, capacity reductions from the causes mentioned are attended by other and usually more serious difficulties.

The operation of a rotary pump is such that as rotation progresses, closures that fill and discharge in succession are formed. If the fluid pumped is compressible, such as a mixture of oil and air, the volume within each closure is reduced as it comes in contact with the discharge pressure. This produces pressure pulsations, the intensity and frequency of which depend upon the discharge pressure, the number of closures formed per revolution, and the speed of rotation. Under some conditions the pressure pulsations are of high magnitude and can cause damage to piping and fittings or even the pump, and they will almost certainly be accompanied by undesirable noise.

The amount of dissolved air or gas may be reduced by lowering the suction lift. This may often be controlled by pump location, suction-pipe diameter, and piping arrangement.

Many factors are associated with the amount of entrained air that can exist in a given installation. It is prevalent in systems in which the liquid is handled repeatedly and during each cycle is exposed to or mechanically agitated in air. Unfortunately in many cases the system is such that air entrainment cannot be entirely eliminated, as in the lubrication system of a reduction gear. Considerable work has been done by oil companies on foam dispersion, and while it has been recommended that special oils that are inhibited against oxidation and corrosion be used, all agree that the best cure is to remove or reduce the cause of foaming, namely, air entrainment.

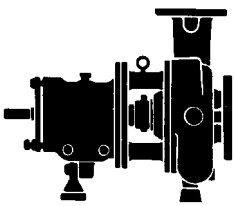
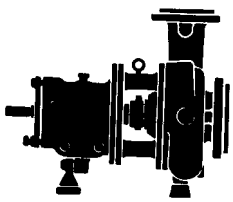
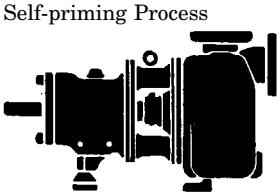
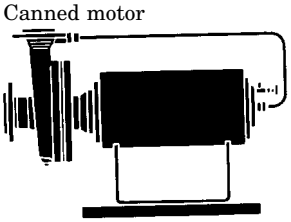
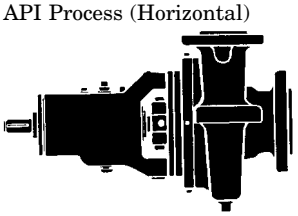
Even though air entrainment cannot be entirely eliminated, in many cases it is possible, by adhering to the following rules, to reduce it and its ill effects on rotary-pump performance.

1. Keep liquid velocity low in the suction pipe to reduce turbulence and pressure loss. Use large and well-rounded suction bell to reduce entrance loss.
2. Keep suction lift low. If possible, locate the pump to provide positive head on the inlet.
3. Locate the suction piping within a reservoir to obtain maximum submergence.
4. Submerge all return lines particularly from bypass and relief valves, and locate them away from the suction.
5. Keep the circulation rate low, and avoid all unnecessary circulation of the fluid.
6. Do not exceed rated manifold pressures on machinery lubricating systems since the increased flow through sprays and bearings increases the circulation rate.
7. Heat the fluid when practical to reduce viscosity and as an expedient to drive off entrained air. Fluids of high viscosity will entrain and retrain more air than fluids of low viscosity.
8. Avoid all air leaks no matter how small.
9. Provide ample vents; exhauster fans to draw off air and vapors have been used with good results.
10. Centrifuging will break a foam and remove foreign matter suspended in the oil, which promotes foaming.
11. Use a variable-speed drive for the pump to permit an adjustment of pump capacity to suit the flow requirements of the machinery.

Typical Pump Types and Applications

Table P-33 is a general table of typical pump types, typical services, and typical ratings that covers most common applications that a process engineer would ever see.

TABLE P-33 General Table: Pump Types and Applications*

Pump Type	Typical Services	Typical Ratings
ANSI Process 	Corrosive/abrasive liquids, slurries, and solids, high temperature, general purpose pumping process and transfer.	Q to 4500 gpm (1022 m^3/h) H to 730 ft (222 m) T to 700°F (371°C) H to 375 psig (2586 kPa)
Nonmetallic Chemical Process 	Severe corrosives.	Q to 800 gpm (182 m^3/h) H to 490 ft (149 m) T to 300°F (150°C) P to 225 psig (1550 kPa)
Self-priming Process 	Corrosive/abrasive liquids, slurries, and suspensions, high temperature, industrial sump, mine dewatering, tank car unloading, bilge water removal, filter systems, chemical transfer.	Q to 1500 gpm (340 m^3/h) H to 375 ft (114 m) T to 500°F (260°C) Suction lifts to 25 ft (7.6 m)
In-line Process 	Process, transfer and general service. Corrosive and volatile liquids. High temperature services.	Q to 1500 gpm (340 m^3/h) H to 700 ft (207 m) T to 500°F (260°C) P to 375 psig (2586 kPa)
Canned motor 	Zero leakage services: toxic liquids, refrigerants, liquefied gas, high temperature heat transfer, explosive liquids, liquids sensitive to atmosphere, carcinogenic and other hazardous services.	Q to 2500 gpm (568 m^3/h) H to 1400 ft (427 m) T to 700°F (371°C) P to 450 psig (3103 kPa)
API Process (Horizontal) 	High temperature and high pressure services, offsite, transfer, heat transfer liquids.	Q to 7500 gpm (1700 m^3/h) H to 1100 ft (335 m) T to 800°F (427°C) P to 870 psig (6000 kPa)

* SOURCE: Goulds Pumps, USA.

TABLE P-33 General Table: Pump Types and Applications (*Continued*)

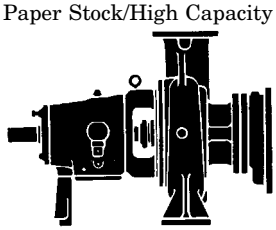
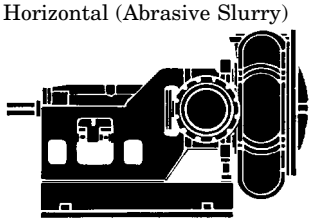
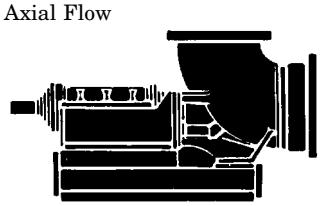
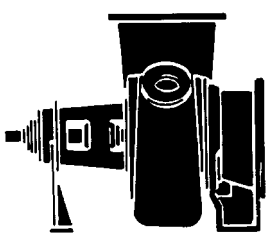
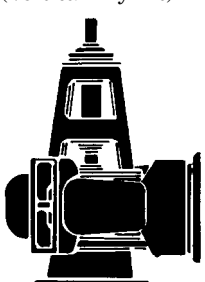
Pump Type	Typical Services	Typical Ratings
API Process (In-line) 	Petrochemical, chemical, refining, offsite, gasoline plants, natural gas processing, general services.	Q to 7500 gpm (1700 m^3/h) H to 750 ft (229 m) T to 650°F (343°C) P to 595 psig (4100 kPa)
Paper Stock/High Capacity Process 	Paper stock, solids and fibrous/stringy materials, slurries, corrosive/abrasive process liquids.	Q to 28,000 gpm (6360 m^3/h) H to 350 ft (107 m) T to 450°F (232°C) P to 285 psig (1965 kPa)
	Medium consistency (8 to 14%) paper stock.	Q to 1800 TPD (1650 MTPD) H to 400 ft (125 m) T to 250°F (120°C)
Horizontal (Abrasive Slurry) 	Corrosive/abrasive services. Coal, fly ash, mill scale, bottom ash, slag, sand/gravel, mine slurries. Large solids.	Q to 10,000 gpm (2273 m^3/h) H to 350 ft (107 m/stage) T to 400°F (204°C) P to 300 psig (2068 kPa) Spherical solids to 4 in (102 mm)
Axial Flow 	Continuous circulation of corrosive/abrasive solutions, slurries and process wastes. Evaporator and crystallizer, reactor circulation, sewage sludge recirculation.	Q to 200,000 gpm (35,000 m^3/h) H to 30 ft (9 m) T to 350°F (180°C) P to 150 psig (1034 kPa) Solids to 9 in (228 mm)
Large Solids Handling (Horizontal) 	Pumps for extra demanding municipal and industrial services; large pulpy and fibrous solids, sewage, abrasives.	Q to 100,000 gpm (22,700 m^3/h) H to 240 ft (73 m) T to 202°F (43°C) P to 300 psig (2065 kPa) Solids to 10 in (254 mm)
(Vertical Dry Pit) 		

TABLE P-33 General Table: Pump Types and Applications (*Continued*)

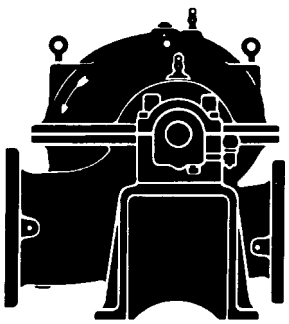
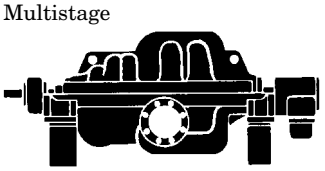
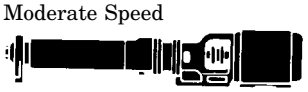
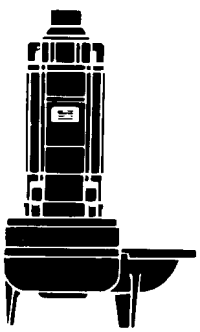
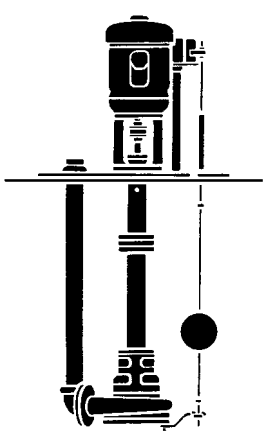
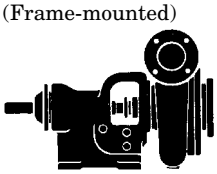
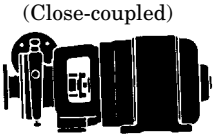
Pump Type	Typical Services	Typical Ratings
Double Suction 	Cooling tower, raw water supply, booster service, primary and secondary cleaner, fan pump, cooling water, high lift, low lift, bilge and ballast, fire pumps, river water, brine, sea water, pipelines, crude.	*Q* to 72,000 gpm (16,300 m^3/h) *H* to 570 ft (174 m) *T* to 350°F (177°C) *P* to 275 psig (1896 kPa)
Multistage 	Refinery, pipeline, boiler feed, descaling, crude oil charging, mine pumping, water works . . . other high pressure services. Water, cogeneration, reverse osmosis, booster service, boiler feed, shower service. Boiler feed, mine dewatering and other services requiring moderately high heads.	*Q* to 3740 gpm (850 m^3/h) *H* to 6000 ft (1824 m) *T* to 375°F (190°C) *P* to 2400 psig (16,546 kPa)
Low Flow/High Head MultiStage Moderate Speed 	Reverse osmosis descaling, high pressure cleaning, process water transfer, hydraulic systems, spraying systems, pressure boosters for high-rise buildings, all low flow applications where efficiency is critical	*Q* to 280 gpm (64 m^3/h) *H* to 2600 ft (792 m) *T* to 400°F (204°C) *P* to 1100 psig (7584 kPa)
Submersible • Wastewater • Solids Handling • Slurry 	Flood and pollution control, liquid transfer, sewage and waste removal, mine dewatering, sump draining. Large stringy or pulpy solids. Abrasive slurries.	*Q* to 4000 gpm (910 m^3/h) *H* to 210 ft (65 m) *T* to 140°F (60°C) Solids to 2 in (50 mm)
Vertical Submerged (Submerged Bearing and Cantilever) • Process • Solids Handling • Slurry 	Industrial process, sump drainage, corrosives, pollution control, molten salts, sewage lift, wastewater treatment, extremely corrosive abrasive slurries, large or fibrous solids.	*Q* to 7500 gpm (1703 m^3/h) *H* to 310 ft (95 m) *T* to 450°F (232°C) Solids to 10 in (254 mm)

TABLE P-33 General Table: Pump Types and Applications (*Continued*)

Pump Type	Typical Services	Typical Ratings
Vertical Turbine 	Irrigation, fire pumps, service water, deep well, municipal water supply mine dewatering, cooling water, seawater and river water intake, process, utility circulating, condenser circulating, ash sluice, booster, petroleum/refiner, boiler feed, condensate, cryogenics, bilge, fuel oil transfer, tanker and barge unloading.	Q to 150,500 gpm (34,065 m^3/h) H to 3500 ft (1070 m) T to 700°F (371°C)
General Service (Frame-mounted)  (Close-coupled) 	Close-coupled and frame-mounted pumps for water circulation, booster, OEM packages, irrigation, chemical process, transfer, and general purpose pumping.	Q to 2100 gpm (477 m^3/h) H to 400 ft (122 m) T to 300°F (149°C)

Common Pump Applications in the Process Industry*

Technical information on process, double suction between bearings, vertically suspended double casing, overhung vertical in-line integral bearing frame flexibly coupled pump, multistage pumps, and heavy-duty double-case multistage pumps. See Table P-34.

API 610 process pumps (CAP8 Type)

Application ranges. The pumps designated CAP8 are designed for pumping applications covering the full range of refinery services, including water, gasoline, propanes, and light products, as well as hard-to-handle crude oil and fractionator bottoms. Typical applications include

- Refinery use
- Petrochemical plants
- Gas processing

* Source: Sulzer Pumps, USA, a division of Sulzer-Burckhardt, amalgamating what used to be Bingham Pumps.

TABLE P-34 CAP8 Pumps

Operating Data	SI Units	U.S. Units
Sizes	25 to 300 mm	1 to 12 in
Capacities	to 1700 m^3/h	to 7500 US gpm
Heads	to 365 m	to 1200 ft
Pressures	to 50 bar	to 735 psi
Temperatures	to 425°C	to 800°F
Speeds	to 3600 rpm	to 3600 rpm

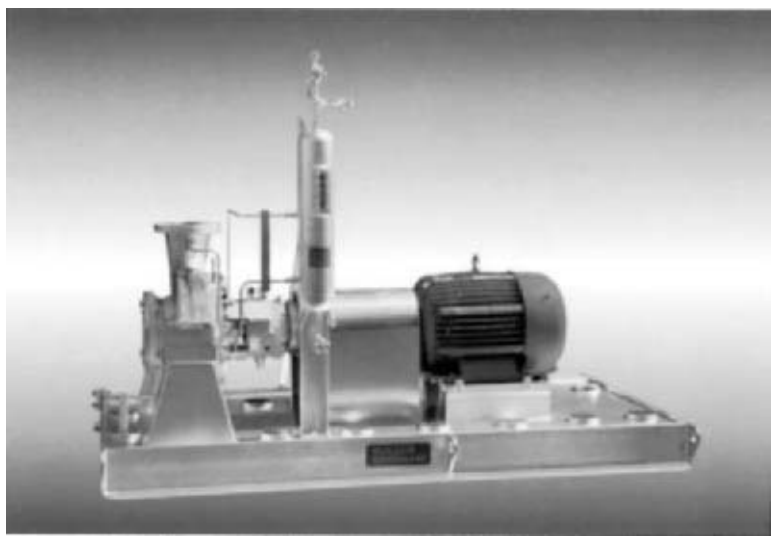


FIG. P-292 API 610 process pump. (Source: Sulzer Pumps.)

- Coal processing
- Offshore installation

Design. They are in full compliance with API 610 8th edition specification. Features include horizontal, overhung, radially split, centerline mounted, centerline discharge, and heavy duty process pumps with enclosed-type single-suction impellers. They also are equipped with.

- End suction and top discharge nozzles
- Fully confined compression gaskets between case, case cover and seal gland; pump shaft designed for cartridge-type mechanical seals
- Seal chamber sized to accommodate all single, tandem, double, and cartidge seal configurations; complies fully with API 682 specification Table 1
- Connections for seal flush and cooling water piping are available; heavy, drain-type, fabricated steel baseplates are standard

See Figs. P-292 through P-299 and Table P-35.

Materials. Standardized API 610 material classes: S-4, S-6, C-6, A-8. Other material combinations are available.

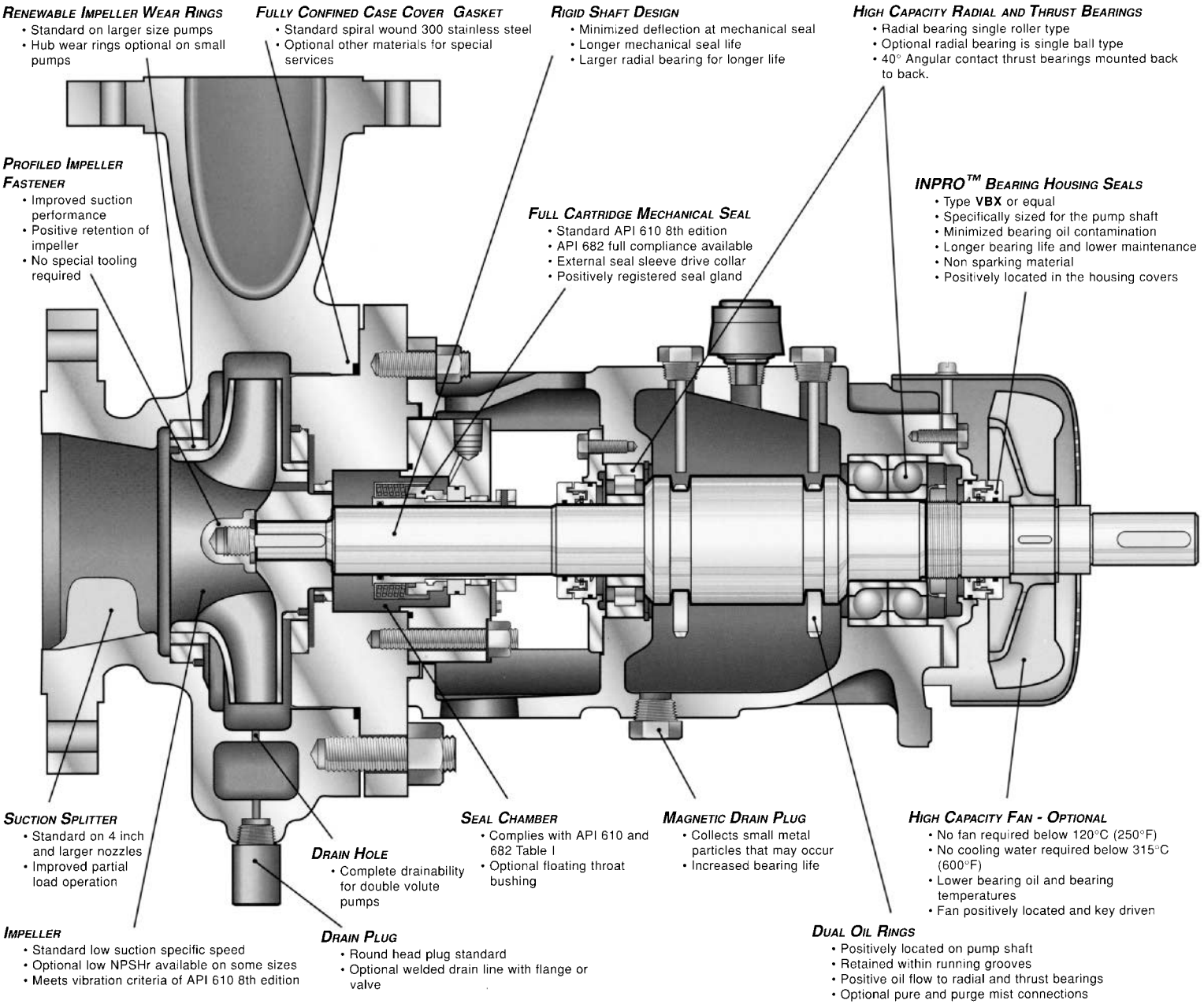


FIG. P-293 Pump main features (CAP8 type). (Source: Sulzer Pumps.)

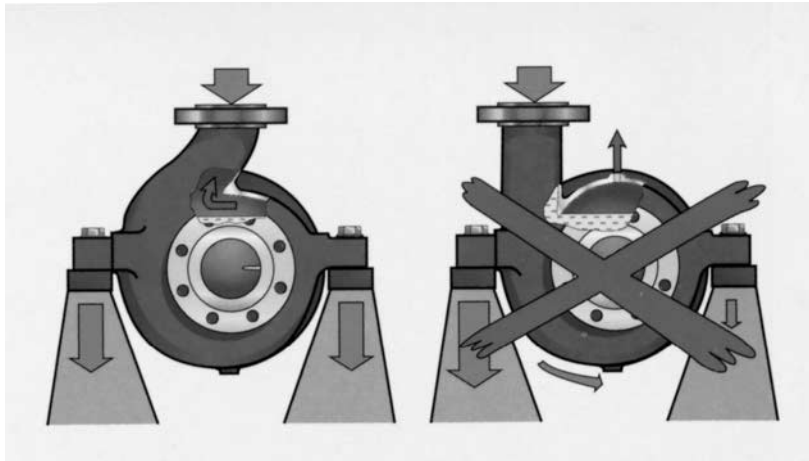


FIG. P-294 Centerline discharge and mounting. Nozzle loads transferred equally to both case feet; withstand nozzle loadings in excess of API 610 8th edition requirements; self-venting. (Source: Sulzer Pumps.)

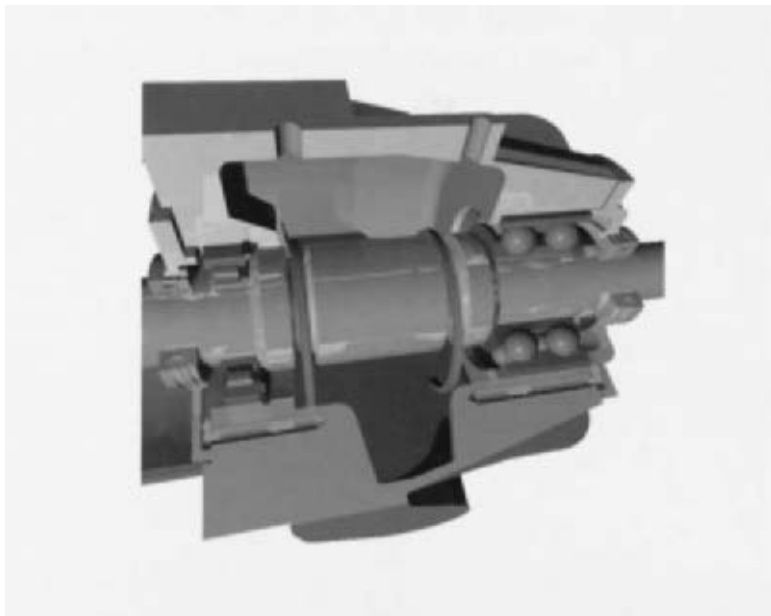


FIG. P-295 Efficient lube oil circulation system. The CAP8 bearing bracket is provided with the two rotation-independent lube oil systems, one for each bearing. The oil ring arrangement distributes cool oil from the bearing bracket sump into drilled grooves. The oil flows through holes to the back side of each bearing, then circulates through the bearings back to the oil sump. Result: Efficient lubrication and cooling of each radial and thrust bearing. (Source: Sulzer Pumps.)

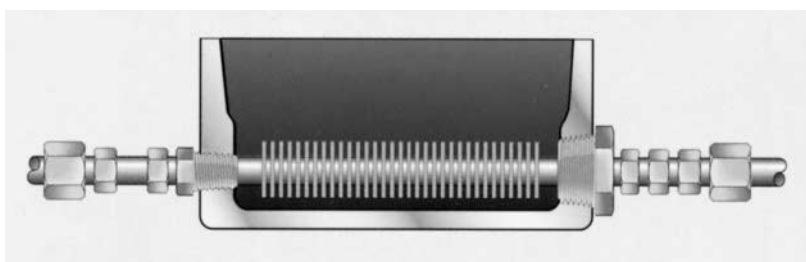


FIG. P-296 Fin tube oil cooler. Optional cooling for over 315°C (600°F). (Source: Sulzer Pumps.)

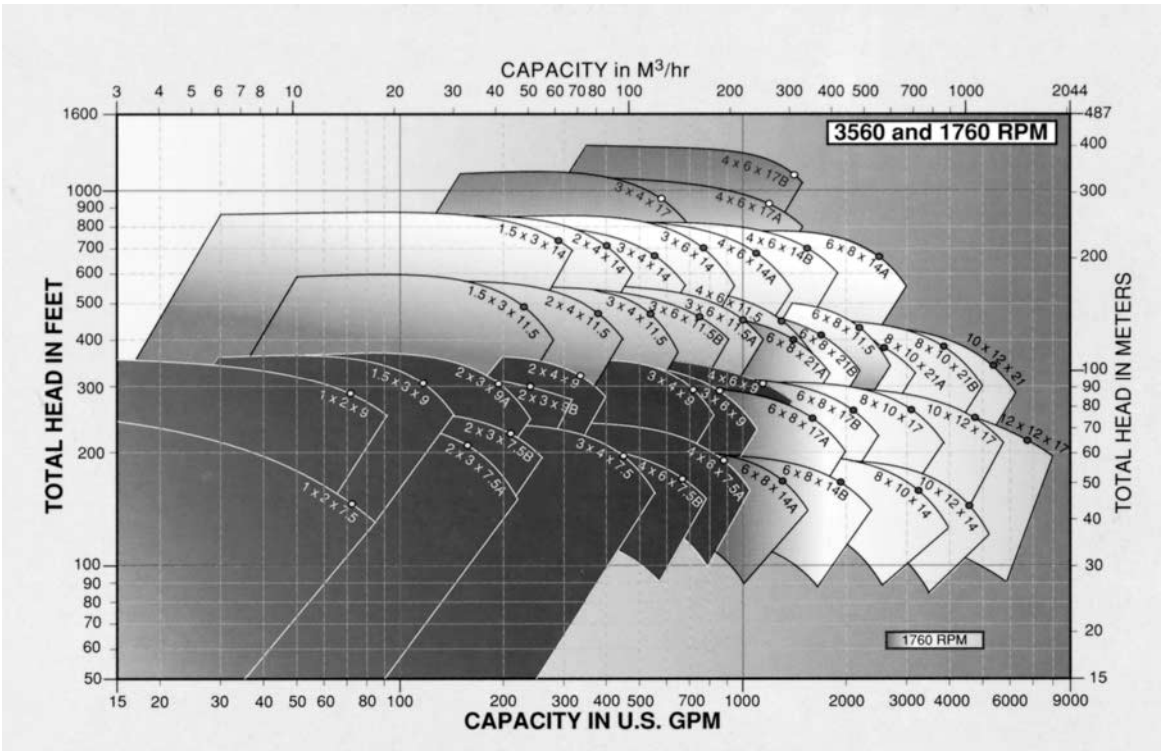


FIG. P-297 Hydraulic range chart. (Source: Sulzer Pumps.)

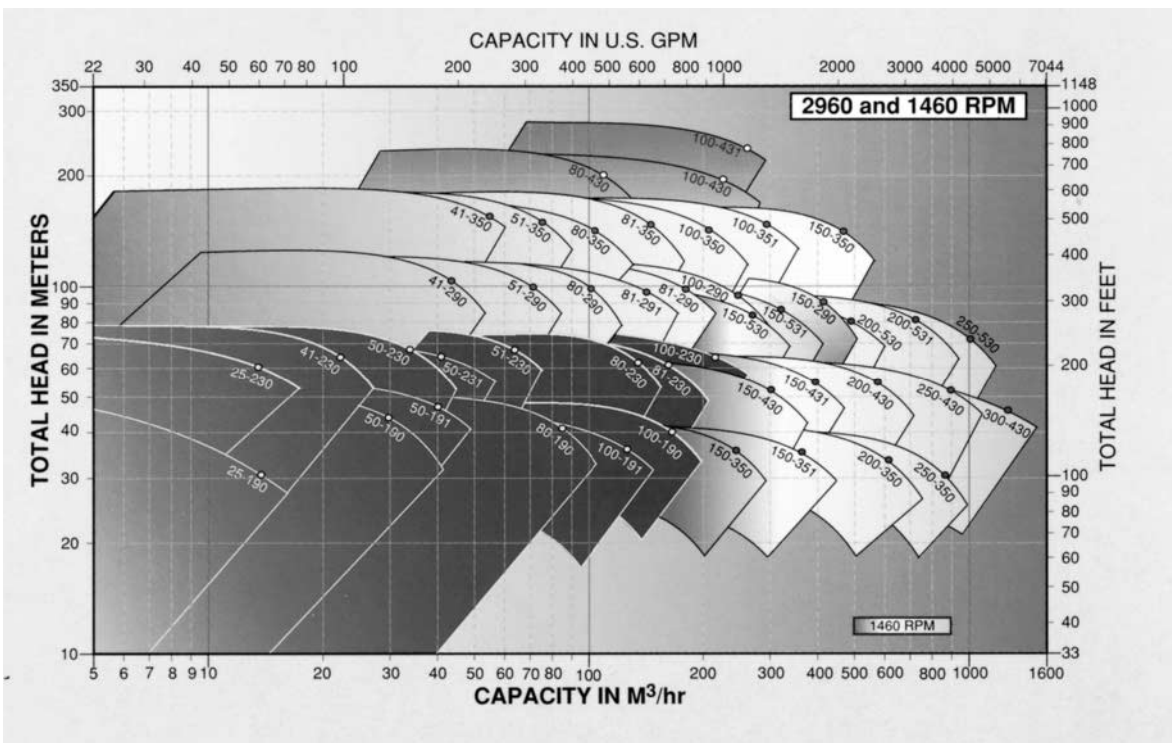


FIG. P-298 Hydraulic range chart. (Source: Sulzer Pumps.)

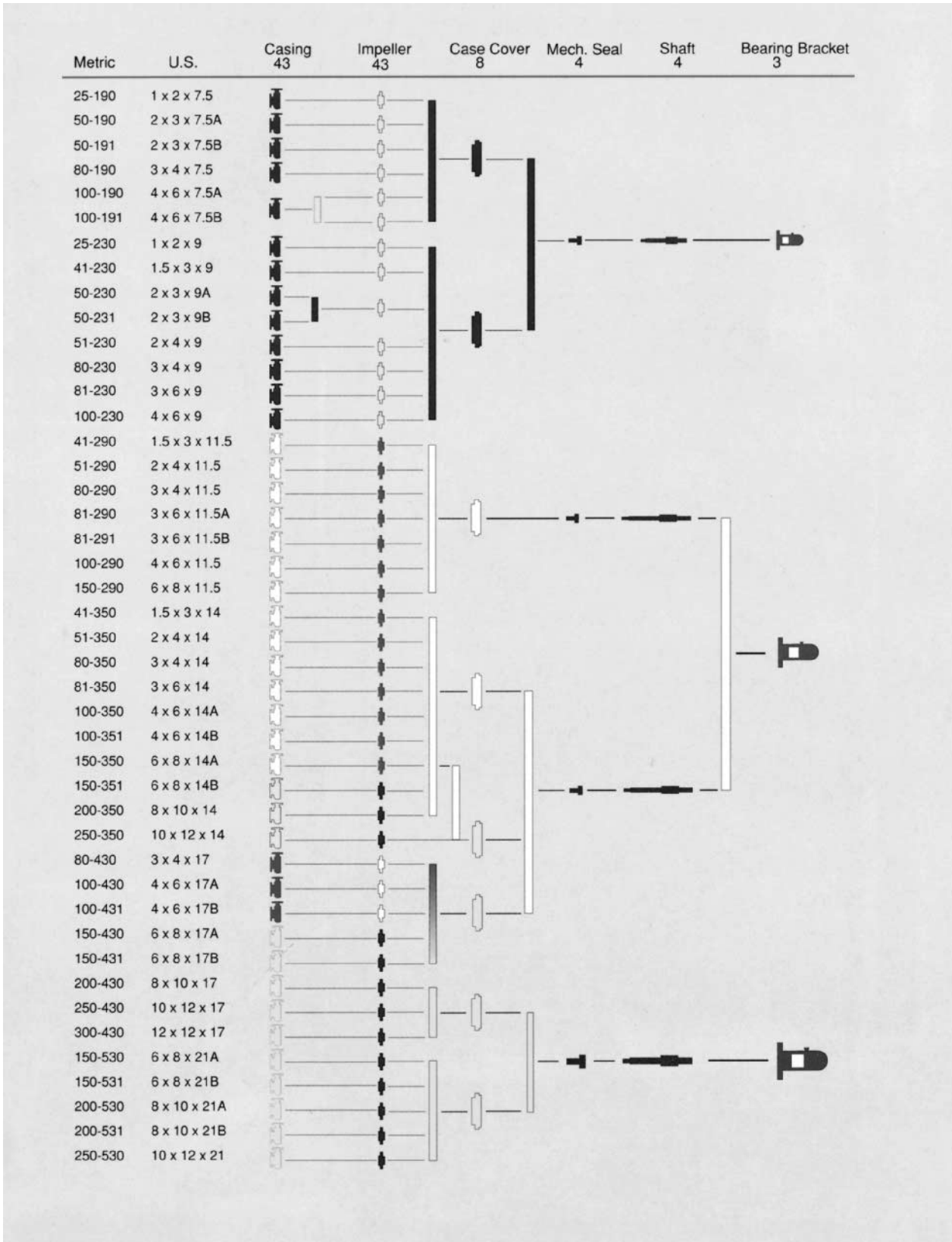


FIG. P-299 Maximum interchangeability minimizes spare parts inventory requirements. (Source: Sulzer Pumps.)

TABLE P-35 CAP8 Design Features and Benefits

	Design Features	Advantages	Benefit
Pump casing	■ Volute casing centerline mounted with robust feet	■ Operation at elevated temperatures	■ Reduced alignment and maintenance problems
	■ Vertical centerline mounted discharge nozzle	■ Self-venting nozzle ■ Loads transferred to pump pedestals equally	■ No special vent piping or startup procedures ■ Simpler piping layout and installation
	■ Double volute (4 in and larger)	■ Reduced vibration at off-peak efficiency conditions	■ Increase life for bearings and seals
	■ Case bolting unified inch or metric	■ Pump may be specified and installed throughout the world	■ Similar maintenance practices, training and parts interchangeability
Impeller	■ Nss < 11,000 Standard (S < 214)	■ Reduced vibration at low flows	■ Longer seal and bearing life and flexibility of operation
	■ Nss < 14,000 Optional (S < 271)	■ Reduced NPSHr	■ Lower vessel height and greater NPSH margins
	■ Enclosed design	■ High efficiencies and no impeller adjustment	■ Lower operating cost and reduced maintenance
	■ Wide performance range	■ Broad selection of hydraulics to match application requirements	■ Reduced operating cost and single vendor for all applications
Shaft	■ Optimized shaft diameter to overhang ratio	■ Lower shaft deflection at seal	■ Longer seal life and lower cost
	■ Positive shouldering of bearing and seals	■ Ensures proper assembly	■ Reduced maintenance time and longer operating periods
Bearing bracket	■ Large single piece finned bearing assembly	■ Positive housing to case cover alignment ■ Large surface area for cooling	■ Reduced maintenance cost and longer operating periods ■ Longer bearing life
	■ Self-contained oiling system with internal circulation	■ No external oil system required and adequate flow of oil throughout housing	■ Longer bearing life and cooler operating bearings
	■ Pure and purge mist oil systems optional	■ Ease of converting one system to another	■ Reduced cost in converting from one system to another
	■ INPRO labyrinth shaft seals	■ Minimum external contamination of oil	■ Longer bearing life and extended operating periods
	■ Fan cooling standard for temperatures of 120 to 315°C (250 to 600°F)	■ No external coolant required for bearings	■ Reduced installation cost for utilities as well as operating cost
	■ Finned tube water cooling optional	■ Large surface area for positive heat transfer and temperatures to 425°C (800°F)	■ Extended range of applications for the product

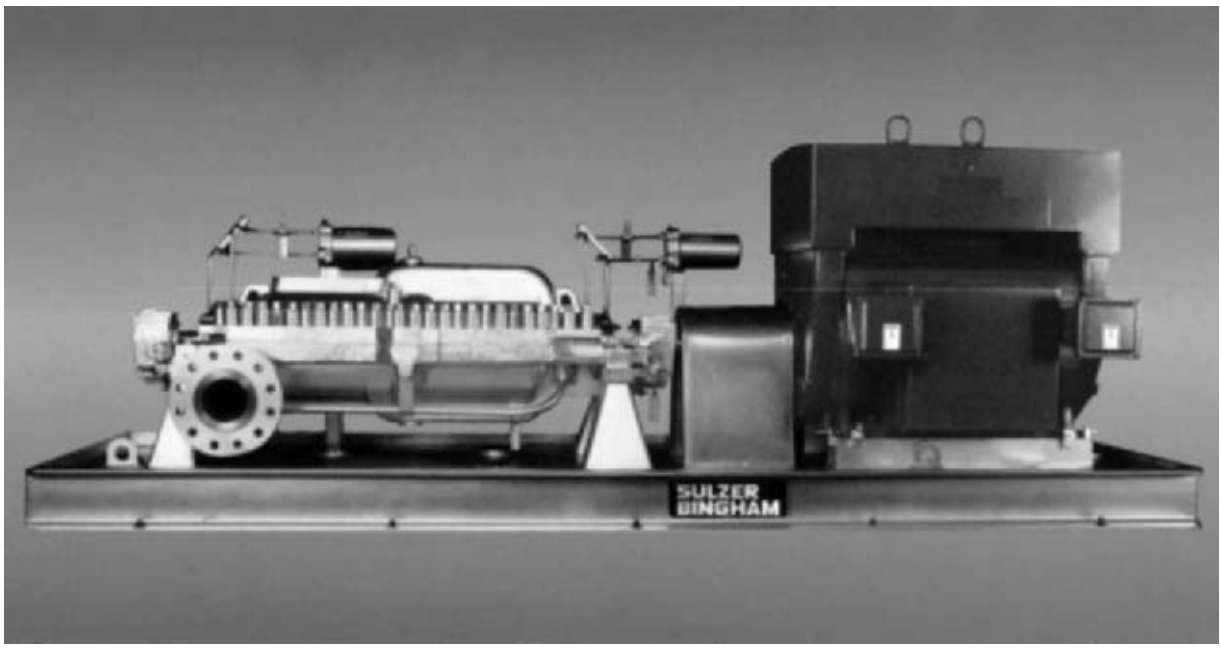


FIG. P-300 Multistage pump assembly. (Source: Sulzer Pumps.)

Multistage pumps

Design features. These include:

- Full compliance to API 610 8th edition including API 682 seal chamber dimensions
- Preengineered packages for specific applications
- Fan-cooled bearing assemblies
- Simplified rotating element installation
- Designed to accommodate a large range of wear parts, including nonmetallic

See Figs. P-300 through P-305.

Materials. Standardized API material classes: S-6, C-6, A-8. Other material combinations are available.

Applications

- Hydrocarbon processing, high-pressure processing
- Oil production, water injection, transfer
- Pipeline, crude and product
- Power, boiler feedwater

See Tables P-36 and P-37 as well as Fig. P-306.

API 610 overhung vertical in-line integral bearing frame flexibly coupled pump (CVA8 type)

Application ranges. As previously seen, this genetic designation of pumps is designed for pumping applications covering the full range of refinery services,

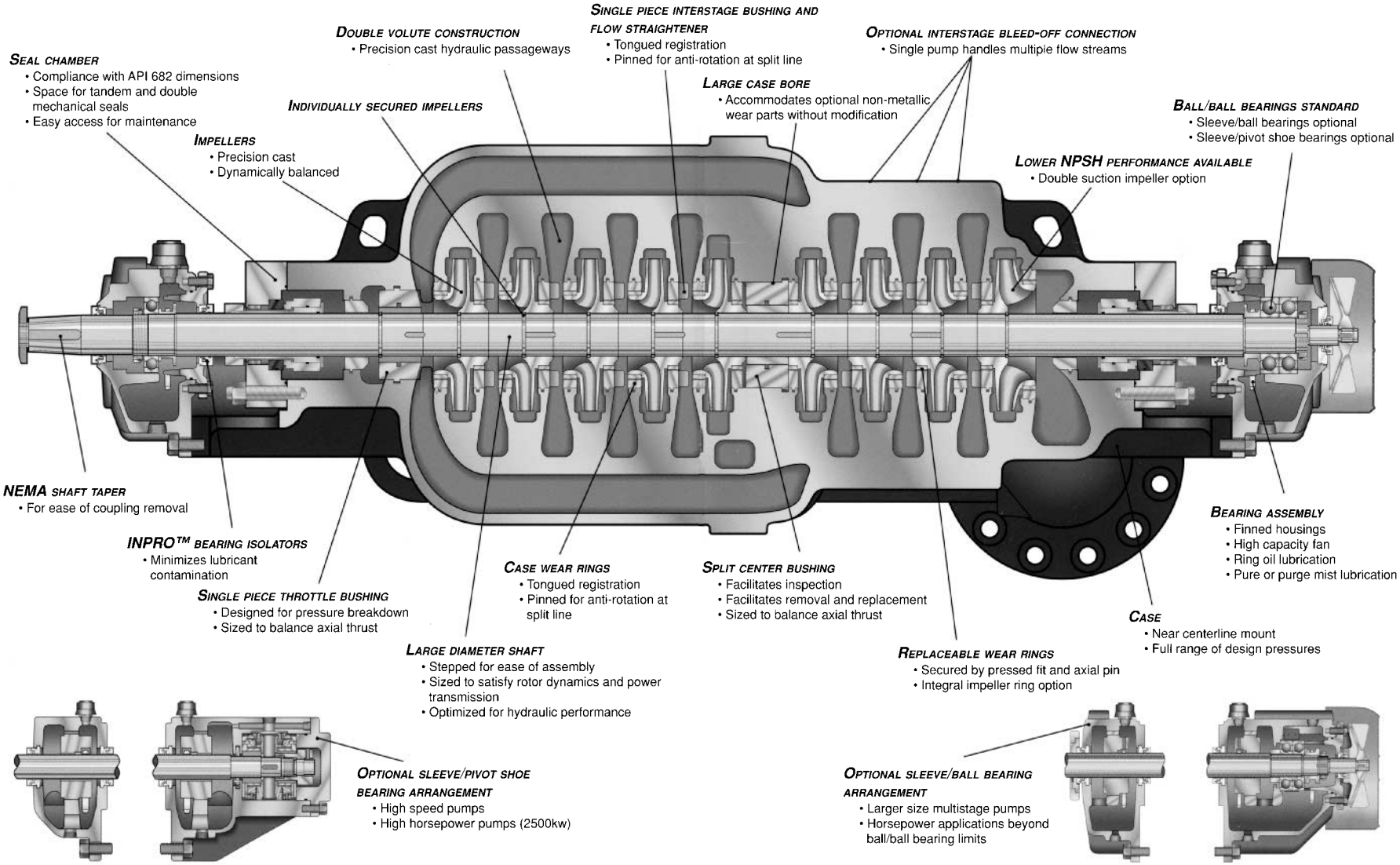


FIG. P-301 Multistage pump main features. (Source: Sulzer Pumps.)

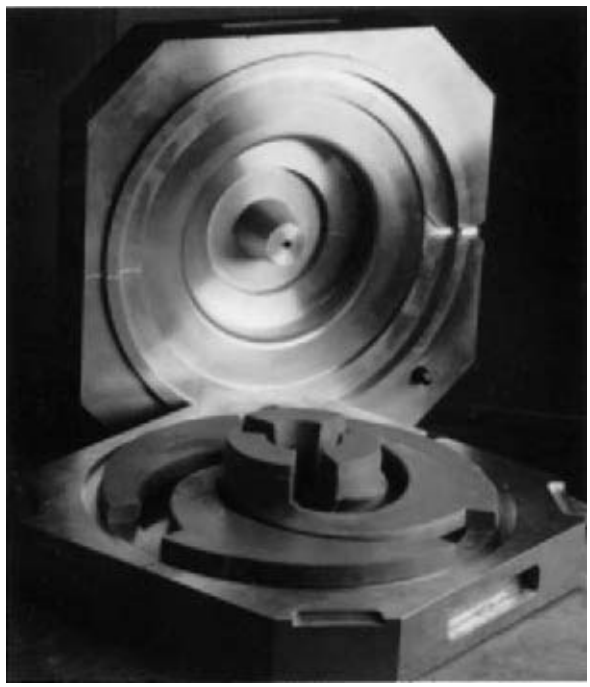


FIG. P-302 Impeller investment tooling: improves casting integrity and dimensional stability while achieving maximum efficiency. (Source: Sulzer Pumps.)



FIG. P-303 Boarded case patterns and shell core crossovers: for repeatable quality casting and surface finish. (Source: Sulzer Pumps.)



FIG. P-304 Coordinate measuring machine: measures the dimensional accuracy of castings and confirms the performance of the MC-60 machining center. (Source: Sulzer Pumps.)



FIG. P-305 MC-60 machining center: machines pump casings with consistent precision in less time than previous generations of manufacturing equipment. (Source: Sulzer Pumps.)

TABLE P-36 Multistage Pumps

Design Criteria	SI Units	U.S. Units
Discharge size	80 to 400 mm	3 to 16 in
Impeller diameter	230 to 660 mm	9 to 26 in
Capacity	to 3200 m^3/h	to 14,000 gpm
Head	to 3000 m	to 9850 ft
Pressure	to 310 bar	to 4500 psi
Temperature	to 204°C	to 400°F
Speed	to 6500 rpm	to 6500 rpm

including water, gasoline, propanes, and light products, as well as hard-to-handle crude oil. Applications include

- Refinery use
- Petrochemical plants
- Gas processing
- Coal processing
- Offshore installation

Design features. These include:

- In-line suction and discharge nozzles
- Full compliance with API 610 8th edition specifications
- Overhung, vertical in-line, integral bearing frame, flexibly coupled pump
- Seal chamber designed to accommodate all single and dual cartridge seal configurations; complies fully with API 682 specification Table 1
- Back pullout device

See Figs. P-307 through P-313 and Table P-38.

Materials. Standardized API 610 material classes: S-4, S-6, C-6 and A-8. Other material combinations are available.

CVAR rigid coupled pump

See Fig. P-314 and Table P-39.

API 610 with rigid or flexible coupling

See Figs. P-315 through P-321 and Tables P-40 through P-42.

API 610 double suction between bearings process pump CD8 type

Application ranges. These pumps (designated CD8) are designed for pumping applications covering the full range of refinery services, including water, gasoline, propanes, and light products, as well as hard-to-handle crude oil and fractionator bottoms. Applications include:

- Refinery
- Petrochemical plants
- Gas processing

TABLE P-37 Multistage Pump Design Features and Benefits

	Features	Advantages	Benefits
Case	■ Large bore	■ Accommodates nonmetallic wear parts	■ Increased efficiencies, nongalling
	■ Full range of design pressures	■ Satisfies many applications	■ Case appropriate for application
	■ Established hydraulic designs	■ Optimum match of operating requirements to the pump	■ Lower operating cost
	■ State-of-the-art pattern equipment	■ High-quality castings	■ Predictable production and performance
	■ Accessible fasteners for internal parts	■ Ease of assembly and disassembly	■ Lower maintenance cost
	■ Preengineered bleed-off connection at optimal location	■ Handles multiple requirements	■ Eliminates additional pump and reduces capital cost
	■ Axially split	■ Ease of disassembly and inspection	■ Lower maintenance cost
	■ Opposed impeller design	■ Minimizes axial thrust	■ Optimum bearing life
	■ Double volute design	■ Minimizes radial thrust	■ Optimum seal and bearing life
Impeller	■ Full range of hydraulics	■ Higher head per stage and higher efficiency	■ Lower operating cost
	■ Precision investment pattern equipment	■ Repeatable high-quality castings with a smooth surface finish	■ On time delivery, predictable and repeatable performance
	■ Optional impeller/rotor balance to 4 W/n	■ Reduces vibration	■ Improved mean time between maintenance
	■ Optional integral wear ring	■ Reduced number of parts	■ Reduced maintenance cost
Shaft and seal chamber	■ Bearing bracket enlarged and extended	■ Accommodates double and tandem cartridge seals with ease of access for maintenance	■ Lower maintenance cost
	■ API 682 compliance	■ Accommodates double and tandem cartridge seals with ease of access for maintenance	■ Reduced seal inventories
	■ NEMA tapered shaft extension	■ Easy coupling removal	■ Lower maintenance cost
Bearing housing	■ INPRO™ bearing isolators	■ Minimizes lubricant contamination	■ Promotes longer bearing life
	■ Finned housing	■ Increased heat transfer area	■ Eliminates the requirement for costly water or product cooling
	■ Optional fan cooling	■ Improved convective heat dissipation	■ Eliminates the requirement for costly water or product cooling
	■ Optional fin tube cooler	■ Additional cooling efficiency	■ Increased life for bearings
	■ Ring oil lubrication	■ Self-contained lubrication design	■ Reduced installation and maintenance cost
Baseplate	■ Standardized designs for most sizes	■ Predictable and repeatable installation details	■ Shorter delivery lead times
	■ Optional job specific baseplates	■ Accommodates high nozzle loads with minimum shaft deflection	■ Increased operating reliability
	■ Optional job specific baseplates	■ Preengineered installation details ■ Reduced stress concentration on foundation pad ■ Complies with API 610 8th edition	■ Shorter delivery lead times ■ Increased operating reliability ■ Increased operating reliability

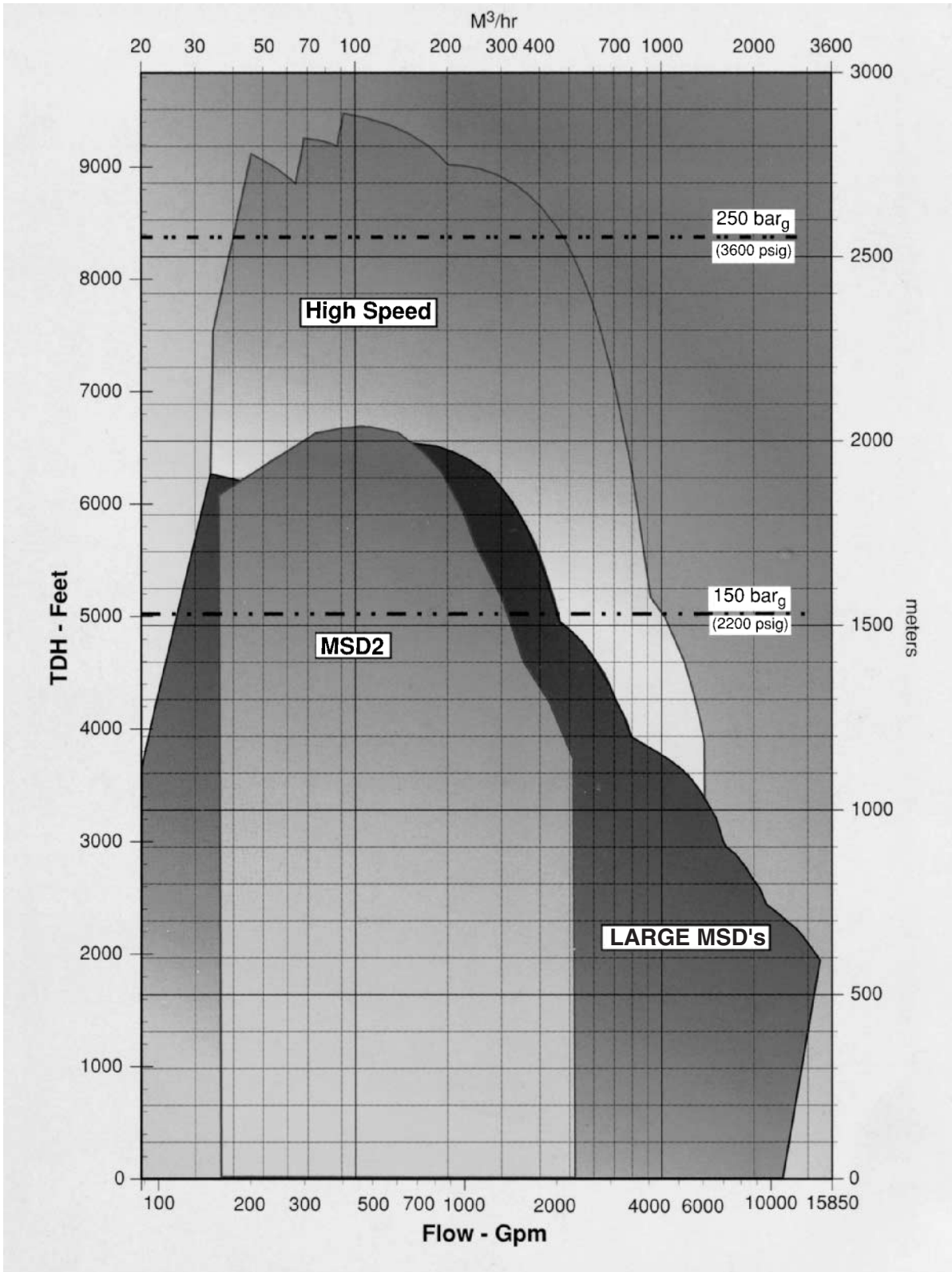


FIG. P-306 Hydraulic range chart: multistage pumps. (Source: Sulzer Pumps.)

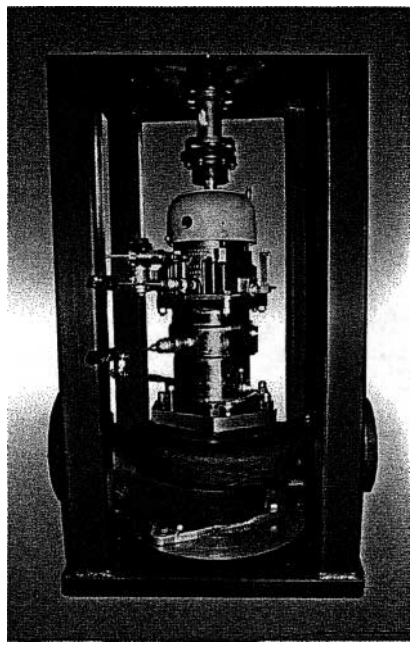


FIG. P-307 Vertical inline pump, type CVA8. (Source: Sulzer Pumps.)

- Coal processing
- Offshore installation

Design features. These include

- Full compliance with API 610 8th edition specifications
- Horizontal, between bearings, radially split, centerline mounted, top suction, top discharge, heavy duty process pump with enclosed type double entry (suction) impeller
- Fully confined gaskets between case, cover, and seal gland plate
- Seal chamber designed to accommodate all single, tandem, dual and cartridge seal configurations; complies fully with API 682 specification Table 1.

Some newer manufacturing techniques used in making these pumps are discussed in Figs. P-302 through P-305.

See Tables P-43 and P-44 as well as Figs. P-322 through P-332.

Materials. Standardized API 610 material classes: S-4, S-6, C-6 and A-8. Other material combinations are available.

Heavy duty double case multistage pumps

This information source designates its radially split, double-volute, double-case, horizontal multistage pump designed for high-pressure, high-speed, high-temperature services requiring reliability and long service life as "Type CP."

The CP's rotating element is housed in a horizontally split inner case, which is itself contained in a cast or forged outer barrel. This design provides for easy maintenance and element removal without piping disturbance.

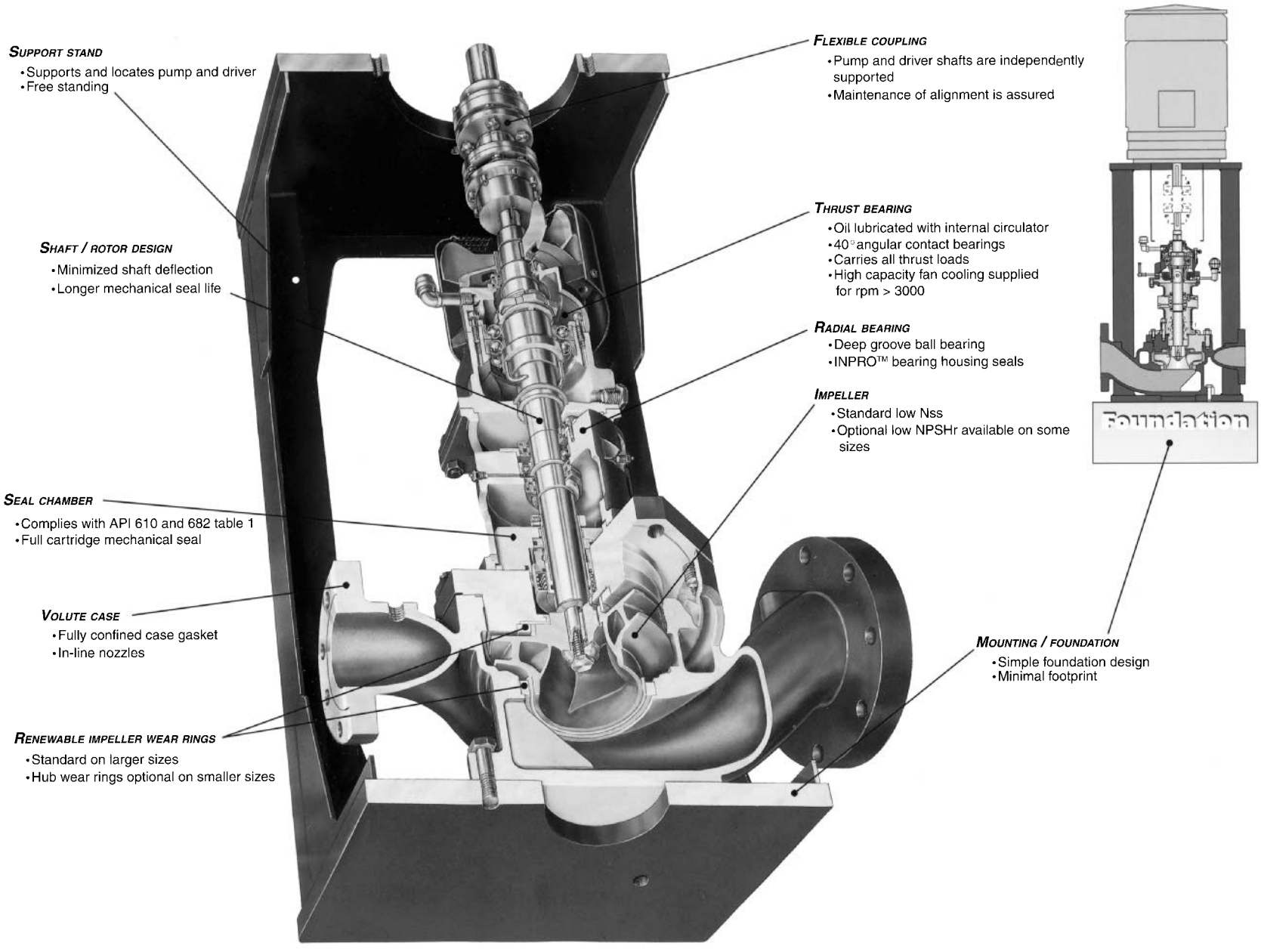


FIG. P-308 Design features, type CVA8. (Source: Sulzer Pumps.)

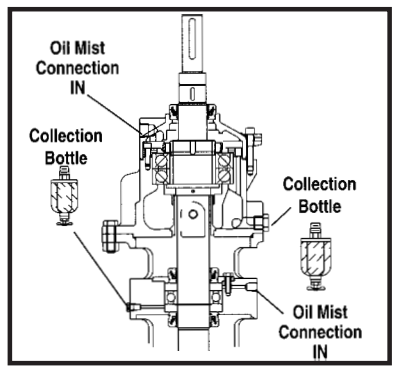


FIG. P-309 Oil mist (CVA8 option). Pure oil mist option; complete with connections and oil drain reservoir. (Source: Sulzer Pumps.)

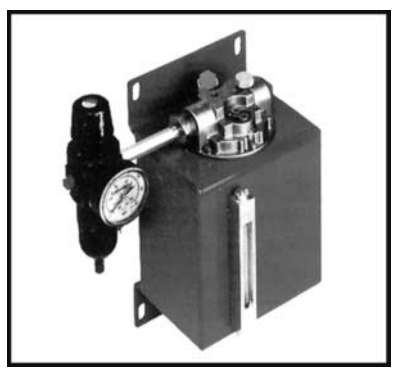


FIG. P-310 Oil mist system (CVA8 option). Stand-alone oil mist generator; integrally mounted; ready for plant air hookup. (Source: Sulzer Pumps.)

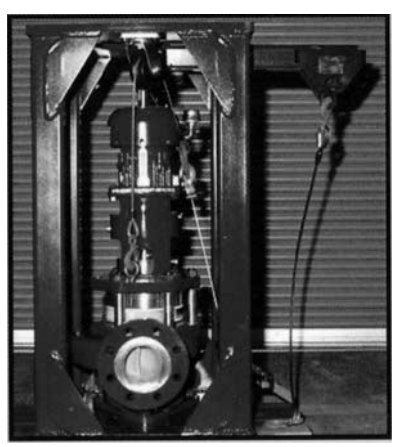


FIG. P-311 Back pull-out tooling (CVA8 option). Available for all pump sizes; ease of in-situ maintenance. (Source: Sulzer Pumps.)

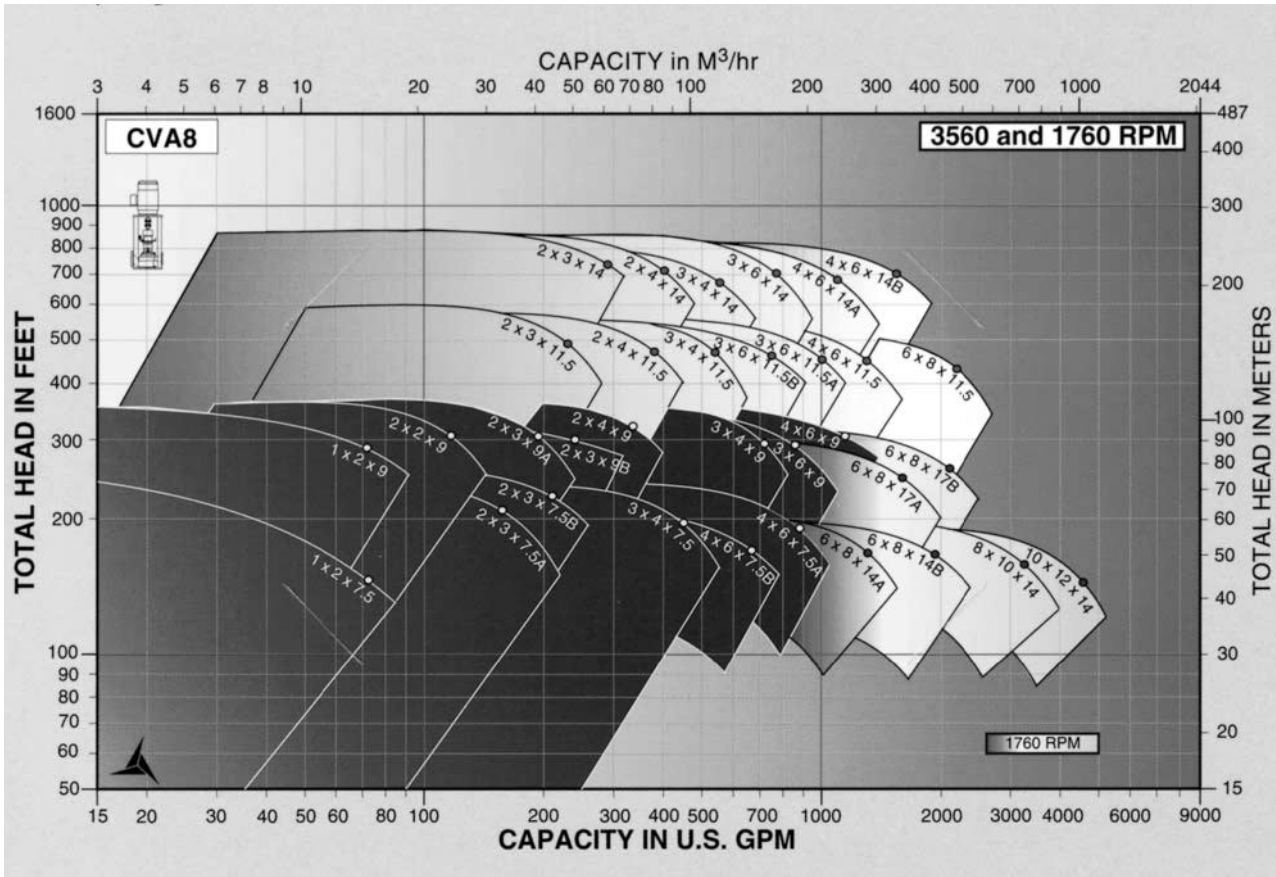


FIG. P-312 Hydraulic range chart (CVA8 type). (Source: Sulzer Pumps.)

Suction and discharge nozzles are normally positioned at top centerline, but can be rotated to meet specific application requirements. Also, CP can be supplied with a double-suction first-stage for low NPSH. The opposed impeller and double volute inner case minimize axial thrust and control radial balance.

CP is used principally for specialized refinery and industrial services, and for boiler feed applications in power plants.

See Figs. P-333 and P-334.

General specifications

Capacities to 15,000 gpm (3400 m^3/h)

Heads to suit

Speeds to 7200 rpm

Temperatures to 800°F (425°C)

Pressure (hydrotest) to 6000 psi (410 bar)

Applications. Petroleum:

- Refinery and petrochemical services
- Production services

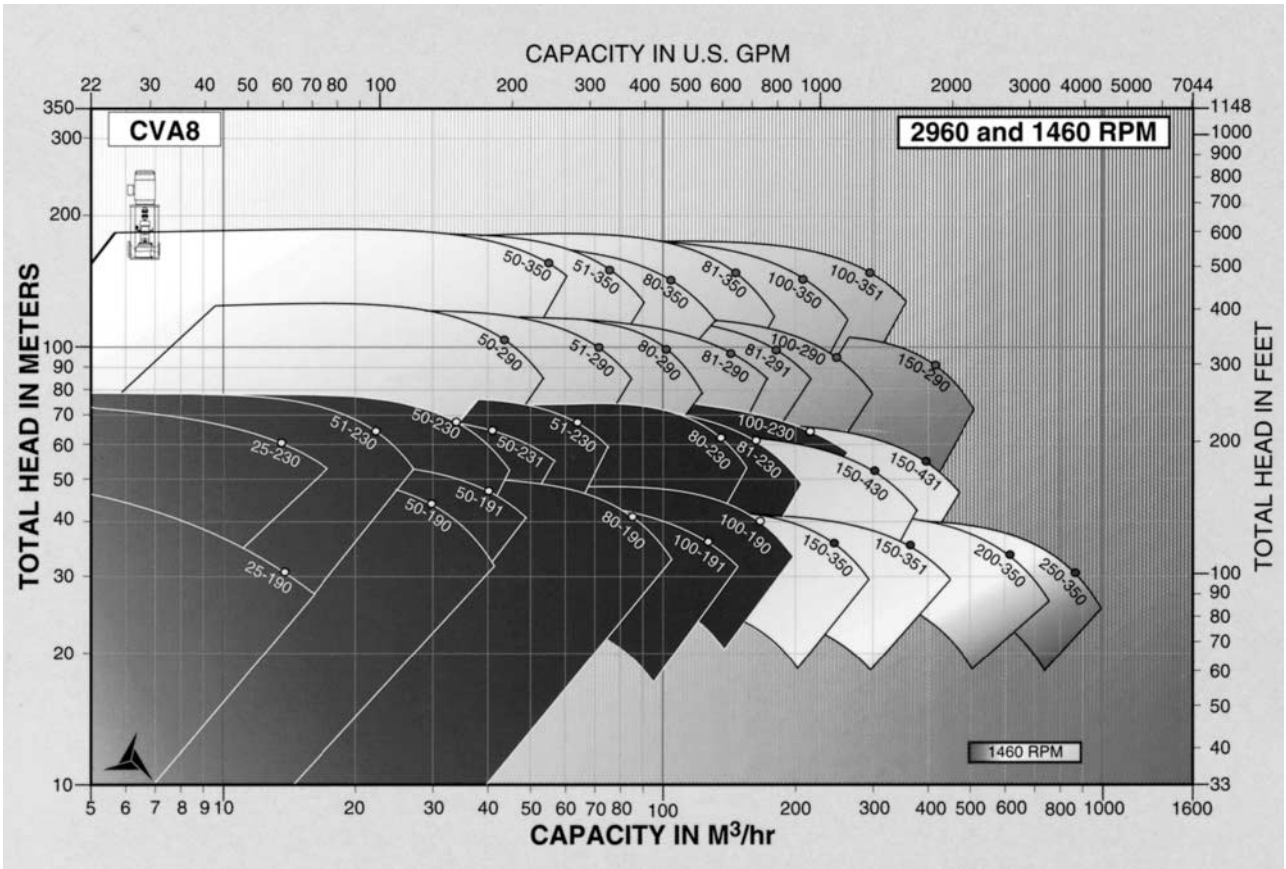


FIG. P-313 Hydraulic range chart (CVA8 type). (Source: Sulzer Pumps.)

TABLE P-38 CVA8 Pumps

Operating Data	SI Units	U.S. Units
Discharge sizes	25 to 250 mm	1 to 10 in
Capacities	to 1070 m^3/h	to 4700 gpm
Heads	to 220 m	to 720 ft
Pressures	to 50 bar	to 735 psi
Temperatures	to 230°C	to 450°F
Speeds	to 3600 rpm	to 3600 rpm

Power:

- Boiler feed

Options that can be supplied. These typically include

- Double-suction first stage
- Suction/discharge nozzle orientation
- Bearing arrangements and lube systems

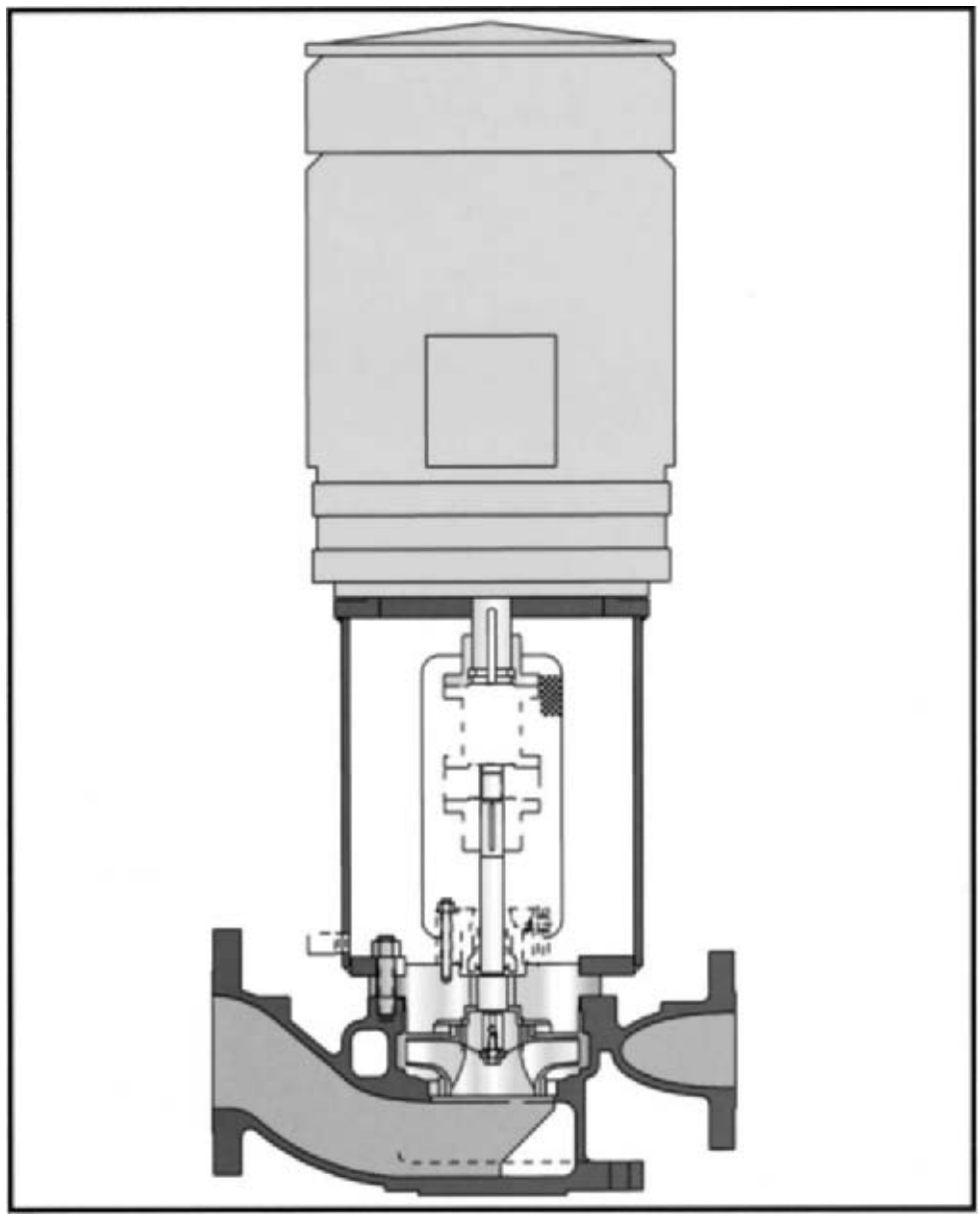


FIG. P-314 Offered as a low-cost alternative to the horizontal overhung and integral thrust bearing frame flexible coupled vertical in-line. Includes equivalent hydraulic performances and many mechanical features required by API services. Includes four-piece rigid coupling design that transmits thrust load from the pump to the driver. Freestanding vertical in-line configuration; API 682 Table 1 seal chamber; interchangeable components with CAP8 and CVA8; rigid four-piece coupling. (Source: Sulzer Pumps.)

- Seals and piping
- Stuffingbox covers
- Cooling jackets
- Custom baseplates for special drivers
- Metallurgy combinations

TABLE P-39 CVA8 Features, Functions, and Benefits

	Features	Functions	Benefits
Design	■ Vertical in-line assembly	■ Allows installation in pipeline with minimal footprint	■ Simplified piping arrangement and foundation design
	■ Freestanding	■ Supports all equipment	■ Minimal foundation preparation ■ Nongrouted
Pressure casing	■ Volute case design capable of withstanding two times API 610 nozzle loads ■ In-line suction and discharge nozzles	■ Supports typical piping installation ■ Arranged for installation in pipe	■ Simplified piping arrangement
	■ Double volute (4 in and larger)	■ Reduced vibration at peak efficiency conditions	■ Increased life for bearings and seals
	■ Fully confined controlled compression gasket	■ Positive location and compression	■ Simplified assembly ■ Improved sealing reliability ■ Reduced emissions
	■ Heavy duty construction with 1/8 in (3 mm) corrosion allowance	■ Pressure rating equivalent to ANSI B16.5 Cl 300 designed per ASME Section VIII Division 1	■ Long service life ■ Standard pressure class rating
Impeller	■ Nss < 11,000 (S < 214)	■ Reduced vibration at low flows	■ Long bearing and seal life and flexibility of operation
	■ Nss < 14,000 (S < 271)	■ Reduced NPSHr	■ Lower vessel height and greater NPSH margins
	■ Enclosed design adjustment	■ High efficiencies and no impeller-reduced maintenance	■ Lower operating cost and reduced maintenance
	■ Wide performance range match application requirements	■ Broad selection of hydraulics to single vendor for all applications	■ Reduced operating cost and single vendor for all applications
Shaft	■ Optimized shaft diameter to overhang ratio	■ Lower shaft deflection at seal	■ Longer seal life and lower maintenance cost
	■ Positive shouldering of bearing	■ Assures proper assembly	■ Reduced maintenance time and longer operating periods
Seal chamber	■ Seal chamber designed to API 682 Table 1 dimensions	■ Allows installation of API 682 seals ■ Cartridge seals	■ Simplified assembly ■ Improved sealing reliability ■ Reduced emissions ■ Standard seals interchangeable between CAP8 and CVAR
Bearing assembly	■ Self-contained oil-lubricated angular contact thrust bearing	■ All loads accommodated by the pump	■ Allows standard driver selection
	■ Grease lubricated radial bearing	■ Carries transmitted radial loads and controls shaft orbit	■ Low maintenance and improved seal life
	■ Oil mist lubrication	■ Clean cool lubrication	■ Improved bearing life
	■ INPRO™ shaft seals	■ Minimum external contamination of oil	■ Longer bearing life and extended operating periods
	■ Fan cooling standard >3,000 rpm or >250 °F	■ No external coolant required for bearings	■ Reduced installation cost for utilities and operating costs

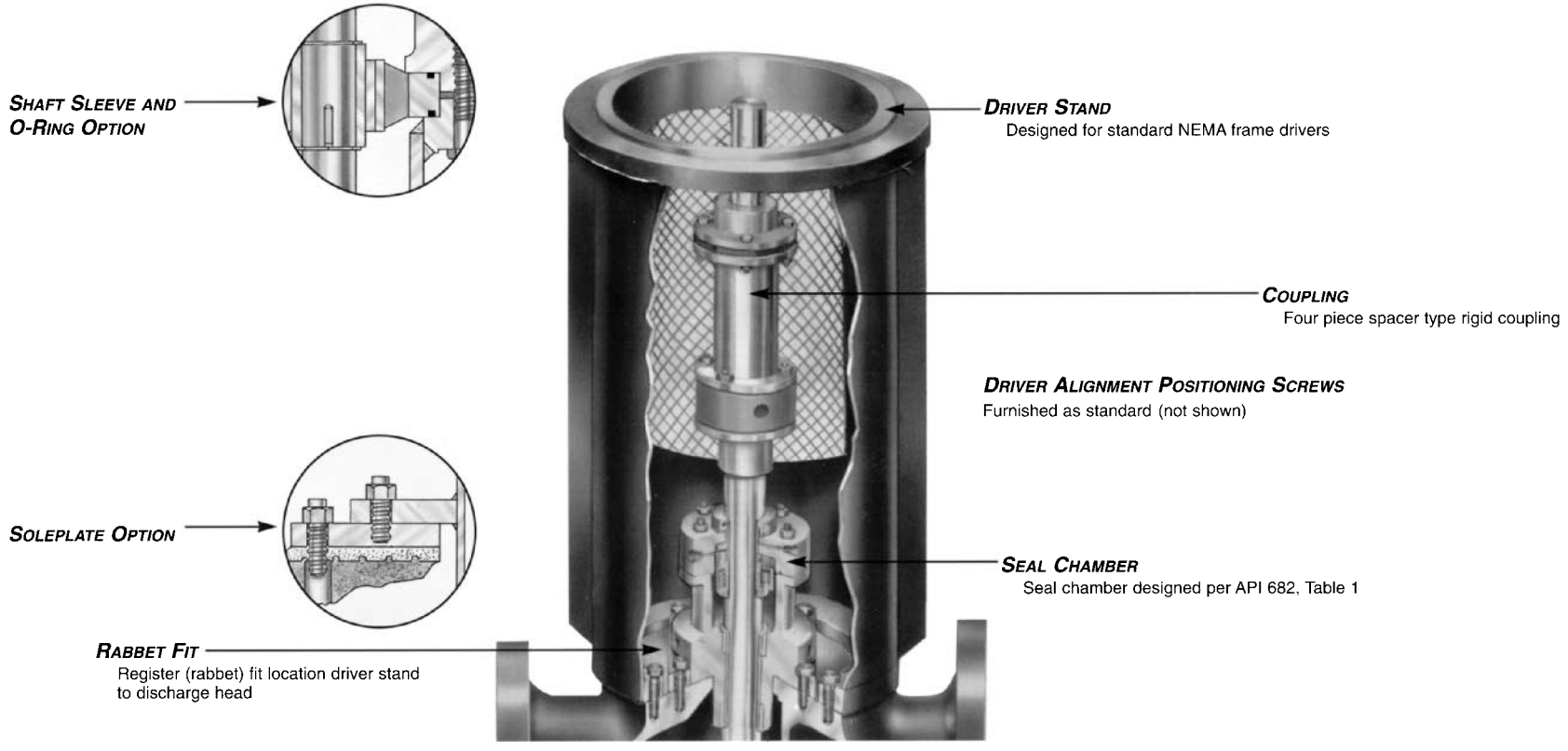


FIG. P-315 API 610 8th edition compliant rigid coupled without bearing housing. (Source: Sulzer Pumps.)

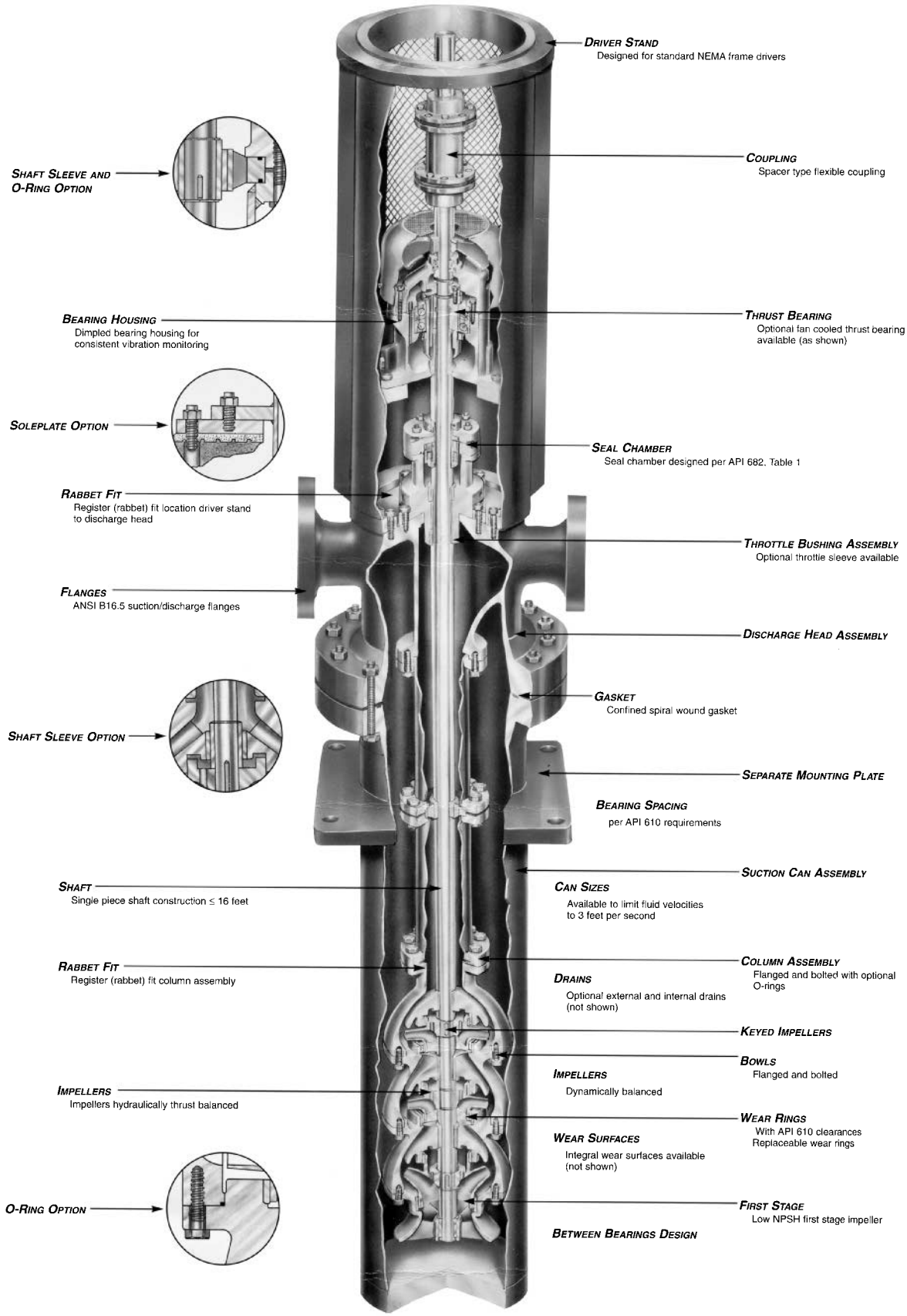


FIG. P-316 API 610 8th edition compliant flexible coupled with integral thrust bearing. (Source: Sulzer Pumps.)

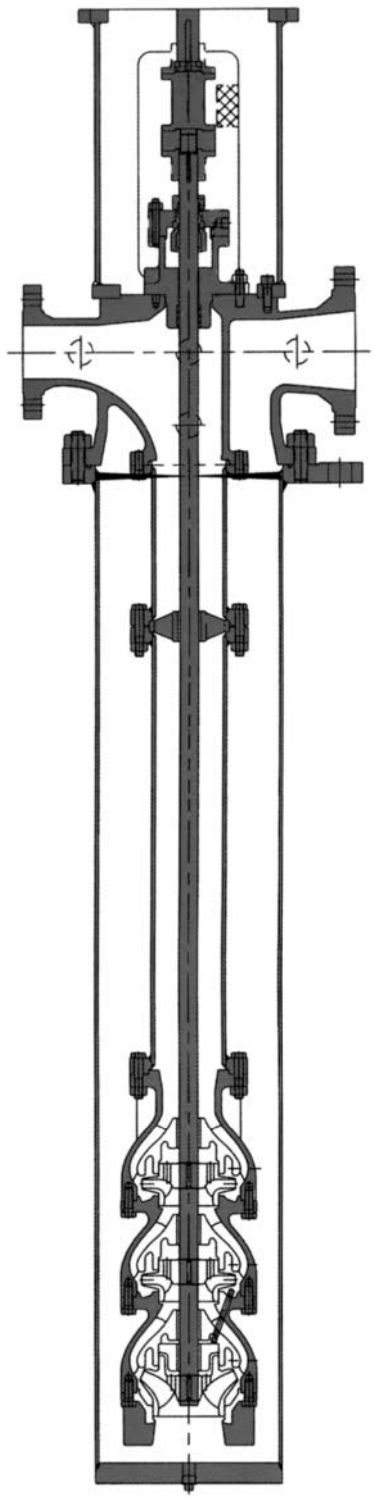


FIG. P-317 VC_2R8 commercial standard. API 682, Table 1, seal chamber dimensions; nozzle load capability per API 610; integral mounting plate; optional suction can thicknesses available; flanged and bolted columns with optional O-rings; single piece shaft construction ≤ 16 feet; flanged and bolted bowls with register fit; hydraulically balanced impellers; keyed and axially retained impellers. (Source: Sulzer Pumps.)

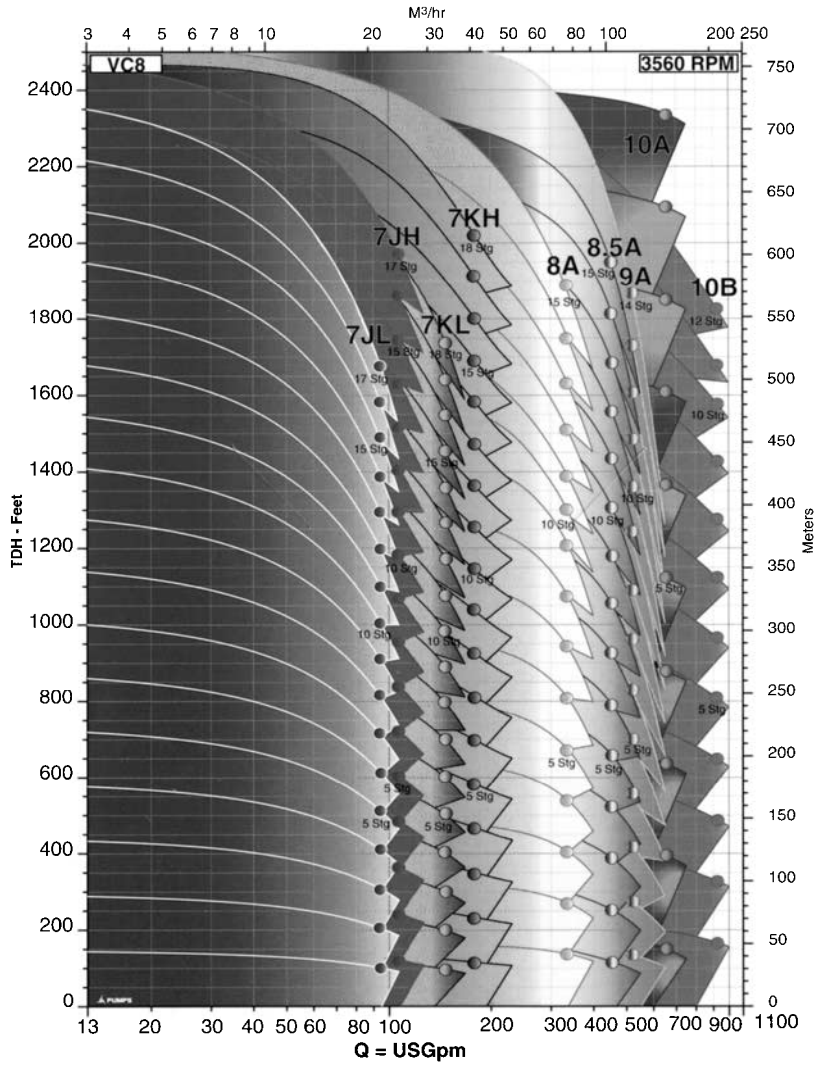


FIG. P-318 Hydraulic range chart (Type VR 8). (Source: Sulzer Pumps.)

- Vibration detectors
- Seal leakage detectors
- Temperature detectors
- Temperature insulation

Vacuum Pumps*

An amazing number of process plant applications require vacuum systems for continuous or intermittent services. Central vacuum systems are often found in power stations and in industrial and marine installations for the purpose of priming

* Sources: As acknowledged in captions to illustrations.

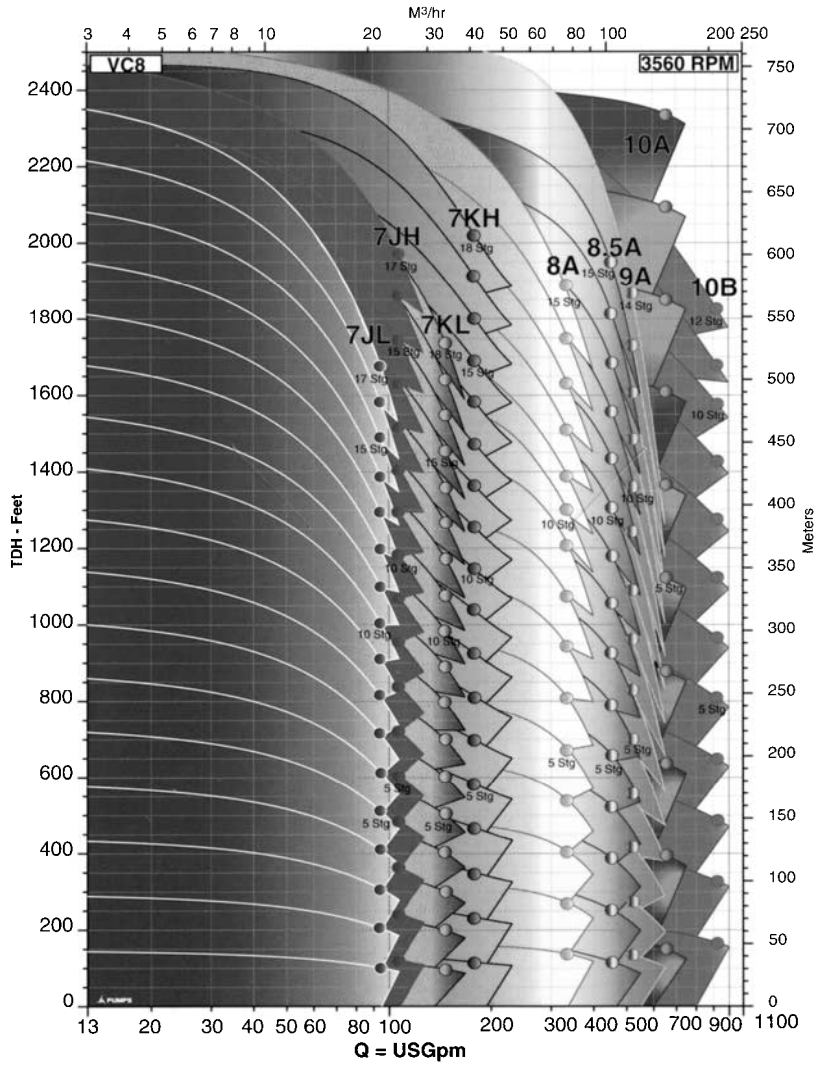


FIG. P-318 Hydraulic range chart (Type VR 8). (Source: Sulzer Pumps.)

- Vibration detectors
- Seal leakage detectors
- Temperature detectors
- Temperature insulation

Vacuum Pumps*

An amazing number of process plant applications require vacuum systems for continuous or intermittent services. Central vacuum systems are often found in power stations and in industrial and marine installations for the purpose of priming

* Sources: As acknowledged in captions to illustrations.

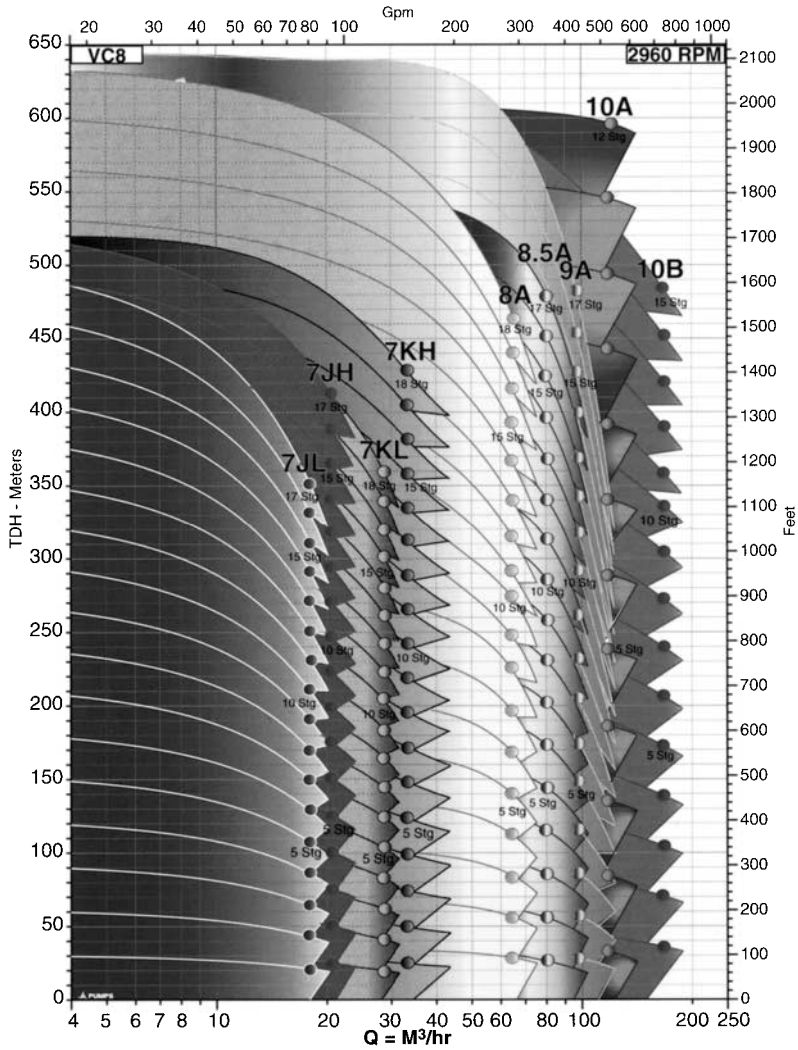


FIG. P-319 Hydraulic range chart (Type VR 8). (Source: Sulzer Pumps.)

large centrifugal pumps; similar systems provide vacuum for surgical and clinical purposes in hospitals, dentist's offices, and laboratories.

In sterilization processes, vacuum pumps remove air to permit rapid penetration of steam and/or gas into articles requiring sterilization. Food processing plants utilize vacuum systems for anything from poultry evisceration to final packaging in bags, windows, or blister packs for enhanced appearance and marketability. Vacuum filtration is used for sewage, chemicals, foods, and many mined or fibrous products.

Many condensing processes utilize vacuum pumps to evacuate condensers and to remove air leakage and noncondensibles. Smaller installations use liquid ring vacuum pumps in a dual role, removing condensate as well as noncondensibles, sometimes utilizing the cooled condensate as a service liquid. Vacuum pumps are also used in impregnation and metallurgical treatment processes where air and gas have to be removed prior to impregnation with appropriate chemical fluids; or prior

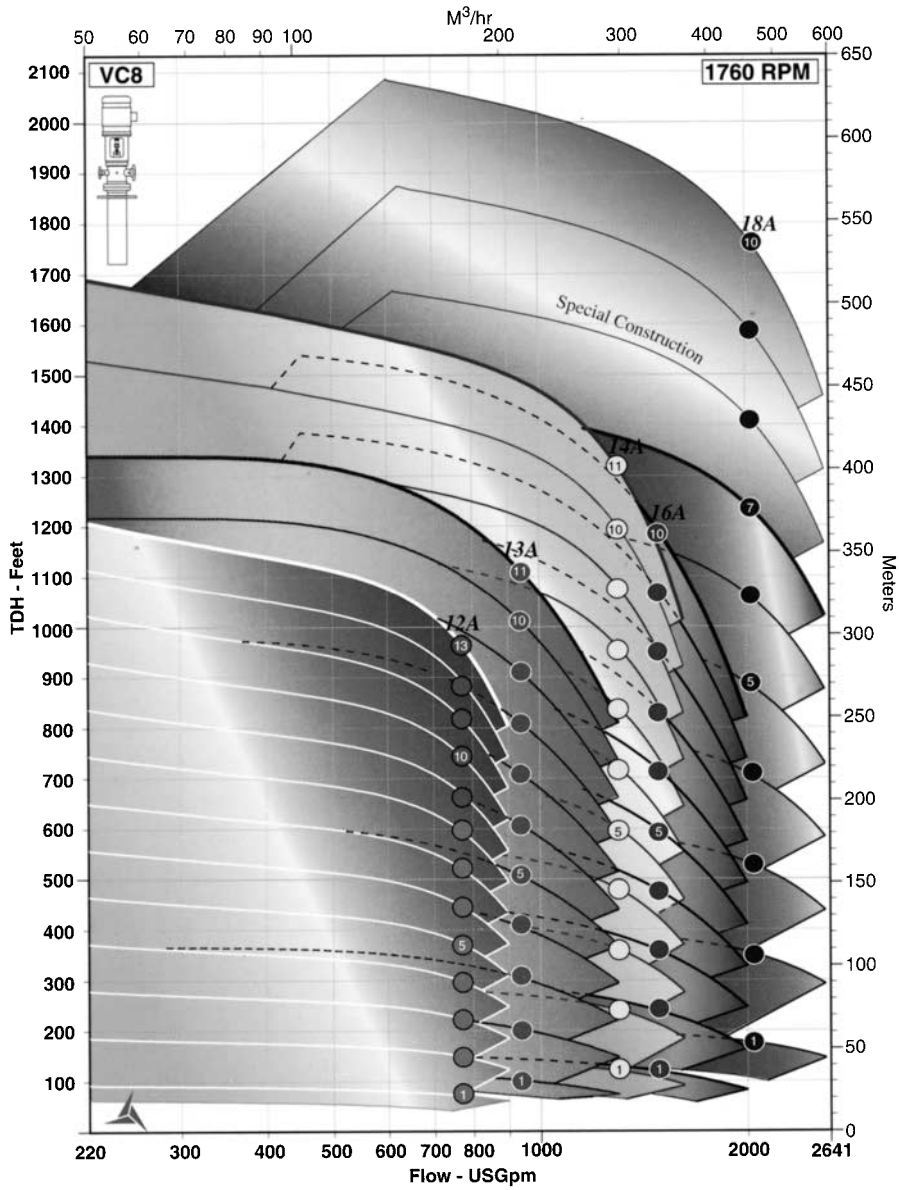


FIG. P-320 Hydraulic range chart (Type VCR 8). (Source: Sulzer Pumps.)

to the application of diffusion coatings, or in advanced ion and plasma-type surfacing techniques.

Vacuum pumps are used for drying, distillation, and evaporation. Lower boiling temperatures attained under vacuum preserve nutrients and improve taste, quality, and shelf life of products such as candies, jams, pharmaceuticals, and many mild products. Deaeration is needed for products such as meat pastes, sauces, soups, cellulose, latex, bricks, tiles, sewer pipes, and pottery clay. Also, vacuum conveyance of dangerous, viscous, contaminated, powdery, flaky, bulky, or simply hard-to-handle materials or products is used. The author remembers the ease and utter simplicity with which laminated plastic toothpaste tubes are transferred in partially

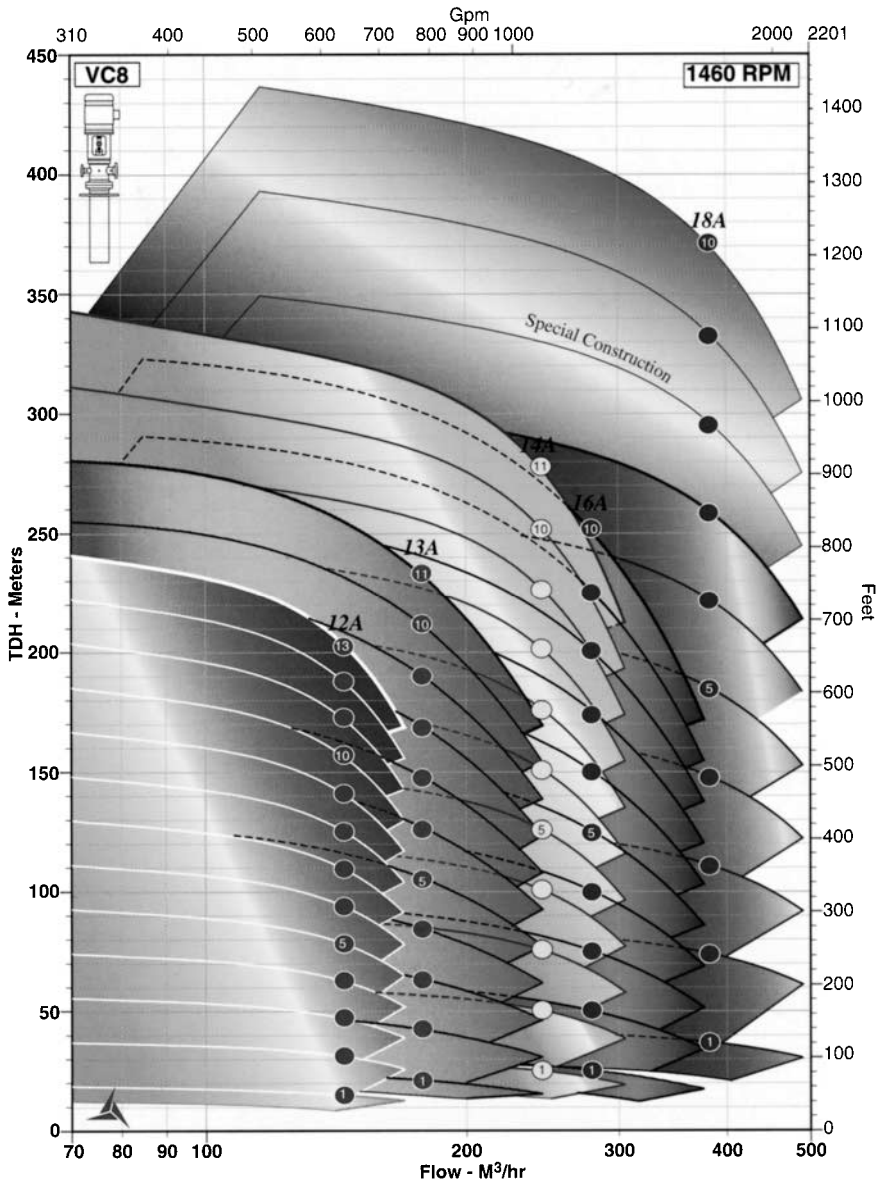


FIG. P-321 Hydraulic range chart (Type VCR 8). (Source: Sulzer Pumps.)

evacuated, transparent plastic pipes from the forming machine at one end of the plant to the filling equipment at the other end of the building.

With a profusion of processes and applications thus benefiting from vacuum pumps, it is not surprising that many different types and styles, sizes and models, and configurations and variations of vacuum producing machinery are available to the user. The familiar steam, gas, and fluid jet injectors/eductors must be acknowledged as prime vacuum producers; however, we will only mention them in passing because they lack moving parts and thus do not fit our definition of "machinery."

Vacuum pumps are often classified in two broad categories: dry type and liquid type. Dry types include lobe, rotary piston, sliding vane, and even diaphragm

TABLE P-40 VCR8 Design Features, Advantages, and Benefits

	Features	Advantages	Benefits
Driver stand assembly	■ Vertical motor with solid shaft	■ Precision alignment	■ Improved seal and bearing life
	■ Integral thrust bearing	■ Axial and radial setting of rotor ■ Controls hydraulic thrust	■ Standard driver may be selected ■ Improved seal life
	■ Driver alignment positioning screws	■ Controls motor radial position	■ Reduced maintenance time
	■ Spacer coupling	■ Allows seal and bearing maintenance without disturbing driver	■ Reduced maintenance time
	■ Dimpled location	■ Consistent vibration monitoring	■ Trend for planned maintenance
Head assembly	■ ANSI B16.5 class 300 flanges	■ Consistent with process piping standards	■ Use of standard pipe flanges
	■ Nozzle load capability per API 610	■ Simplifies piping layout	■ Reduces piping layout costs
	■ Complies with API 682, Table 1	■ Allows use of API 682 single or dual seals as required ■ Allows interchangeability	■ Improved seal life, reduced emissions ■ Reduced inventory
	■ Throttle bushing assembly	■ Controls seal chamber pressure ■ Provides rotor stiffness ■ Allows use of various API 610 piping plans	■ Improved seal life ■ Greater flexibility
Column assembly	■ Flanged and bolted with register fit	■ Controls axial and radial position of bowl assembly	■ Improved reliability ■ Ease of maintenance
	■ Bearing bushing spacing per API 610	■ Ensures adequate separation margin from critical speeds	■ Improved reliability
Bowl assembly	■ Flanged and bolted with register fit	■ Controls axial and radial position of bowl assembly	■ Improved reliability ■ Ease of maintenance
	■ Low NPSH first-stage impeller	■ Reduced pump setting length	■ Reduced construction cost
	■ Between bearings first-stage impeller design	■ Reduced deflection	■ Improved reliability
	■ Impeller keyed to shaft	■ Positive drive and positioning	■ Improved reliability
	■ Impellers hydraulically thrust balanced	■ Reduced thrust load	■ Improved bearing life
	■ Single piece shaft construction ≤16 feet	■ Simplifies assembly ■ Controls runout	■ Ease of maintenance ■ Improved reliability
	■ Dynamically balanced enclosed-type impeller	■ Reduced unbalance	■ Improved seal life ■ Improved reliability
	■ Replaceable wear surfaces	■ Allows refurbishment to as-new condition	■ Reduced total life cycle cost
Suction can assembly	■ 600 psi pressure rating	■ Consistent with process piping design	■ Improved plant reliability
	■ Controlled fluid velocities	■ Reduces internal losses ■ Reliable suction performance	■ Improved first-stage impeller life
	■ Internal or external drain	■ Allows evacuation of process fluids	■ Reduces maintenance costs
	■ Separate mounting plate	■ Allows through bolting on discharge head	■ Improved maintenance and reliability
	■ Soleplate optional	■ Allows foundation to be completed prior to pump installation	■ Simplified construction process
	■ Confined gasket	■ Controlled compression ensures reliable pressure retention	■ Reduced risk of leakage

TABLE P-41 VCR8 Pumps

Design Criteria	Bowl Sizes	Capacity	Head (per Stage)	Pressure	Temperature	Speed
SI units	150 to 255 mm	to 230 m^3/h	to 60 m	to 70 bar	to 205°C	to 3600 rpm
U.S. units	6 to 10 in	to 1000 gpm	to 200 ft	to 1000 psi	to 400°F	to 3600 rpm

TABLE P-42 VCR8 Pumps

Design Criteria	Bowl Sizes	Capacity	Head (per Stage)	Pressure	Temperature	Speed
SI units	300 to 450 mm	to 475 m^3/h	to 60 m	70 bar	to 205°C	to 2960 rpm 12 in
U.S. units	12 to 18 in	to 2100 gpm	to 200 ft	1000 psi	to 400°F	to 1800 rpm other sizes

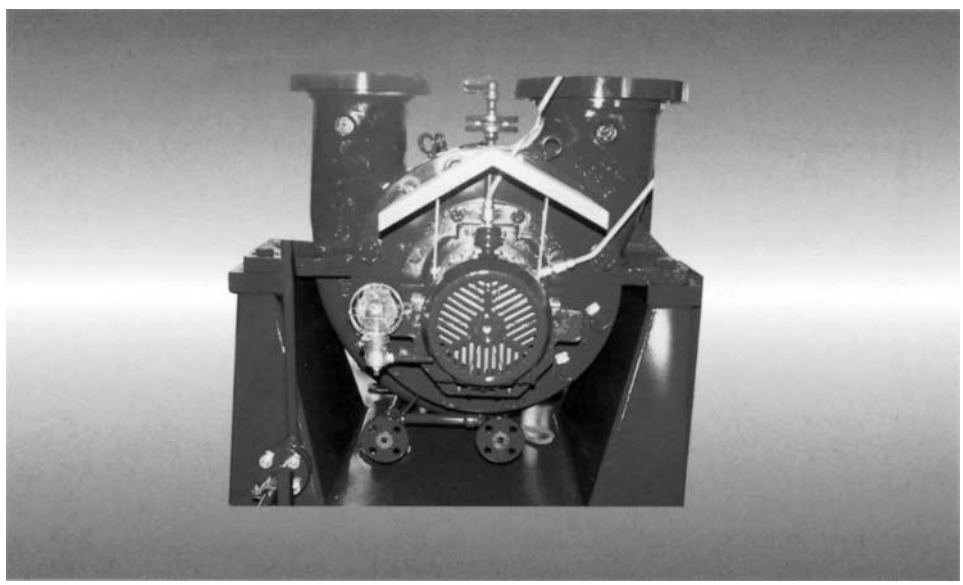


FIG. P-322 Pump external view (type CD 8). (Source: Sulzer Pumps.)

TABLE P-43 CD8 Pumps

Operating Data	SI Units	U.S. Units
Discharge sizes	150 to 300 mm	6 to 12 in
Capacities	to 2750 m^3/h	to 17,000 gpm
Heads	to 400 m	to 1400 ft
Pressures	to 50 bar	to 735 psi
Temperatures	–28 to 425°C	–20 to 800°F
Speeds	to 3100 rpm	to 3800 rpm

TABLE P-44 CD8 Features, Functions, and Benefits

	Features	Functions	Benefits
Design	■ Full-range coverage	■ Selections fall within 80% to 110% of best efficiency point	■ Smooth operation ■ Longer service life ■ Optimized efficiency
	■ Full compliance with API 610 8th edition requirements	■ Heavy duty design and construction	■ Longer reliable service ■ Suitable for 3 years uninterrupted service and 20 year service life
Pressure casing	■ Symmetrical, double end cover construction	■ Improved maintenance access, cleanout and decontamination capabilities ■ Uniform warming ■ Improved flexibility of application	■ All surfaces accessible ■ Completely drainable ■ Reliable high-temperature operation ■ Clockwise or counterclockwise rotation with same component parts
	■ Cast construction with double volute and double suction entry	■ Reduced radial loads ■ Symmetry of flow into impeller	■ Reduced rotor deflection ■ Improved bearing and seal life ■ Improved suction characteristics
	■ Centerline mounting with robust feet	■ Suitable for operation in wide range of temperatures up to 800°F	■ Reduced misalignment problems ■ Reduced maintenance
	■ Integral end cover and bearing hanger	■ Stiffer support eliminates possible frame resonance ■ Reduced number of component parts	■ Reduced frame vibration ■ Improved bearing and seal life ■ Simplified maintenance
	■ Available in various metallurgies, including S-4, S-6, C-6 and A-8	■ Suitable for operation in wide range of services	■ Optimized material selection to ensure appropriate service life
	■ Seal chamber dimensions compliant with API 682 Table 1	■ Suitable for state-of-the-art mechanical seal technology	■ Improved seal interchangeability and seal life ■ Reduced emissions
Impeller	■ Double suction impeller	■ Minimal axial loads ■ Low NPSHr	■ Improved bearing life ■ Reduced vessel heights ■ Improved NPSH margins
	■ 9000 to 11,000 Nss suction hydraulics available	■ Stable suction performance throughout entire flow range	■ Reduced vibration ■ Improved bearing and seal life
	■ 5 vane staggered construction available	■ Reduced hydraulic pulsations	■ Reduced vibration ■ Improved bearing and seal life
	■ Simple and effective impeller retention	■ Axially and radially secured in all directions	■ Secure in operation through transient operating conditions ■ Easily maintained
	■ Enclosed impeller	■ Higher efficiencies ■ No impeller setting	■ Reduced power consumption
	■ Dynamic balance to 4 W/N	■ Minimized dynamic unbalance forces	■ Reduced vibration ■ Improved bearing and seal life
Shaft	■ Heavy duty shaft with minimum bearing span	■ Higher torque transmission capability ■ Lower static and dynamic deflection	■ Higher torsional stress safety margin ■ Improved reliability
	■ Stiff shaft design	■ Ensures separation from critical speeds throughout entire operating range	■ Smoother operation at all allowable operating speeds
	■ Taper shaft extension	■ Simplified coupling, bearing and seal maintenance	■ Reduced maintenance downtime
Bearing	■ All steel load bearing components ■ 40° angular contact thrust bearing ■ Deep groove radial bearing	■ Reliable long-term service ■ Selected for minimum 25,000 hours L10 bearing life ■ Heavy duty carrying capability	■ Maximized bearing reliability and service life
	■ INPRO™ labyrinth seals fitted as standard	■ Minimized ingress of oil contaminants	■ Improved bearing life
	■ Fan cooling or water cooling options available	■ Efficient cooling features ensure cool running of bearings under all pump operating temperatures	

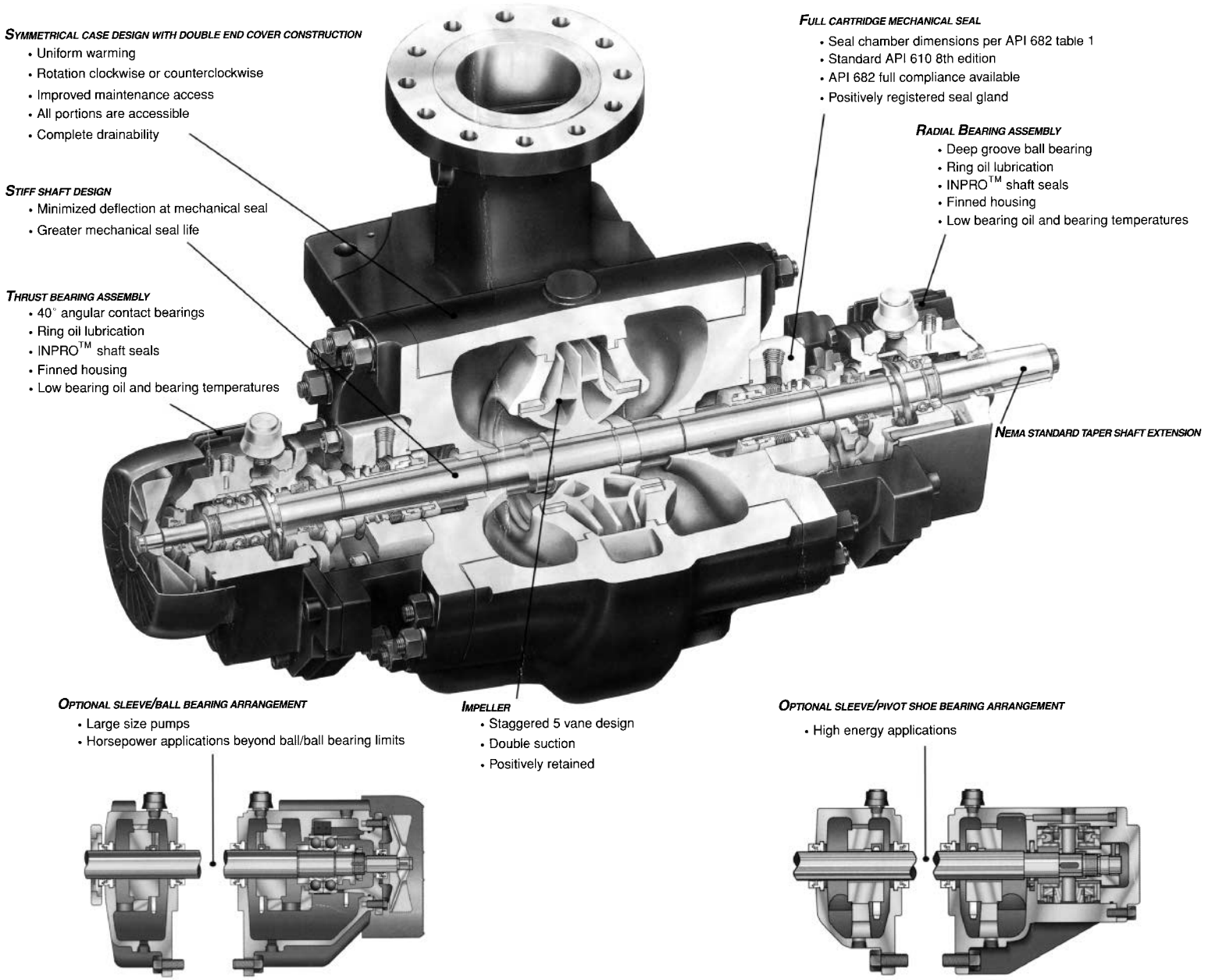


FIG. P-323 Design features (type CD 8). (Source: Sulzer Pumps.)

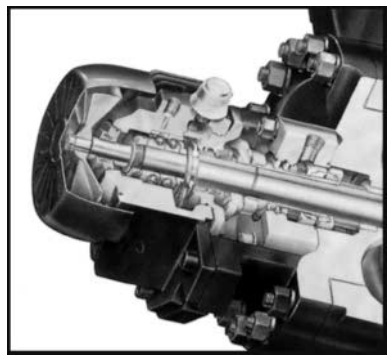


FIG. P-324 Thrust bearing assembly (type CD 8 option). High capacity fan; water cooling; inboard heat dissipator; purge or pure mist oil lubrication. (Source: Sulzer Pumps.)

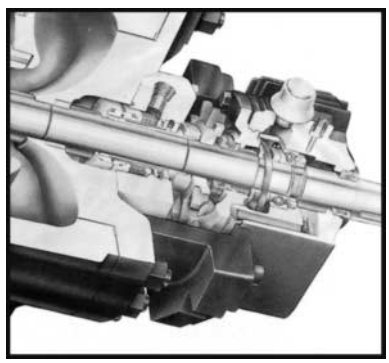


FIG. P-325 Radial bearing assembly (type CD 8 option). Water cooling; inboard heat dissipator; purge or pure mist oil lubrication. (Source: Sulzer Pumps.)

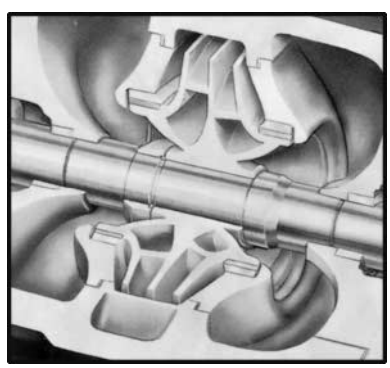


FIG. P-326 Impeller (type CD 8 option). Integral wear surfaces; nonmetallic wear rings. (Source: Sulzer Pumps.)

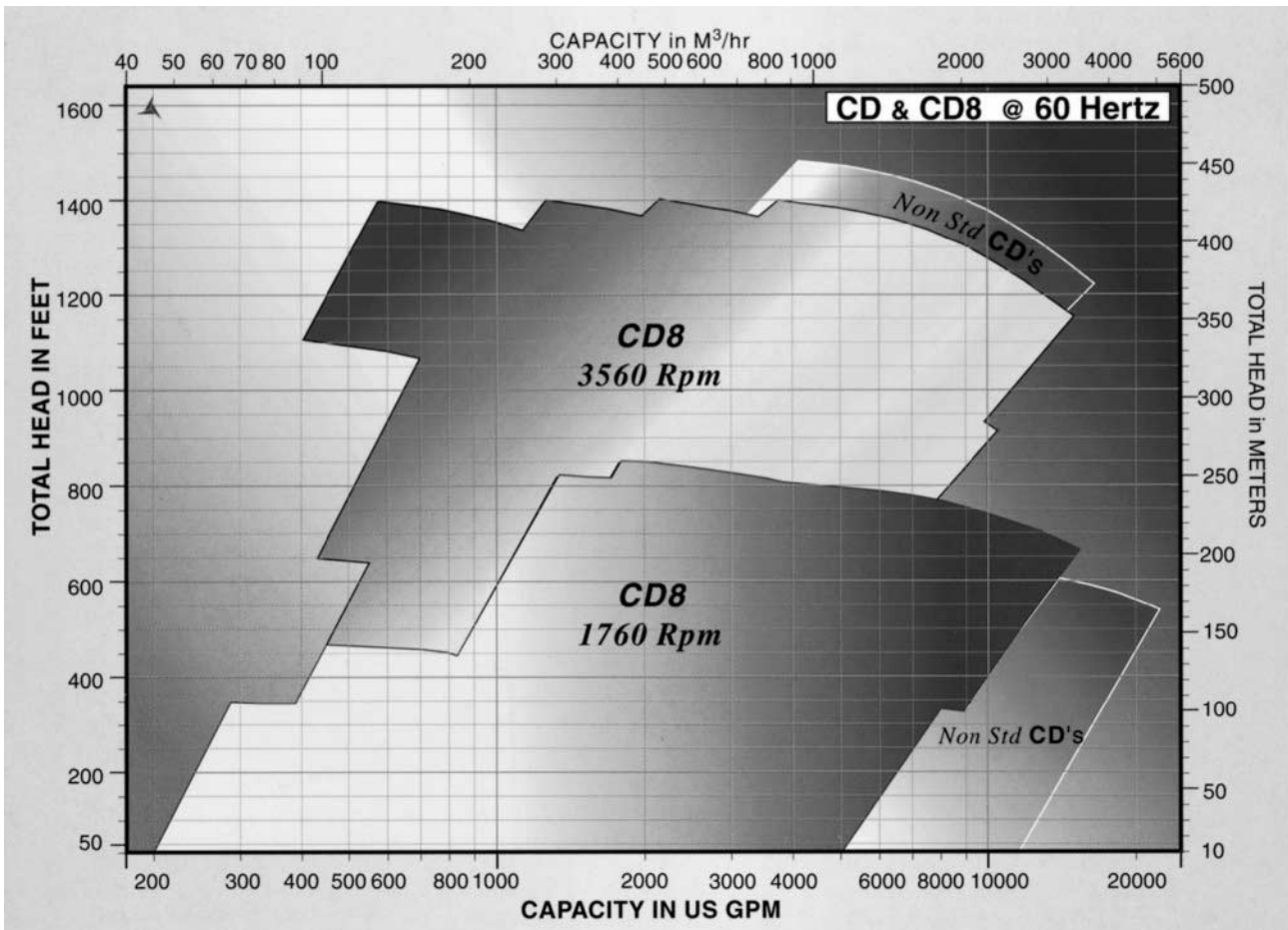


FIG. P-327 Hydraulic range chart (type CD 8). (Source: Sulzer Pumps.)

pumps. Liquid vacuum pumps include liquid jet and liquid ring pumps. Figure P-335 shows the operating ranges for many of these pumps. It should be noted that there is considerable overlap among ranges.

The most important vacuum producers and their respective operating modes and features are of interest to use in the order listed in Fig. P-335.

Single-stage liquid ring pumps

Figure P-336 depicts the operating principle of a liquid ring pump. Its circular pump body (A) contains a rotor that consists of a shaft and impeller (B). Shaft and impeller centerlines are positioned parallel, but eccentrically offset relative to the centerline of the pump body. The amount of eccentricity is related to the depth of the liquid ring (C). The liquid ring is formed by introducing service liquid, normally water, via the pump suction casing (L) and through the channel (D) positioned in the suction port plate (E). The centrifugal action of the rotating impeller forces the liquid toward the periphery of the pump body. By controlling the amount of service liquid within the pump body where the impeller blades are completely immersed to their root at one extreme (F) and all but their tips exposed at the other extreme (G), optimum pumping performance will be attained.

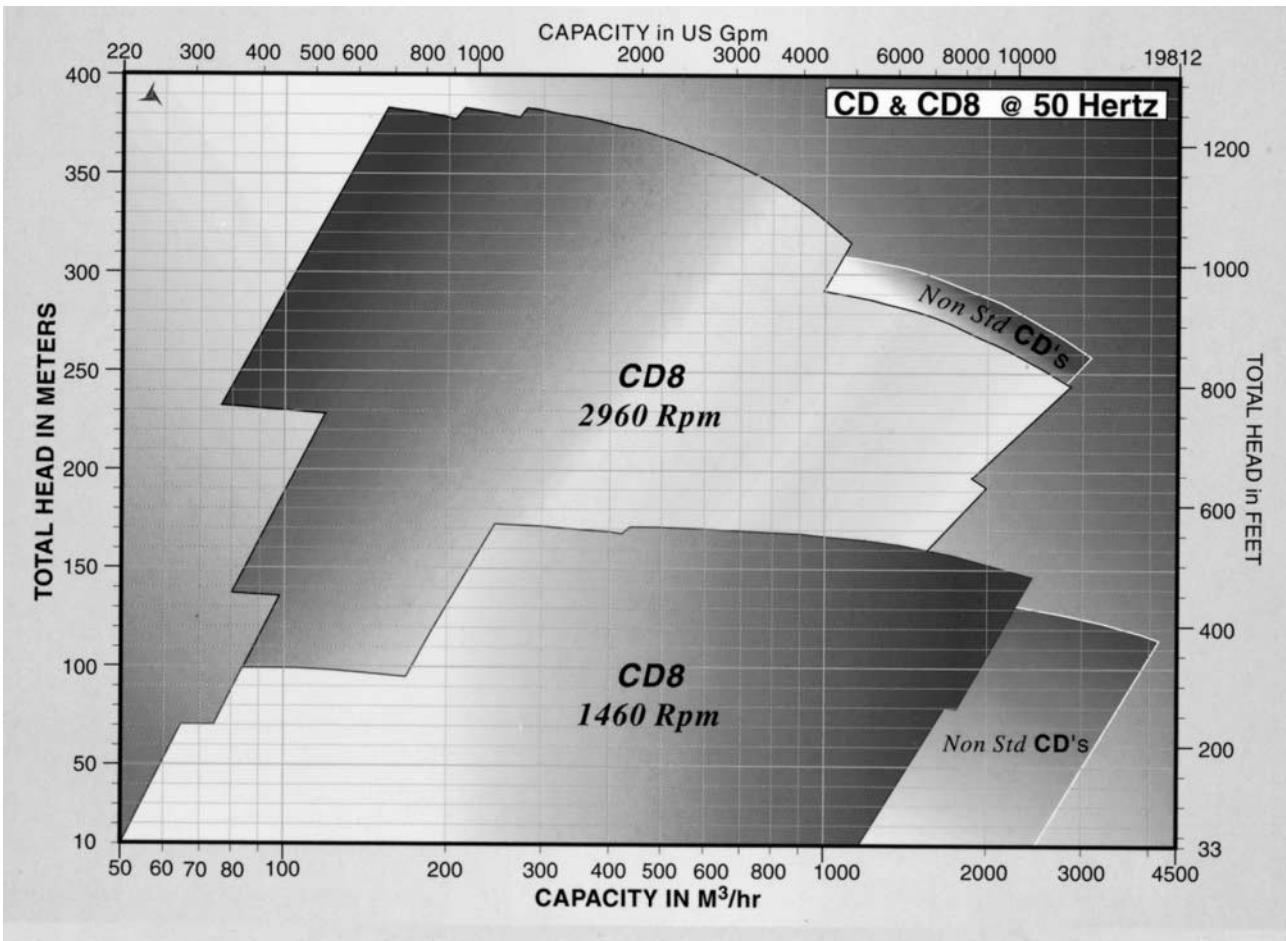


FIG. P-328 Hydraulic range chart (type CD 8). (Source: Sulzer Pumps.)

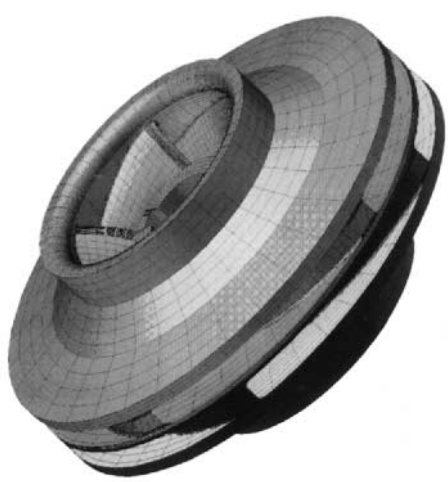


FIG. P-329 The use of rapid prototyping and computational fluid dynamics allows the optimization of pump performance. (Source: Sulzer Pumps.)

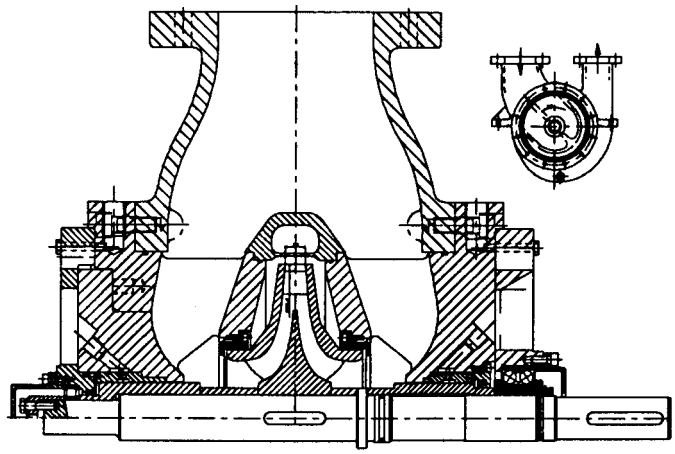


FIG. P-330 Model testing, the next step, is necessary in the development process to validate numerical calculation done during computational fluid dynamics (CFD) calculations. (Source: Sulzer Pumps.)



FIG. P-331 Once successful model testing has been achieved, 3D computer-aided design (CAD) and computer-aided manufacturing (CAM) are implemented. (Source: Sulzer Pumps.)

When this pumping action is achieved, the vapor to be handled is induced through the suction port (H) when the depth of impeller blade immersion is being decreased. Then as the immersion increases, the vapor is compressed and discharged through the discharge port (J) in the intermediate port plate (K). As there is no metal-to-metal contact between the impeller and the pump body and

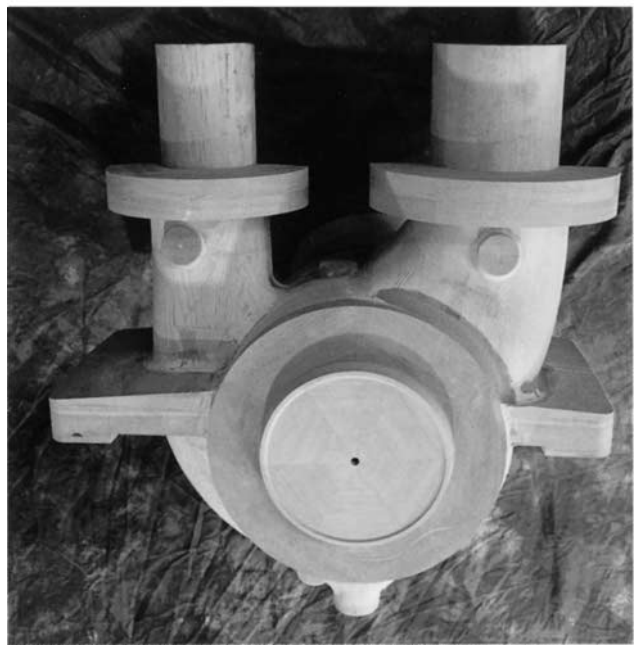


FIG. P-332 The final step is the creation of patterns. Modern techniques including stereolithography are adopted to ensure foundry patterns accurately reflect design and manufacturing requirements during the casting process. (Source: Sulzer Pumps.)

intermediate plates, the need for lubrication is eliminated and wear is reduced to a minimum.

During the compression cycle, heat is being imparted to the liquid ring. In order to maintain a temperature below the vapor point, cooling must be applied. This cooling is achieved by continuously adding a cool supply of service liquid to the liquid ring. The amount of coolant added is equal to that discharged through the discharge port (J) together with the compressed vapor. The mixture of vapor and liquid is then passed to subsequent stages and eventually through the pump discharge for separation.

An entire vacuum pumping system is shown in Fig. P-337. This so-called full sealant recovery system is used to conserve sealant and/or where suitable or compatible sealant is not available from an outside source. Periodic sealant makeup and/or purge may be required. Full recirculation of sealant is provided from the discharge separator tank. Cooling is provided by running recirculated sealant through a heat exchanger. Separate cooling liquid or gas is required.

Liquid jet vacuum pumps

A typical liquid jet pump is illustrated in Fig. P-338. A centrifugal pump circulates water (the usual hurling liquid) through the multijet nozzle and venturi and returns it to the separation chamber. The water, forced at high velocity across the gap between the nozzle and venturi, entrains the air and gases in multiple jet streams, creating a smooth, steady vacuum in the air suction line and vacuum system. This

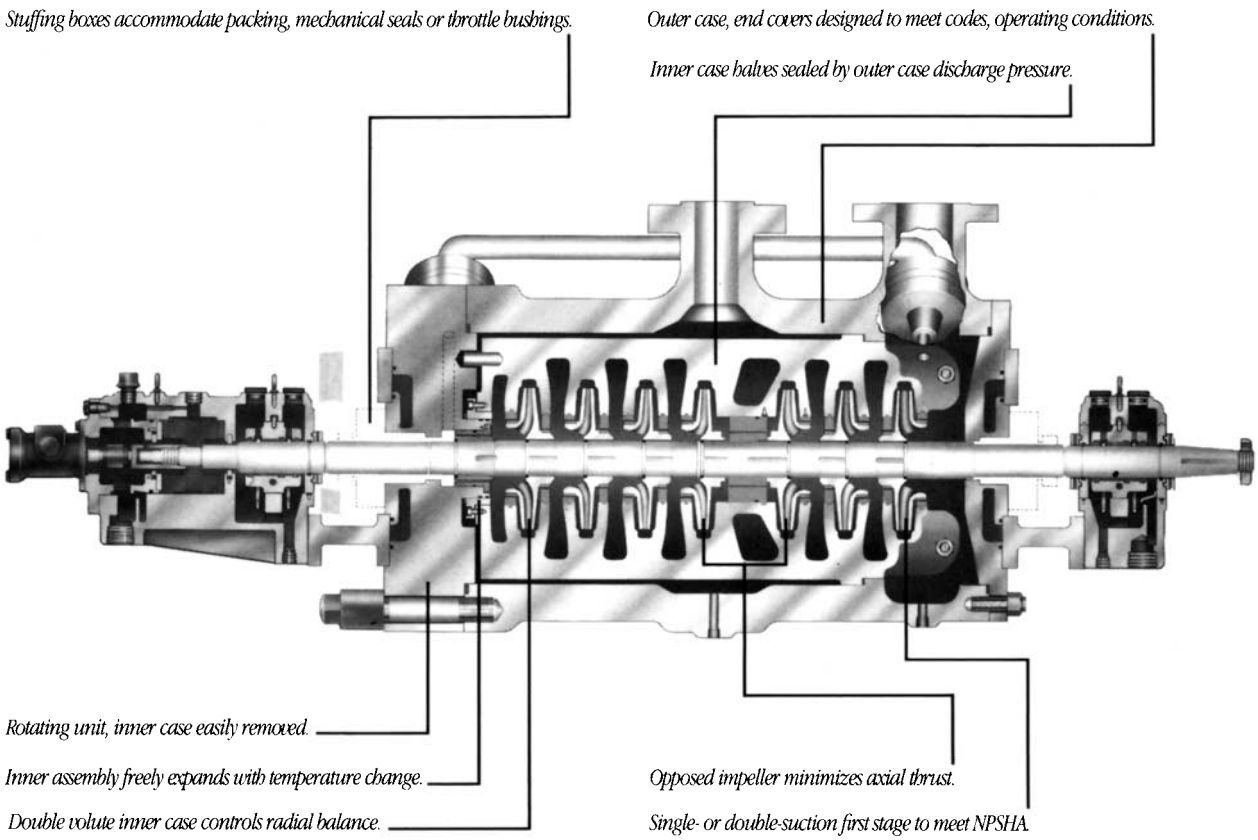


FIG. P-333 Pump section (type CP). (Source: Sulzer Pumps.)

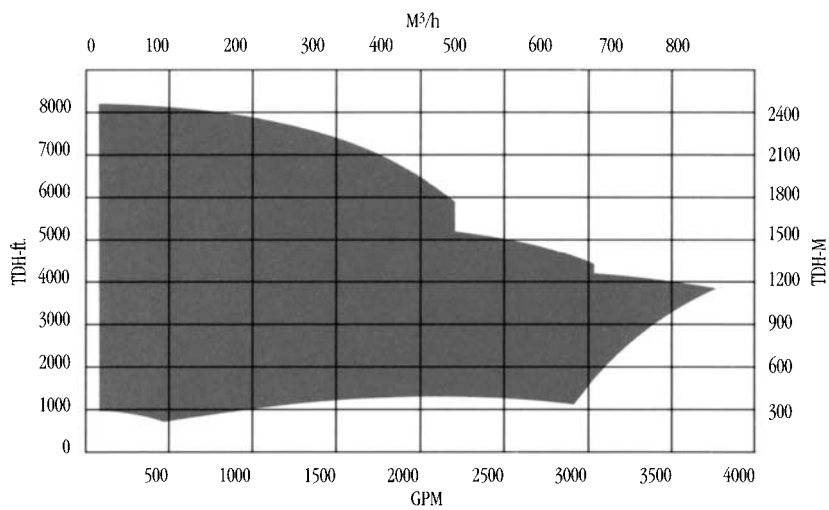


FIG. P-334 Pump performance range (type CP). (Source: Sulzer Pumps.)

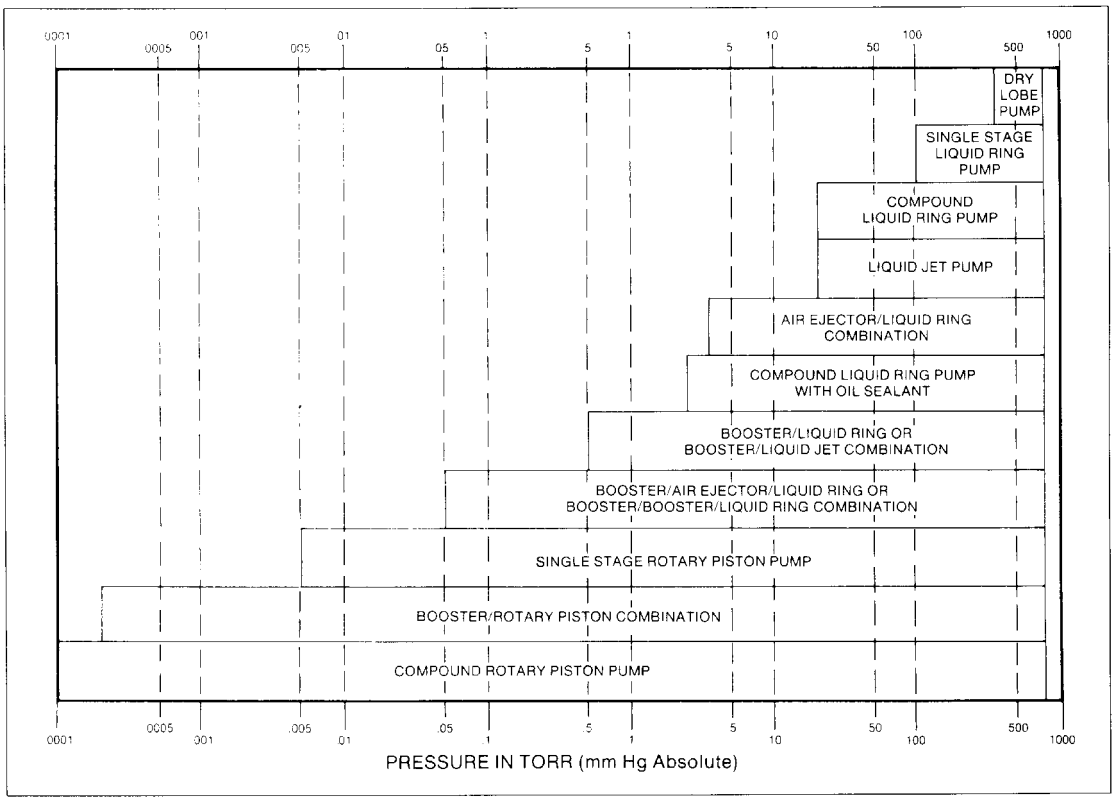


FIG. P-335 Typical pressure ranges for various vacuum pumping devices. (Source: Stokes Division of Pennwalt Corporation, Philadelphia, Pa.)

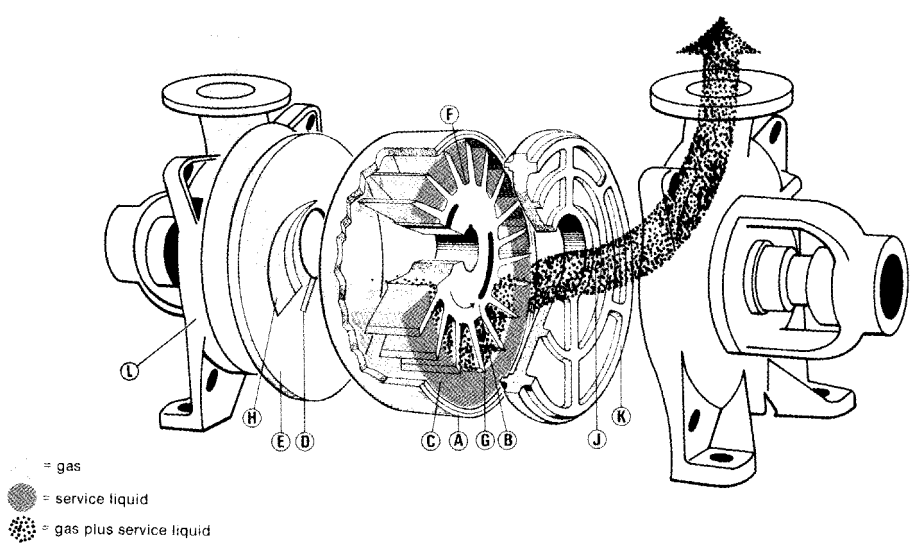


FIG. P-336 Operating principle of liquid vacuum pumps. (Source: SIHI Pumps, Inc., Grand Island, N.Y.)

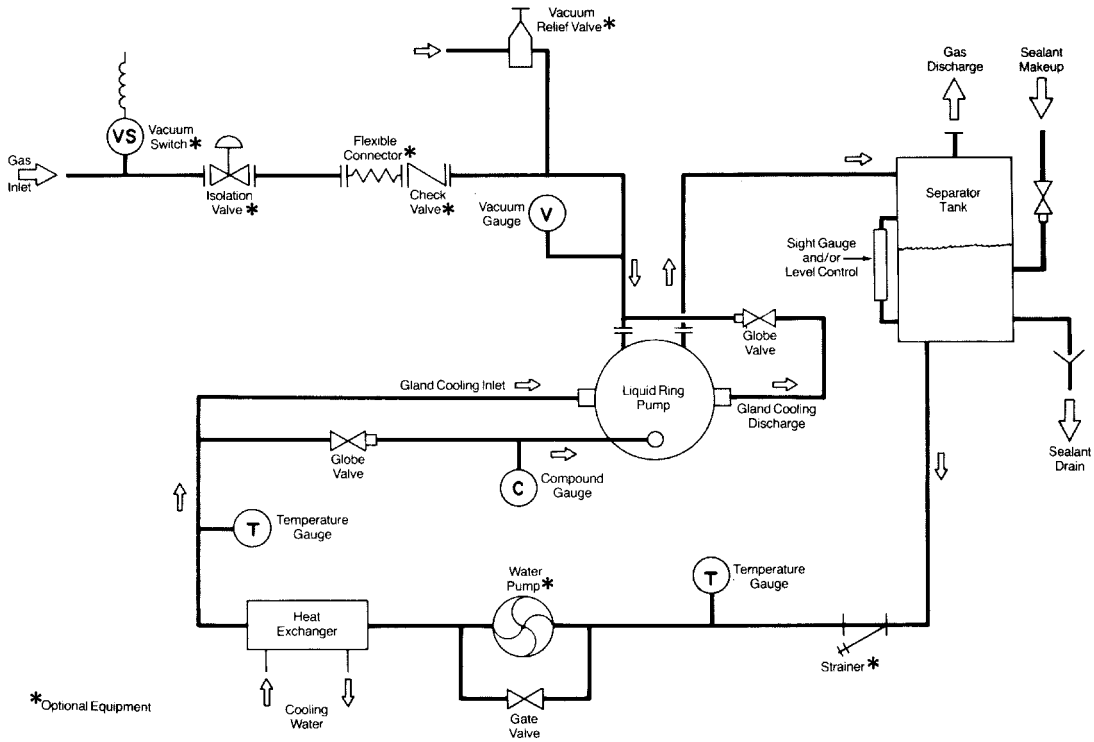
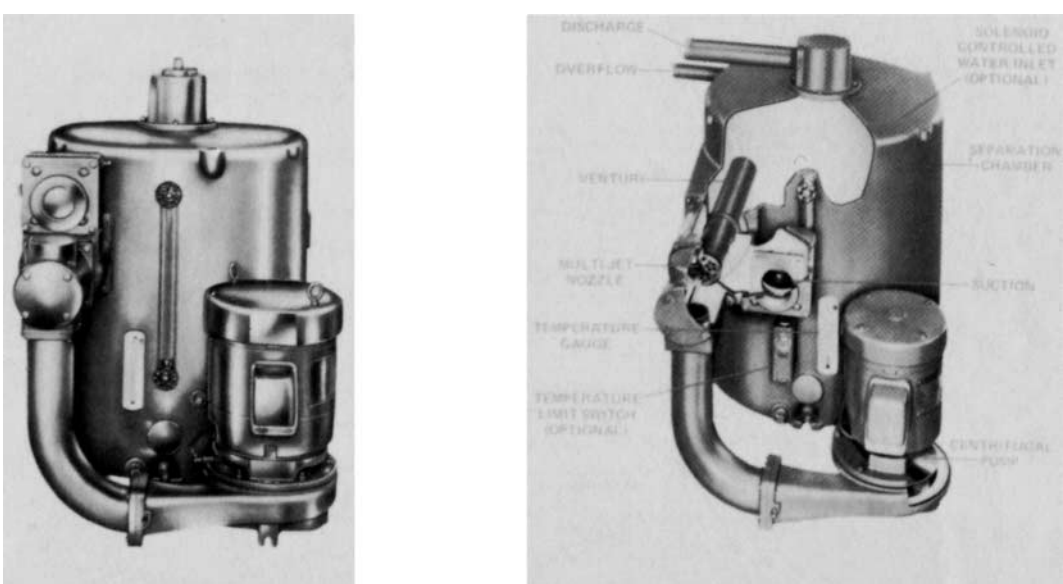


FIG. P-337 Liquid ring vacuum pumping system with full sealant recovery. (Source: Kinney Vacuum Company, Boston, Mass.)



A B

FIG. P-338 (*A*) Typical liquid jet vacuum pump. (*B*) Cutaway view of liquid vacuum pump. (Source: Kinney Vacuum Company, Boston, Mass.)

mixture is discharged through the venturi tangentially into the separation chamber, causing the water in the separation chamber to rotate, which results in a centrifugal action that forces the water to the periphery of the chamber, while the air is separated and discharged. When the hurling liquid is water, it is cooled by a continuous flow of cooling water into the separation chamber. Where process requirements allow and economy is an important factor, automatic controls and other cooling methods are often utilized.

R

Reactors; Chemical Reactors

There are two main distinctions between reactors, batch and continuous. In a batch reactor a certain amount of the reactants is handled at one time. In continuous reactors, the process continues indefinitely. This is the most common type of reactor in petrochemical and refinery service.

A *batch reactor* is a closed system. An example is a batch of paper pulp being made for a specific or customized application.

A *semibatch reactor* is not a closed system. This type is useful in cases such as the manufacture of certain chemicals where a volatile chemical must be added slowly to a nonvolatile chemical (examples include the manufacture of certain glycols).

Tubular reactors (either long bent tube or shell and tube) may be either batch or continuous reactors.

Continuous reactors are "at work" all the time. This means newly introduced reactants mix to some extent with products. This extent is termed *backmixing*. A tower has many plates or baffles in it and experiences less backmixing as, for instance, a tank with no plates. Continuous reactors can then be found within towers and columns. Towers may be packed or plate (bubble cap or sieve tray) type. Optimum reactor design attempts to curtail the amount of "dead space" or areas where no reaction is taking place. It is also possible to have reactants take a shorter path than is necessary for optimum reaction. This is called *shortcircuiting*.

Catalytic reactors are continuous reactors more often than not. The main subdivision types include: fluidized or fixed bed. Fixed bed types may be either tubular, bed, or multitray types. Fluidized bed types further break down into stationary or moving (recirculating) bed types and tubular (transfer tube) types. The catalyst is generally in powdered suspension and may be removed either in batches or continuously withdrawn and regenerated. In transfer tube types, the catalyst stays in suspension with the fluid flow through the tubes.

Reactor performance is measured by its divergence from ideal conditions. Plug flow means all the fluid in the reactor has the same residence time in the reactor (no mixing with fluid streams that entered the reactor at different times). Very long tube reactors with turbulent flow can approximate this condition. Perfect mixing condition means the entering fluid in the reactor is homogeneous with the material already in the vessel on a molecular scale (perfect mixing case). In segregated mixing, the mixing is not uniform and pockets of fluid behave as "minireactors."

Refineries, Petroleum*

Crude oil is the principal raw material for a petroleum refinery. It may be of natural origin (from underground geological formations) or synthetic (recovered from tar sands). Crude oil is a mixture of many hydrocarbons and, depending on its source, varies considerably in composition and physical properties. Its elementary

---

* Source: Environment Canada, extracts from EPS/1/PN/4, October 1995.

composition (by mass) usually falls within the following ranges: 84 to 87 percent carbon, 11 to 14 percent hydrogen, 0 to 3 percent sulfur, 0 to 2 percent oxygen, 0 to 0.1 percent nitrogen, 0 to 1 percent water, and 0 to 0.1 percent mineral salts. Crude oil may also contain trace amounts of heavy metals such as iron, arsenic, chromium, vanadium, and nickel.

Crude oils are broadly classified by hydrocarbon composition as paraffinic, naphthenic, asphaltic, mixed (contains paraffinic and asphaltic material), and aromatic base (prevalent in the Middle East).

The major steps in converting crude oil to various products are *separation*, *conversion*, *treatment*, and *blending*. In the first step, crude oil is separated into selected fractions mainly by distillation and to a lesser extent by solvent extraction and crystallization. Conversion processes are then used to change the size and shape of the hydrocarbon molecules to increase their monetary value. These processes include breaking molecules into smaller ones (catalytic cracking), rearranging molecules (catalytic reforming and isomerization), and joining molecules together (alkylation and polymerization). Impurities such as sulfur, nitrogen, and oxygen compounds that end up in intermediate products are removed or modified by treatment processes such as desulfurization, denitrification, or treatment with chemicals (caustic soda or acid). In the final step, the refined products are usually blended and some additives are added to improve the quality to meet finished product specifications.

These processes are discussed in more detail in the following subsections. A simplified flow diagram of the various refinery processes and products is provided in Fig. R-1.

Refinery Processes

Separation

Atmospheric distillation. In this process, the crude oil is preheated and mixed with water in a desalter. The water is then separated from the crude, taking with it the salts entrained in the oil from the geological formation. The desalted crude oil is heated and fed to the distillation column at slightly above atmospheric pressure. Next, the crude oil is separated, by distillation and steam stripping, into fractions in a range of specific boiling temperatures. The various fractions are continuously drawn off and diverted for further processing or used as finished products. The lighter products are withdrawn from the top of the column whereas lower points on the tower draw off progressively heavier fractions. The tower bottoms, which contain the heaviest petroleum fraction, are transferred to a vacuum distillation tower for further separation.

Vacuum distillation. In this process, the residue from the atmospheric distillation tower is separated under vacuum into one or more heavy gas oil streams and heavy residual pitch.

Conversion

Cracking processes. Typical cracking processes include catalytic cracking, hydrocracking, and visbreaking or coking, both of which are thermal cracking processes.

1. *Catalytic cracking* is a key process used to increase the quality and quantity of gasoline fractions. The most commonly used process is the fluid bed type, which uses a finely powdered zeolite catalyst that is kept in suspension in the reactor

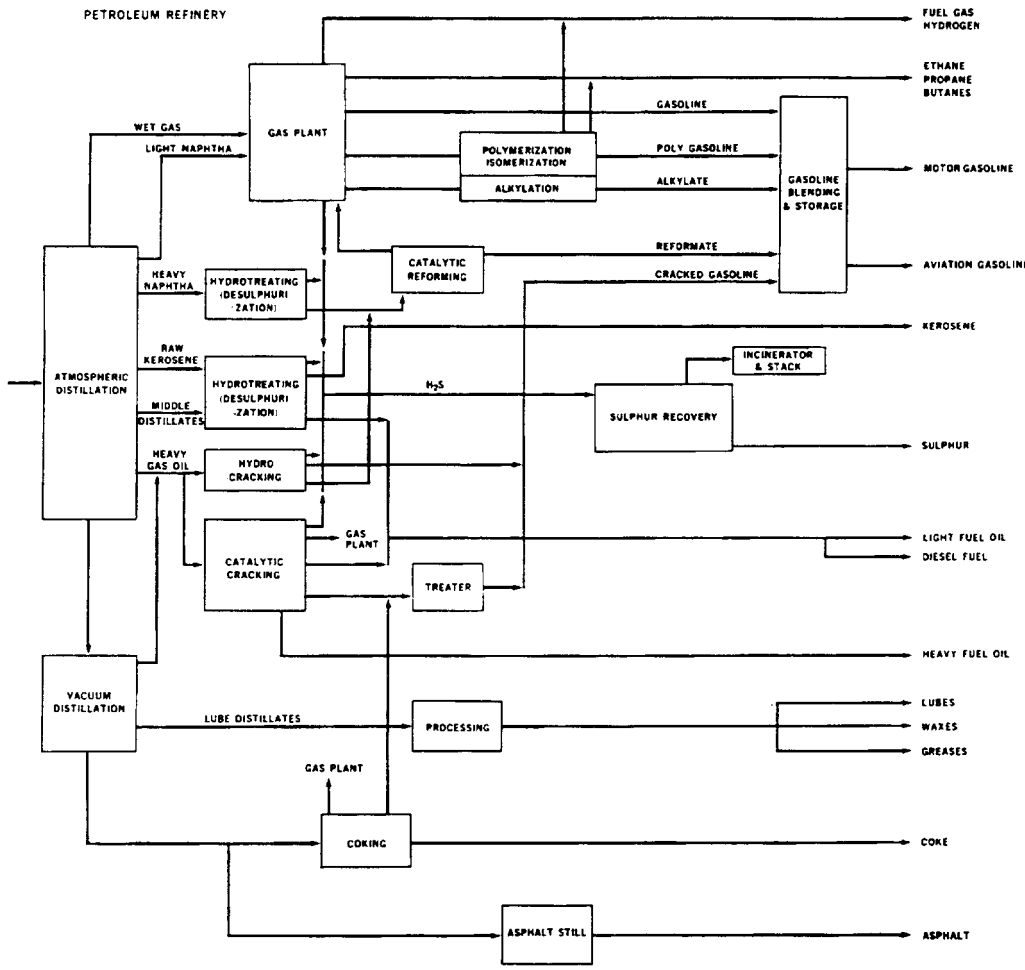


FIG. R-1 Simplified petroleum refinery process flow diagram. (Source: Environment Canada.)

by the incoming oil feed from the bottom of the reactor. Upon contact with the hot catalyst, the oil vaporizes and is cracked into smaller molecules. Vapors from the reactor are separated from the entrained catalyst and fed into a fractionator, where the desired products are removed and heavier fractions are returned to the reactor. The catalyst is deactivated by thermal degradation and through contact with heavy metals in the feed, necessitating regeneration or replacement.

2. *Hydrocracking* is basically a catalytic cracking and a hydrogeneration process. In this process, polycyclic compounds are broken to produce single ring and paraffin-type hydrocarbons. In addition, sulfur and nitrogen are removed to produce hydrogen sulfide and ammonia. These reactions occur at high temperatures and pressures, in the presence of hydrogen and a catalyst.
3. *Visbreaking* is an old process that was replaced by catalytic cracking and hydrocracking. It involves a mild thermal cracking operation designed to reduce the viscosity of the charge stock. The feed is heated and thermally cracked in the furnace. Cracked products are routed to a fractionator where the low boiling materials are separated into light distillate products, while the heavy portion may be used for coker feed or as plant fuel.

4. *Coking processes* (fluid or delayed) are used by only a few refineries in Canada. Coking is a severe thermal cracking process in which the feed is held at high cracking temperature and low pressure so that coke will form and settle out. The cracked products are sent to a fractionator where gas, gasoline, and gas oil are separated and drawn off, and the heavier material is returned to the coker.

Rearranging processes. Catalytic reforming, which is the most widely used rearranging process, improves the octane quality of gasoline obtained from crude oil. This is achieved by molecular rearrangement of naphthenes through dehydrogenation and of paraffins through isomerization and dehydrocyclization. The reformer catalyst, commonly platinum chloride on an alumina base, may also contain an activity-increasing noble metal such as rhenium. In many units, the catalyst is regenerated or replaced every 6 to 12 months. In other units, the catalyst is withdrawn continuously and regenerated on-site for further use. Refineries are more often choosing continuous reformers that do not require periodic shutdown for catalyst regeneration as conventional reformers do. The dehydrogenation and dehydrocyclization reactions produce large amounts of hydrogen as a by-product that can be used for various hydrogen-treating processes.

Combining processes. Two processes, alkylation and polymerization, are used to produce gasoline-blending stocks from the gaseous hydrocarbons formed during cracking processes.

1. *Alkylation* is the reaction of an olefin with an isoparaffin (usually isobutane) in the presence of a catalyst (either 98 percent sulfuric acid or 75 to 90 percent hydrofluoric acid) under controlled temperatures and pressures to produce high octane compounds known as alkylate. These products are separated in a settler where the acid is returned to the reactor and the alkylate is further processed. This hydrocarbon stream is scrubbed with caustic soda to remove acid and organically combined sulfur before passing to the fractionation section. Isobutane is recirculated to the reactor feed, the alkylate is drawn off from the bottom of the debutanizer, and the normal butane and propane are removed from the process.
2. *Polymerization* is a reaction that joins two or more olefin molecules. The use of this process has been declining as both the yield and quality of the gasoline product are inferior to those derived from the alkylation process. The feed must first be treated with caustic soda to remove sulfur compounds and then with water to remove nitrogen compounds and excess caustic soda. These treatments are required to protect the catalyst in the reactor. After treatment, the hydrocarbon feed is contacted with an acid catalyst in the reactor under high temperature and pressure. The catalyst is usually phosphoric acid or, in some older units, sulfuric acid. The polymerized product from the reactor is then treated to remove traces of acid.

Treating

Hydrotreating. Hydrotreating is a relatively mild hydrogenation process that saturates olefins and/or reduces sulfur, nitrogen, and oxygen compounds, along with halides and trace metals present in the feed, without changing the boiling range of the feed. This process stabilizes the product by converting olefins and gum-forming unstable diolefins to paraffins and also improves the odor and color of the products. Although there are various types of hydrotreating units, each has essentially the same process flow. The feed is combined with recycled hydrogen, heated to the

reaction temperature, and charged to the reactor. In the presence of a catalyst (metal-sulfide), the hydrogen reacts with the hydrocarbons to form hydrogen sulfide, ammonia, saturated hydrocarbons, and free metals. The metals remain on the catalyst and other products leave the reactor with the oil-hydrogen stream. The reactor products are cooled and hydrogen sulfide is removed, while hydrogen is returned to the system. The hydrocarbons are sent to a fractionator where the various products are separated. This process is ideally suited for the production of low sulfur diesel and furnace fuel oil.

Chemical treating. A number of chemical methods are used throughout the refinery to treat hydrocarbon streams. These can be classified into three groups: acid treatment, sweetening processes, and solvent extraction.

1. *Acid treatment* consists of contacting the hydrocarbons with concentrated sulfuric acid to remove sulfur and nitrogen compounds, to precipitate asphaltic or gumlike materials, and to improve color and odor.
2. *Sweetening processes* oxidize mercaptans to less odoriferous disulfides without actually removing sulfur. The most common sweetening processes are the Merox processes; others include the lead sulfide, the hydrochloride, and the copper chloride processes. In the Merox process, a catalyst composed of iron group metal chelates is used in an alkaline environment to promote the oxidation of mercaptans to disulfides using air as a source of oxygen.
3. *Solvent extraction* involves the use of a solvent that has an affinity for the undesirable compounds and is easily separated from the product. Mercaptans are extracted using a strong caustic solution. The solvent is usually regenerated by heat, steam stripping, or air blowing.

Gas treating. This process is used to remove the sulfur compounds from the various gaseous streams. Hydrogen sulfide (H_2S) can be extracted by an amine solution to produce a concentrated stream of H_2S that can be sent to a sulfur recovery plant.

Treatment by physical means. Physical methods are intermediate steps in crude oil processing operations and are often used to treat hydrocarbon streams or remove undesirable components. These methods include electrical coalescence, filtration, adsorption, and air blowing. Physical methods are applied in desalting crude oil, removing wax, decolorizing lube oils, brightening diesel oil (to remove turbidity caused by moisture), and other processes.

Deposits and compliance assessment

Refineries are held to making reports on deposits and compliance assessment. Table R-1 is a sample of this report for the Ontario region, Canada.

Blending and additives

A number of intermediate streams, called base stocks, are blended to produce a product that will meet various specifications, e.g., specific volatility, viscosity, and octane. The blending operation involves the accurate proportioning of the base stocks along with proper mixing to produce a homogeneous product.

A number of additives are used to improve the properties of the products. For example, MMT is usually added to gasoline to increase the octane number since recent regulations forbid the use of lead in gasoline. Other additives, such as anti-oxidants, anti-icing agents, and metal deactivators, are also used.

TABLE R-1 Deposits and Compliance Assessment—Ontario Region

		Refinery		
		Esso Sarnia	Petro-Canada Mississauga	Shell Corunna
A. DEPOSITS	(All guidelines and regulated deposits are for monthly averages.)			
Yearly average of daily deposits (kg/1000 m^3 of crude oil)	Guideline Deposits Esso Sarnia, P.C. Miss., Shell	Actual Deposits	Actual Deposits	Actual Deposits
Oil and Grease	17.1	0.22	4.1	0.50
Phenols	1.7	0.02	0.01	0.01
Sulfide	0.6	0.02	0.05	0.00
Ammonia nitrogen	14.3	0.70	0.86	0.03
Total suspended matter	41.1	11.3	20.5	9.7

B. COMPLIANCE ASSESSMENT

a) Number of deposits in excess of limits set in Guidelines/Regulations	M	O	D	M	O	D	M	O	D
Oil and grease	0	0	0	0	0	1	0	0	0
Phenols	0	0	0	0	0	0	0	0	0
Sulfide	0	0	0	0	0	0	0	0	0
Ammonia nitrogen	0	0	0	0	0	0	0	0	0
Total suspended matter	0	0	0	0	0	3	0	0	1
pH			0			0			0
Toxicity			0			0			0
Total	0	0	0	0	0	4	0	0	1
Percentage by region	0	0	0	0	0	80.0	0	0	20.0
Percentage of time in compliance	100	100	100	100	100	99.7	100	100	99.9
b) Number of monthly amounts exceeding the limits by:									
0 to 24%	0			0			0		
25 to 49%	0			0			0		
50 to 99%	0			0			0		
100 to 199%	0			0			0		
Over 200%	0			0			0		

M, Monthly Amount; O, One-day Amount; D, Maximum Daily Amount

Actual crude rate (1000 m^3/day)	16.3	6.1	10.4
Reference crude rate (1000 m^3/day)	19.1	5.7	11.3
Status	Existing	Existing	Existing
Number of months in operation	12	12	12
Number of tests reported	1,163	1,157	1,163

Glossary: Common Terms in the Refining Industry

Activated carbon	Carbon that is specially treated to produce a very large surface area and is used to adsorb undesirable substances.
Actual deposits	The amount of contaminants discharged in refinery effluents.
Adsorption	Attraction exerted by the surface of a solid for a liquid, or a gas, when they are in contact.

Petro-Canada Oakville			Suncor Sarnia			Esso Nanticoke			Novacor Corunna					
Guideline Deposits	Actual Deposits		Guideline Deposits	Actual Deposits		Regulated Deposits Esso, Novacor	Actual Deposits		Actual Deposits			Average Authorized Deposits		Average Actual Deposits
15.1	0.80		15.3	1.2		8.6	0.83		0.08			14.3		0.90
1.6	0.01		1.5	0.01		0.9	0.00		0.00			1.4		0.01
0.6	0.06		0.5	0.02		0.3	0.02		0.09			0.5		0.03
13.3	1.8		12.7	0.94		10.3	0.06		0.84			12.9		0.66
36.1	7.6		36.6	7.2		20.5	1.7		1.3			34.3		8.1
M	O	D	M	O	D	M	O	D	M	O	D	M	O	D
0	0	0	0	0	0	0	0	0	0	0	0	0	0	1
0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0	0	0	4
		0			0			0			0			0
		0			0			0			0			0
0	0	0	0	0	0	0	0	0	0	0	0	0	0	5
0	0	0	0	0	0	0	0	0	0	0	0			
100	100	100	100	100	100	100	100	100	100	100	100	100	100	99.9
0			0			0			0			0		
0			0			0			0			0		
0			0			0			0			0		
0			0			0			0			0		
0			0			0			0			0		
7.2			11.5			13.3	7.1		71.9					
8.8			9.0			12.5	7.9		74.3					
Existing 6.7 + Expanded 2.1			Existing 6.9 + Expanded 2.1			New	New		Existing 49.7 + Expanded 4.2 + New 20.4					
12			12			12	12							
1,163			1,163			1,160	1,145		8,114					

Aerobic bacteria	Bacteria that require free oxygen to metabolize nutrients.
Air blowing	The process used to produce asphalt by reacting residual oil with air at moderately elevated temperatures.
Altered refinery	An existing refinery at which the primary crude oil atmospheric distillation tower was replaced after October 31, 1973.

Anti-icing additive	A fuel additive used to minimize ice formation.
Anti-knock compound	Chemical compounds added to motor and aviation gasolines to improve their performance and to reduce knock in spark-ignition engines.
Antioxidants	Chemicals added to products such as gasoline and lubricating oil to inhibit oxidation.
APHA	American Public Health Association
API	American Petroleum Institute
Authorized deposits	The amount of contaminant to be discharged with the effluent of a refinery as authorized by the federal Regulations and Guidelines.
Blowdown	Removal of liquid from a refinery vessel (storage or process) through the use of pressure. The term "blowdown" is also used to refer to the actual liquid removed.
BOD	Biochemical oxygen demand. The amount of oxygen required by aerobic microorganisms to biodegrade organic matters contained in wastewater. The BOD test is used to measure the organic content of wastewater and surface water.
BPT	Best practical treatment.
Catalyst	A substance that promotes a chemical reaction without itself being altered.
COD	Chemical oxygen demand. The amount of oxygen equivalent of the organic matter required to complete chemical oxidation in an acidic medium. The COD test is used to measure the organic content of wastewater and natural water.
Cooling tower	A large structure, usually wooden, in which atmospheric air is circulated to cool water by evaporation.
CPPI	Canadian Petroleum Products Institute
Existing refinery	A refinery that began operation prior to November 1, 1973.
Expanded refinery	An existing refinery that has declared a revised Reference Crude Rate of more than 115 percent of the initial Reference Crude Rate.
Fractionator	A cylindrical refining vessel where liquid feedstocks are separated into various components or fractions.
GVRD	Greater Vancouver Regional District.
Landfill	A location where solid waste is buried in layers of earth in the ground for disposal.
Leachate	A solution resulting from the dissolving of soluble material from soil or solid waste by the action of percolating water or rainfall.
Liquid-liquid extraction	The process whereby two immiscible liquids come in contact to allow for the soluble material in the carrier liquid to be extracted in the solvent.
LPG	Liquefied petroleum gas.
Maximum daily amount	A limit set in the federal Regulations and Guidelines for a number of parameters pertaining to refinery effluents. The refinery effluent should not exceed this limit on any day of the month.
Mercaptans	A group of organosulfur compounds having the

	general formula R-SH where "R" is a hydrocarbon radial such as CH_3 and C_2H_5. Mercaptans have strong, repulsive, garliclike odors and are found in crude oil.
Monthly amount	A limit set in the federal Regulations and Guidelines for a number of parameters pertaining to refinery effluents. This limit represents the amount that should not be exceeded in the refinery effluent on a daily average basis over each month.
New refinery	A refinery that has not commenced the processing of crude oil prior to November 1, 1973.
96-hour flow-through bioassay	A test procedure required by the federal Guidelines to evaluate the acute lethal toxicity of refinery effluent to fish. The procedure consists of exposing fish to a continually renewed effluent under controlled conditions over a 96-hour period. The percent mortality of fish is observed after the four-day period.
96-hour static bioassay	A test procedure similar to the 96-hour flow-through method but in which the effluent is not renewed during the period of test.
Octane	A number indicating the relative antiknock value of a gasoline. The higher the octane number, the greater the antiknock quality.
Once-through cooling water	Water that has been circulated once through heat exchangers in order to remove heat from process streams without coming into contact with the stream.
One-day amount	A limit set in the federal Regulations and Guidelines for a number of parameters pertaining to refinery effluents. Each refinery is allowed to exceed this limit only once during a month.
Ozonation	Water treatment method that uses ozone as an oxidant to remove pollutants, i.e., chemical pollutants present in small concentrations that are difficult to remove, or to disinfect water.
Photosynthetic action	A process by which organic compounds (mainly carbohydrates) are synthesized by chlorophyll-containing plant cells. The reaction takes place in the presence of light, carbon dioxide, and water.
Priority pollutants	A list of 129 toxic pollutants having known or suspected adverse effects on human health or the environment. The United States Environmental Protection Agency (USEPA) established this list and has the mandate, under the *Clean Water Act*, to control these pollutants in wastewater discharged to the environment.
Reference Crude Rate (RCR)	The quantity of crude oil, expressed in 1000 m^3/d, declared by a refinery and used to calculate the authorized deposits.
Residual pitch	A black, heavy residue produced in the processing of crude oil.
Sour water	Water containing impurities, mainly sulfide and/or ammonia, that make it extremely harmful.

Stripping	A process for removing the more volatile components from a mixture. Generally, the hot liquid from a flash drum or tower is passed into a stripping vessel, through which open steam or inert gas is passed to removed the liquid's more volatile components.
24-hour static bioassay	A test procedure similar to the 96-hour static method but in which the percent mortality of fish is observed after a 24-hour period.
Zeolite catalyst	A catalyst that contains any of the various silicates, e.g., hydrated aluminum and calcium (or sodium) silicates, used in catalytic cracking units.

Refrigerant(s)

Refrigeration comes under heavy government scrutiny because of the potential use of halofluorocarbons that are greenhouse gases, but many times more capable of causing greenhouse effects than CO_2 (carbon dioxide). Information on refrigerants can also be found in other sections (such as Ozone and Pollutants, Chemical) in this book.

Refrigerant Technology*

Alternative refrigerants

Although the selection criteria described above appear to be very restrictive, there are several refrigerants available that will meet or exceed all the requirements.

In the decade following the signing of the Montreal Protocol, an extensive search was undertaken for alternatives that would replace chlorofluorocarbons (CFCs). Among several groups of alternatives, two—the hydrochlorofluorocarbons (HCFCs) and the hydrofluorocarbons (HFCs)—are the most useful. The HCFCs were developed to serve as interim replacements for CFCs. They are used in existing equipment for the remainder of the equipment life and in new systems, until a permanent replacement becomes available. The HCFCs contain chlorine and therefore are still ozone-depleting substances. But their ozone-depletion potential (ODP) is less than that of CFCs. For example, the ODP of R-12 is 1.0, while one of its interim replacements, R-409A, a blend of three HCFCs (R-22, R-124, and R-142b), has an ODP of 0.05. Some of the HCFCs, such as R-22, offer excellent performance characteristics and were in use even before the ozone depletion issue was raised. HFCs were developed to offer long-term alternatives to CFCs and HCFCs. They do not have any chlorine and hence have zero ODP. Many long-term alternative refrigerants are mixtures of two or more HFCs.

Refrigerant mixtures

Widely used in the past, refrigerant mixtures have received renewed interest from designers in the search for new alternatives. Mixtures offer the most attractive solution, since by mixing two or more refrigerants a new working fluid with the desired characteristics can be created. For example, by adjusting the composition of a blend containing a high-pressure and a low-pressure refrigerant, the vapor pressure of the final fluid can be tailored to match that of the CFC or HCFC being replaced. By blending refrigerants, it is possible to create new blends that are nonflammable but still contain moderately flammable refrigerants. In other cases,

* Source: Adapted from extracts from Gopalnarayanan, "Choosing the Right Refrigerant," *Mechanical Engineering*, ASME, October 1998.

blends are created to improve system characteristics, such as compressor discharge temperature, or to improve lubricant circulation by adding a more lubricant-miscible refrigerant to the blend.

Refrigerant mixtures fall into two major groups, zeotropes and azeotropes. In zeotropic mixtures, as the name implies, the liquid-vapor phase change does not occur at a constant temperature (at a fixed pressure) as in the case of pure fluids, but over a range of temperatures. Azeotropic mixtures, on the other hand, boil at a single temperature, much as a pure fluid does. The temperature alteration during phase change is commonly called temperature glide. An intermediate class of mixtures, dubbed near-azeotropes, are really zeotropes that are close to an azeotropic composition, and have a very small temperature glide. In some cases, a pair of fluids may be called near-azeotropes if the maximum temperature glide for the pair is very small, even though they may not form an azeotrope. A 50/50 mixture of R-32 and R-134a, for example, forms a zeotropic blend with a temperature glide of about 5°C at about 950 kPa. A 50/50 blend of R-115 and R-22 will form a perfect azeotrope, while a 50/50 blend of R-32 and R-125 will form a near-azeotrope with a temperature glide of about 0.1°C.

The higher temperature glides in zeotropes, in certain applications, offer the potential for higher efficiencies. By more closely matching the temperature profiles of the refrigerant blend and the fluid being cooled, engineers can reduce the external heat transfer irreversibilities. But this advantage cannot be fully utilized, because there is now an increased mass-transfer resistance in the process of boiling and condensation. The temperature glide also results in a phenomenon called fractionation, whereby higher and lower boiling components tend to separate; the vapor will be richer in the fluid with a lower boiling point and vice versa. In the event of a vapor leak in the system, the leaking vapor will be richer in the more volatile component of the blend. This could result in a change in the mixture's composition, which is undesirable, since it could alter the system's performance. However, performance changes will result even for pure fluid if there is a refrigerant leak, since change in refrigerant inventory will cause a change in performance. Class 1 zeotropic blends are formulated to be nonflammable even under the worst fractionation scenario.

Azeotropes and near-azeotropes, on the other hand, do not have a temperature-glide-induced problem. However, azeotropes and near-azeotropes that match the vapor-pressure characteristics of the CFC or HCFCs they replace are nonexistent. Hence, as can be seen in Fig. R-2, a multitude of alternative blends are zeotropes.

Figure R-2 shows a broad outline of different applications, the refrigerants that were used in the past, and the interim and long-term alternatives available. According to the American Society of Heating, Refrigerating, and Air-Conditioning Engineers' (ASHRAE's) refrigerant-numbering scheme, refrigerants having the form R-4XXX are zeotropic blends of two or more refrigerants, while those with the form R-5XXX are azeotropes. R-134a was a natural answer as a long-term R-12 replacement, since it met almost all the selection criteria that were imposed. Although not compatible with many of the elastomers or lubricants used with R-12, it provided the single-fluid solution, and the issues of compatibility and miscibility were addressed by selecting alternative elastomers and lubricants, respectively. The changeover to R-134a in place of R-12, however, required a concerted effort and considerable expense on the part of equipment manufacturers.

Comfort conditioning

R-22 is being used extensively in comfort conditioning. (See Fig. R-2.) Unlike R-12, however, there is no known single-fluid answer that would meet all the selection criteria. One of the alternatives proposed is propane. Although propane has a low global warming potential (GWP) compared to other alternatives, it may have a

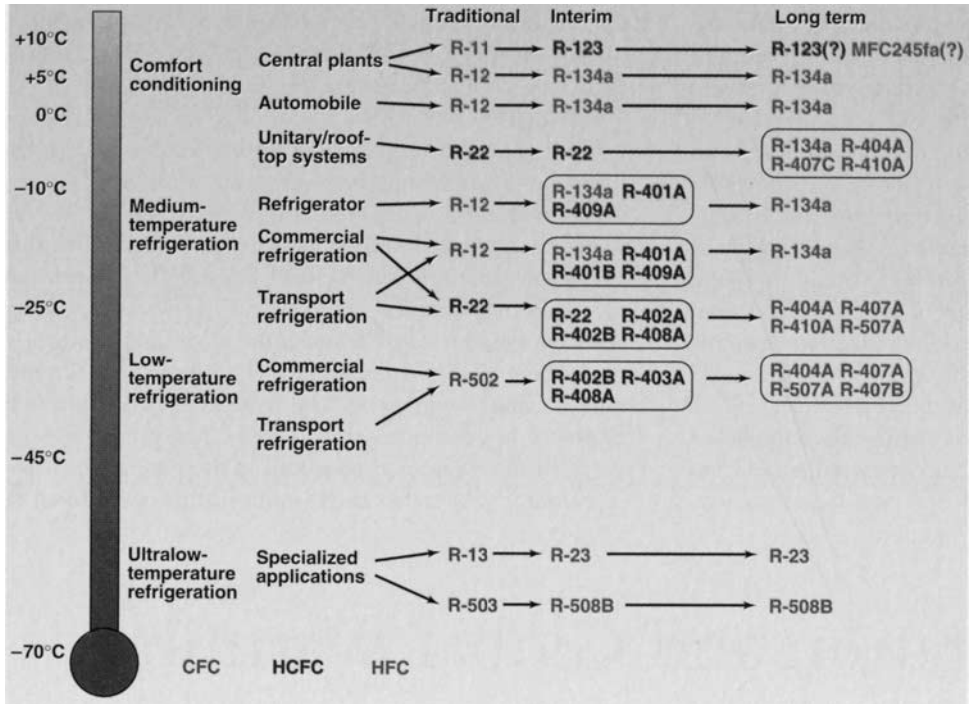


FIG. R-2 For a variety of applications, a number of interim and long-term replacements are available. (Source: Gopalnarayanan.)

higher total equivalent warming impact (TEWI). In addition, being flammable, it will require additional safeguards, resulting in a more expensive system.

Two alternatives that have become acceptable as replacements for R-22 in comfort-conditioning applications, R-407C and R-410A, are both refrigerant mixtures containing HFCs. HFCs are immiscible with conventionally used mineral or alkyl benzene lubricants, and so require either polyalkylene glycol or polyol ester lubricants.

R-407C is a ternary zeotropic mixture of R-134a, R-125, and R-32. One of the advantages of R-407C is that its vapor pressure is only slightly higher than that of R-22, so it can be used as a "retrofit" fluid in R-22 systems with only a lubricant change. It also has performance comparable to R-22. One disadvantage of R-407C is its temperature glide, which can be as high as 6 to 7°C at typical system pressures, leading to problems related to the fractionation discussed earlier.

The other alternative, R-410A, is a near-azeotropic mixture of R-32 and R-125. The vapor pressure of this fluid is almost 50 percent higher than that of R-22, and thus R-410A cannot be used in existing systems. Newly built systems will have to be redesigned to handle the increased pressures. However, the higher pressure and the fact that the fluid is almost an azeotrope provide an opportunity to design more compact systems with greater efficiency.

Which is a better fluid? Again, the choice depends on the application and local regulations. For example, in certain European countries, the phaseout date for R-22 use in new equipment was 1998 (as opposed to 2010 in the United States). In this instance, there may not be a newly designed R-410A unit available for changeover. In some cases, it may not be economically viable to redesign the entire system, considering that even the manufacturing plant will require redesign, while in others, the opportunity for better system performance with R-410A may justify the investment.

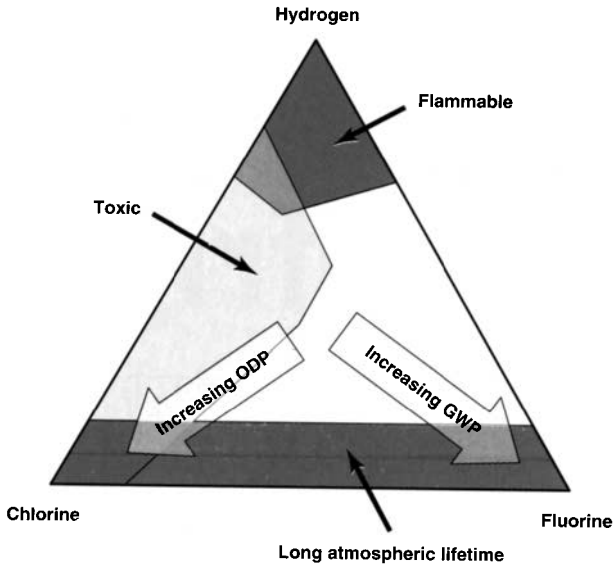


FIG. R-3 When considering appropriate halocarbon alternatives, engineers need to strike a balance between key constraints. (Source: Gopalnarayanan.)

Striking a balance

One of the major challenges in identifying halocarbon alternatives to ozone-depleting substances is to strike a balance between the various affecting factors, namely ODP, TEWI, toxicity, flammability, performance characteristics, and cost (see Fig. R-3). Due to halocarbons' level of GWP, some European countries have begun exploring alternatives to them. Certain naturally occurring compounds, such as hydrocarbons, carbon dioxide, and ammonia, would need significant processing or refining to be used as refrigerants. These processed refrigerants, as well as natural refrigerants such as air or water, continue to be considered and in some cases applied as viable alternatives to synthetic refrigerants. In the United States, there is considerable resistance to use of hydrocarbons or ammonia in applications that use halocarbons. This is due to liability concerns arising from their flammability and, in the case of ammonia, toxicity. Even considering their higher direct GWP, halocarbon refrigerants appear to offer the most complete solution, by fully meeting all the safety, performance, economic, and environmental constraints that are placed on today's refrigerants.

Refrigeration and global warming

If HFCs are the obvious solution to the refrigerant quandary, why is there so much activity in developing alternative refrigerants and technologies? One reason, it is thought, is the higher GWP of HFCs.

For purposes of comparison, R-134a has a GWP of about 1300, while that of the hydrocarbon refrigerant isobutane is about 11. Both of these refrigerants are used as replacement fluids for R-12 in domestic refrigerators. Is the difference in GWP significant? The answer lies in realizing that the contribution to global warming comes not only from the refrigerant that may leak from the system, but also from the amount of carbon dioxide that will be emitted at the power plant in producing the energy needed to run the compressor.

The effect of a refrigeration system on global warming is more accurately described by the popular concept of TEWI, which takes into account both of these

contributions to global warming. TEWI, therefore, ties in the energy efficiency of the system (and that of the power plant) in determining the effect on global warming. In the case of the refrigerator using R-12, only 4 percent of the TEWI comes from direct emission of the refrigerant. Considering that R-134a has only about one-fifth of the GWP of R-12, the contribution to TEWI of using either R-134a or isobutane makes a difference of only 1 percent over the lifetime of the refrigerator.

It is estimated that HFCs currently account for only 0.06 percent of human-induced global warming. Even assuming that HFCs will be fully emitted from the systems in which they are used, estimates place their contribution at about 0.6 percent in 2010, less than 2 percent in 2030, and approximately 2.6 percent by 2100.

Approaches to the global-warming issue taken by various countries are different. In the United States, the consensus is to reduce TEWI by attacking the energy efficiency of the equipment, since that seems to contribute the most to the TEWI. In some European countries, however, emphasis is placed only on reducing the direct contribution from the refrigerants. As in the case of politically sensitive issues like this, the approaches taken by various governments are not necessarily always based only on technical merit.

Natural Refrigerants*

The search for new and environmentally benign refrigerants to replace the existing CFCs and HCFCs has led to the introduction of HFCs. However, HFCs have a much higher global-warming potential and higher costs than natural refrigerants. These concerns have spurred calls for the investigation of alternatives to HFCs. Some environmentalists would like the refrigeration industry to bypass HFCs and employ natural refrigerants as soon as possible.

Natural refrigerants are working fluids based on molecules that occur in nature. Examples are such substances as air, water, ammonia, hydrocarbons, and carbon dioxide. Nevertheless, the actual fluids used in refrigeration systems may very well be synthesized and will not necessarily be extracted from nature. Ammonia, for instance, is synthesized in large quantities, and hydrocarbons undergo an extensive chemical processing procedure. Still, the cost of these fluids is much lower than that of HFC refrigerants, and they do not affect the environment in an unknown way. Also, the amount of fluid produced is negligible compared with the amount available in nature.

Studies of natural refrigerants continue. For example, Annex 22 of the International Energy Agency implemented a three-year project, Compression Systems with Natural Working Fluids, in 1995. Air, water, ammonia, hydrocarbons, and carbon dioxide have a low or zero direct global-warming potential and zero ODP, as shown in Table R-2.

Development of refrigeration systems

Since the first vapor-compression refrigeration system was invented by Jacob Perkins in 1834, more than 50 chemical substances have been used as refrigerants in refrigeration and air-conditioning systems. The development of refrigeration systems using CO_2 as the refrigerant started in 1866, when an ice production machine that used CO_2 was invented by Thaddeus S.C. Lowe. In 1880, the first CO_2

---

*Source: Adapted from extracts from Hwang, Ohadi, and Radermacher, "Natural Refrigerants," *Mechanical Engineering*, ASME, October 1998.

TABLE R-2 Characteristics and Properties of Some Refrigerants

Refrigerants	R-12	R-22	R-134A	NH_3 (R-717)	C_3H_8 (R-290)	CO_2 (R-744)
Natural substance	No	No	No	Yes	Yes	Yes
TLV (ppm)	1,000	1,000	1,000	25	1,000	5,000
IDLH (ppm)	50,000	—	—	500	20,000	50,000
Amount per room volume (vol/kg/m^3)	4.0/0.2	4.2/0.15	—	—	0.44/0.008	5.5/0.1
Flammable or explosive	No	No	No	Yes	Yes	No
Flammability limits in air (volume%)	—	—	—	15.5/27	2.2/9.5	—
Toxic/irritating decomposition products	Yes	Yes	Yes	No	No	No
Approximate relative price	1	1	3–5	0/2	0.1	0.1
Molar mass	120.9	86.5	102.0	17.0	44.1	44.0
Refrigeration capacity at 0°C (kJ/m^3)	2,740	4,344	2,860	4,360	3,870	22,600

TLV: Threshold limit value (the refrigeration concentration limit in air for a normal 8-hour workday; it will not cause an adverse effect on most people).
IDLH: Immediately dangerous to life or health (maximum level from which one could escape within 30 minutes without impairing symptoms or any irreversible health effects).

compressor was designed by Franz Windhausen. After the late 1800s, the use of CO_2 refrigeration systems increased. As a result of continuous efforts to improve efficiency, two-stage CO_2 machines were developed in 1889 by the J. & E. Hall Co. in Great Britain, and the multiple-effect CO_2 cycle was developed by G. T. Voorhees in 1905.

Meanwhile, Thomas Midgley, Jr., and Albert Henne published a paper in 1930 on fluorochemical refrigerants, as a result of a search for stable, nontoxic, nonflammable, and efficient refrigerants. In 1931, dichlorodifluoromethane, CFC-12, was commercially produced. After the introduction of fluorochemical refrigerants, the early refrigerants, including CO_2, were replaced by many other CFCs and HCFCs. This led to a drastic decline in the use of refrigerants other than CFCs and HCFCs after World War II. Only ammonia remained in use, and its application was confined to large industrial systems. In recent years, though, natural refrigerants have gained considerable attention as alternative refrigerants for mobile air conditioners in Europe.

Varieties of natural refrigerants

Although natural refrigerants were used extensively in the early years of refrigeration technology, a number of technical and safety challenges caused them to be abandoned when CFCs became available. These challenges still exist today for air, water, ammonia, hydrocarbons, and carbon dioxide.

Air is used extensively as a refrigerant in aircraft cooling. Its advantages are that open systems require fewer heat exchangers, aircraft have compressed air available already, and systems tend to be low in weight. Its efficiency, however, is quite poor. Nonetheless, German railways have installed air cycles in the latest generation of high-speed trains because of weight concerns and, most importantly, because maintenance time is very short. There is no cumbersome and time-consuming refrigerant reclamation and no system evacuation is required.

Water has the potential to be a very efficient refrigerant, but because it requires operation in a deep vacuum, its vapor density is quite low. This leads to costly large-volume vacuum tanks that must house all the machinery, such as heat exchangers and compressors. Furthermore, water's pressure ratio is very high, imposing additional challenges for compressors that must operate in a deep vacuum.

The only applications where water is used as a refrigerant on a commercial basis are in large-capacity lithium-bromide water-absorption chillers. Over the last

decade, a few large-scale water-vapor compression systems have been used commercially. One such system provides cooling to the LEGO factory in Denmark, another to a mine in South Africa. In both cases, open systems are employed, and the need for heat exchangers is eliminated by using direct-contact heat exchange. Thus, the chilled water that circulates through the facility is also used as the refrigerant. Although various research projects are underway worldwide, a demanding compressor technology, the need for vacuum pumps, and degassing remain great challenges in terms of cost and design.

Ammonia is also a very good refrigerant and is used to a significant extent in large warehouses. Ammonia is toxic and, under certain limited conditions, flammable and even explosive. However, with its intense, pungent odor, it is a self-alarming refrigerant. Ammonia has emerged as a refrigerant for water chillers in Europe. These units are entirely self-contained, including a gas-tight cabinet that houses the entire unit and a water tank to dissolve any ammonia in case of a leak. These measures, to be sure, increase costs considerably.

Hydrocarbons are excellent refrigerants, but they are also flammable and explosive. In North America, any flammability risk is unacceptable, but some countries in Europe and elsewhere have less-stringent liability laws. Since the mid-1990s, virtually all refrigerator production in Germany has used hydrocarbons as the working fluid. Some heat pump manufacturers whose systems are installed entirely outdoors have followed suit, and some commercial installations have recently become publicly known. Nevertheless, the danger of fire remains an overriding concern. To address this challenge with safety features, the cost of a system has to be increased by about one-third.

Carbon dioxide is a refrigerant that operates at very high pressures in a transcritical cycle for most operating conditions. Thus the refrigerant condenser of a conventional refrigeration system serves now as a cooler for supercritical fluid. Only after the expansion process is liquid carbon dioxide available to provide cooling capacity through evaporation. Because of the nature of the transcritical cycle, the efficiency of carbon dioxide is quite poor. However, this is its only disadvantage. All the other characteristics of carbon dioxide are very favorable. It is environmentally safe, has very low toxicity, and allows for extremely compact systems. The vapor pressure of CO_2 is approximately seven times higher than that of R-22. Moreover, the supercritical CO_2 has a higher density than subcritical fluids, so there is potential to reduce the size of hardware. There are indications that with modern materials and technologies, the weight of CO_2 heat exchangers can be reduced considerably, especially for tap water heating, with essentially the same performance.

Gustav Lorentzen and his colleagues revived research on the CO_2 cycle in the 1990s, especially for mobile air-conditioning applications. This group has focused on the experimental evaluation and thermodynamic modeling of mobile air-conditioning systems. Meanwhile, a group of researchers in Germany has focused on railway air systems. In 1994, European automobile manufacturers launched a joint research project, Refrigeration and Automotive Climate Systems under Environmental Aspects (RACE), to investigate an air-conditioning system with the natural refrigerant CO_2. Most efforts conducted so far with respect to CO_2 have focused on experimental evaluation and thermodynamic modeling for R-12 and R-22 replacement.

Investigations of prototypes for automotive air conditioning and some other applications show that well-designed systems can actually perform at levels reached by other refrigerants. One method that would greatly enhance the efficiency of CO_2 cycles is the use of an expander instead of an expansion valve. An expander produces work that can be fed back into the compressor or used otherwise. In

TABLE R-3 Thermophysical Properties of CO_2 at –25/–10/5°C

Refrigerant	R-22	CO_2 (R-744)
Saturated pressure (MPa)	0.201/0.354/0.584	1.683/2.649/3.969
Latent heat (kJ/m^3)	223.5/212.8/201.0	293.2/258.6/215.0
Saturated liquid density (kg/m^3)	1,362.0/1,314.7/1,264.3	1,054.2/982.9/927.4
Saturated vapor density (m^3/kg)	8.96/15.3/24.8	43.9/71.2/114.6
Saturated liquid specific heat (kJ/kg-K)	1.113/1.144/1.184	2.113/2.306/2.726
Saturated vapor specific heat (kJ/kg-K)	0.649/0.699/0.761	1.221/1.519/2.160
Saturated liquid viscosity (μPa-s)	286.1/242.6/206.3	156.8/123.8/95.4
Saturated vapor viscosity (μPa-s)	10.38/11.01/11.65	12.76/13.83/14.75
Saturated liquid thermal conductivity (mW/m-K)	113.6/105.4/97.3	141.3/123.2/105.3
Saturated vapor thermal conductivity (mW/m-K)	8.32/9.02/9.79	14.8/17.7/22.8
Surface tension (mN/m)	15.5/13.2/11.0	9.6/6.4/3.5

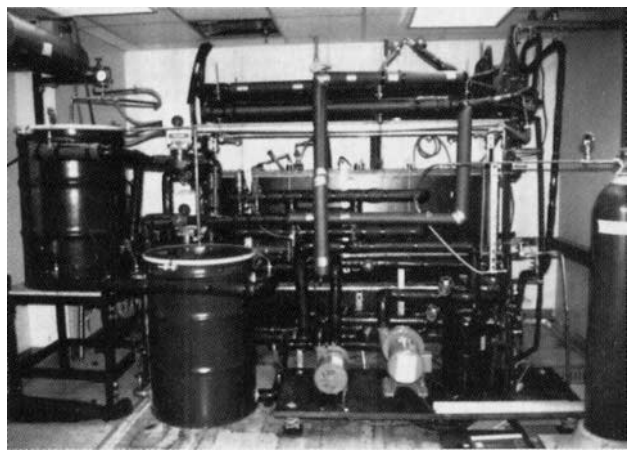


FIG. R-4 This is the experimental loop for a recent CO_2 cycle performance and heat-transfer study done at the University of Maryland in College Park. (Source: Hwang, Ohadi, and Radermacher.)

conventional systems, the expander is beginning to emerge in large-capacity units. However, any kind of machinery added to a cycle will increase cost to an unacceptable degree. Therefore, most research and development projects underway today do not include an expander.

R-22 and carbon dioxide

Thermophysical properties of R-22 and CO_2 are compared for the saturation temperatures of –25, –10, and 5°C in Table R-3. The evaporation pressure of CO_2 is four to eight times higher than that of R-22. The saturated liquid density of CO_2 is approximately 70 percent that of R-22, while the saturated vapor density of CO_2 is approximately five times that of R-22. The higher density offers the opportunity to reduce heat exchanger size and weight. CO_2 has better heat-transfer characteristics (higher latent heat, specific heat, and thermal conductivity) and lower viscosity than R-22. See Fig. R-4.

The first approach to comparing the refrigerants is to compare their ideal cycle performance. An ideal cycle is defined as a refrigeration cycle that has zero approach temperatures and minimum or no pressure drops in the heat exchangers. It should also have isentropic compression, no subcooling (for R-22), and no suction superheating.

This ideal cycle is applied to an R-22 refrigeration simulation model (REFSIM) and a CO_2 simulation model (CO_2SIM). Overall, the ideal-cycle coefficient of performance (COP) of CO_2 is only 50 to 60 percent of R-22's at various chilled water temperatures and various gas-cooler cooling water temperatures.

However, the COP of the actual cycle for the CO_2 water chiller is 94 to 106 percent of the existing R-22 water chillers, based on recent experimental work. This difference between the ideal cycle and actual cycle is analyzed using refrigeration cycle models. The actual cycle is different from the ideal cycle because of the irreversibility of the actual process, which can be attributed to the following parameters: compressor efficiency, approach temperature in heat exchangers, pressure drop in heat exchangers, and degree of subcooling and superheating. In a comparison of the effects of these parameters on the cycle COP change for both refrigerants, the compressor efficiency for both refrigerants has the largest impact on the COP drop. Moreover, this parameter has more impact for R-22 than CO_2, in absolute terms. The effects of the suction gas heating by the motor, as well as mechanical losses, are much less for CO_2 than R-22, due to the higher specific heat of CO_2. The approach temperature and degree of subcooling in the condensing process have the second and third largest impacts on the COP drop of R-22. This difference is a result of the better heat transfer of CO_2. The overall heat-transfer coefficient of CO_2 during the gas-cooling process is approximately double that of R-22 during the flow-condensation process. Another important parameter is the sensitivity of temperature change to the pressure drop. CO_2 is less sensitive than R-22. The more sensitive case has a larger approach temperature and COP drop.

Thus, it can be concluded that the beneficial thermophysical properties of CO_2 contribute considerably to its good overall COP. It is therefore conceivable that CO_2 may be a promising candidate as a viable refrigerant. Extensive research work is required to prove feasibility and, if successful, an industrywide development effort will be required to make this fluid succeed.

Natural refrigerants still have a number of technical and safety challenges to overcome, and each has its unique advantages and disadvantages. Although it is not clear yet which one is the dominant candidate as an alternative refrigerant, CO_2 offers a clear advantage over CFCs and HCFCs from the environmental impact (ODP and GWP) standpoint. In addition to its environmental advantages, CO_2 also offers certain attractive thermal characteristics that can help it provide substantial potential as a long-term replacement if energy efficiency challenges can be addressed.

Design Constraints*

There are five major factors that design engineers must consider in selecting a refrigerant for a particular application: performance, safety, reliability, environmental acceptability, and simple economics. In assessing these areas, design engineers should keep in mind that the relative importance of each depends upon the application and, of late, government regulations, which differ in each country.

Two of the primary criteria in the performance of a refrigeration system are refrigeration capacity and efficiency. The refrigeration capacity is the amount of cooling that the system can produce for a given volumetric flow rate of the refrigerant. Volumetric capacity is dependent not only on the latent heat of

* Source: Adapted from extracts from Gopalnarayanan, "Choosing the Right Refrigerant," *Mechanical Engineering*, ASME, October 1998.

vaporization but also on the density of the refrigerant vapor that enters the compressor.

One good indicator of a refrigerant's capacity is its normal boiling point. The higher the boiling point, the lower the fluid's volumetric cooling capacity. On the other hand, fluids with higher boiling points tend to have greater efficiency. However, they also tend to have a higher pressure drop and, in some cases, lower heat-transfer coefficients. Thus there is a need for compromise. Other refrigeration system requirements include appropriate operating pressure, smaller compressor size, and lower compressor-discharge temperature.

As far as safety is concerned, toxicity, flammability, and pressure govern the proper use of a refrigerant. Refrigerants are broadly classified, based on their toxicity and flammability, by ASHRAE Standard 34-1997. For toxicity, the classification is based on the chronic exposure limits (the level to which an individual can be exposed over his or her working life without ill effects), which are defined by the threshold limit value (TLV), the maximum exposure at any given time, and the time-weighted average (TWA). Refrigerants with no identified toxicity at concentrations less than 400 parts per million (ppm) based on TLV and TWA are considered Class A, while those that show evidence of toxicity at concentrations below 400 ppm are Class B. Most of the refrigerants in use are Class A. However, there are a few Class B refrigerants, such as R-123 and ammonia, that need additional equipment safety.

Regenerator

This term is normally used in conjunction with gas turbines. The thermal efficiency of a gas turbine can be increased by using the exhaust heat from the turbine to heat the gas-turbine compressor discharge air. *See also* Cogeneration; Turbines, Gas.

S

Seals; Gas Seals

The term *seals* is commonly used to refer to seals in pumps and other turbomachinery. The more complex the application, the more complex the seal. Instructions on seal working methods, maintenance, and so forth are provided with seal OEM (original equipment manufacturer) manuals. In the case of simple pumps, the process engineer may have the option of using different bearings and seals than the ones that came with the original equipment. An example is simple grease-packed bearings that are "standard" for a specific shaft size. The best source of information for standards seals is seal vendors' catalogues.

In special applications that are potentially poisonous and explosive, the seal-oil system design may be customized with a purge-oil system that provides a buffer against the hazard and prevents leakage. Major seal vendors have seal models that will have appropriate jets and clearances for these applications.

Gas Seals*

Figures S-1 and S-2 illustrate the standard and advanced product features of gas seals.

Rotating face centering. In both standard and advanced configurations, the rotating face is centered by a spiral spring to ensure concentricity with the seal shaft sleeve. The spiral centering spring is designed to provide a centering force that always exceeds the eccentric force. The spiral centering spring allows for even force distribution around the entire inner diameter of the rotating face. This even distribution minimizes radial stresses in the rotating face.

Shaft sleeve centering. The shaft sleeve is centered utilizing O-rings as a standard where size and speed permit. The O-ring provides ease of installation and enhanced dampening characteristics. An engineered centering strip is used on seals that exceed the size and speed permitted for O-rings. The engineered centering strip is fully captured in a dovetail groove, ensuring positive retention during seal installation and removal. The centering strip is engineered to provide proper force to maintain concentricity without damaging the compressor shaft. The centering strip is available as an option on any seal.

Shrouded rotor option. Shrouding the rotor provides extra protection in the unlikely event of a catastrophic seal failure. This design features a patented distributed-load arrangement utilizing drive springs. These drive springs ensure multiple-point contact across the length of the spring and at the multiple spring locations around the rotor. This further increases the reliability of the GASPAC™ (specific to this information source) compressor gas seal. See Fig. S-3.

Sealing options. The GASPAC compressor gas seal line features a choice of the patented advanced pattern groove unidirectional face pattern and the bidirectional,

* Source: Flowserve, USA. Adapted with permission.

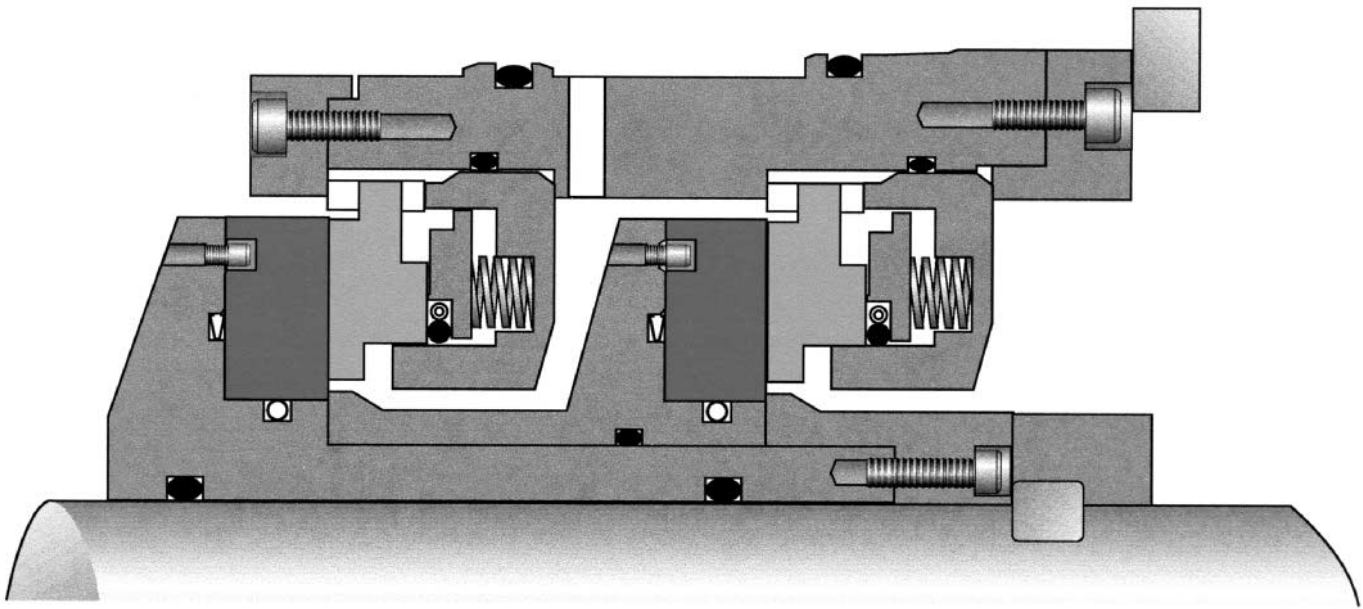


FIG. S-1 Standard features, GASPAC® compressor gas seal. (Source: Flowserve.)

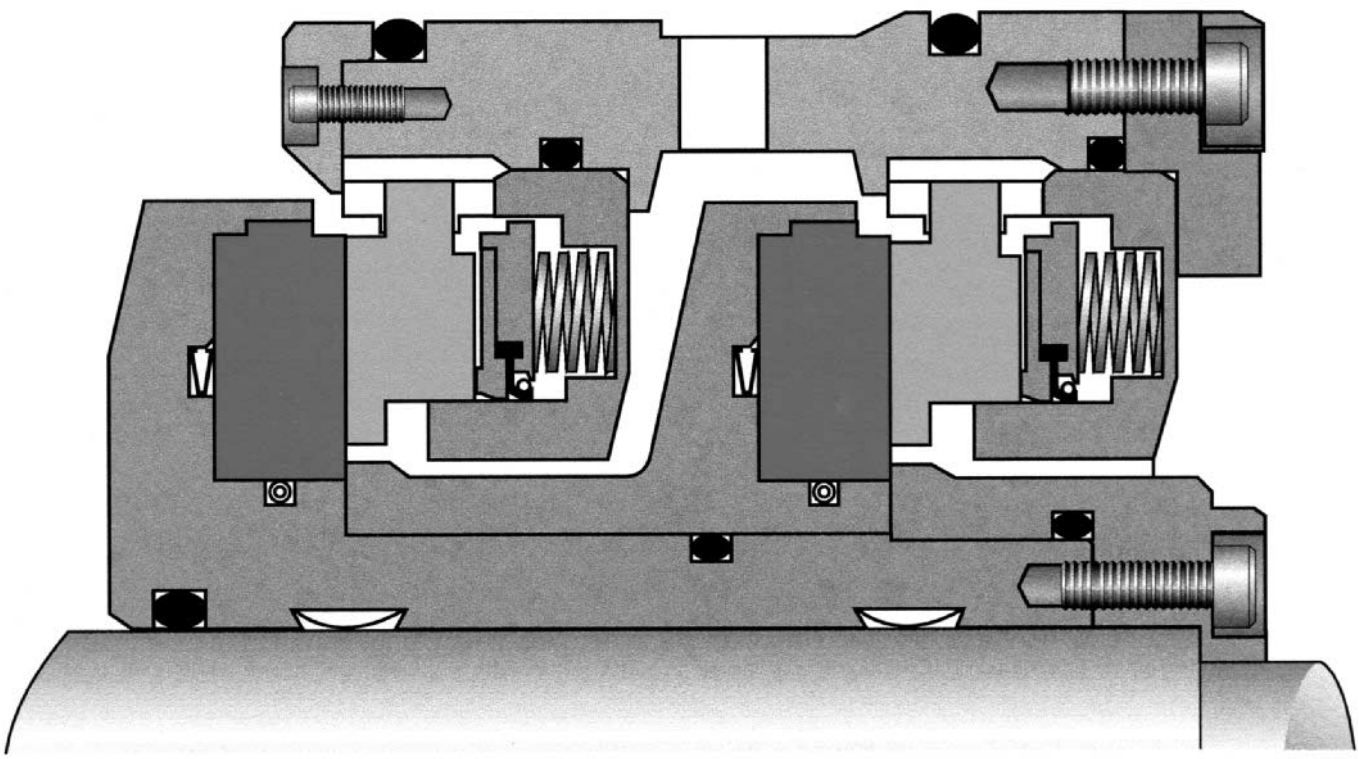


FIG. S-2 Advanced product features, GASPAC® compressor gas seal. (Source: Flowserve.)

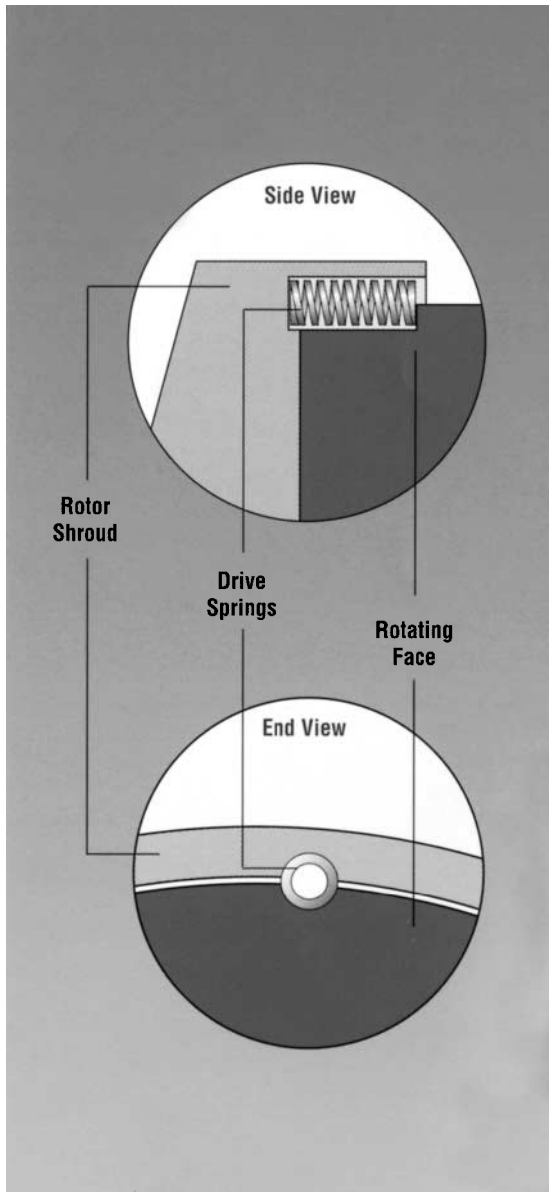


FIG. S-3 Drive springs/shrouded motor option. (Source: Flowserve.)

symmetrical "T" groove face pattern, and a choice of dynamic sealing options, all within a universal housing.

Advanced pattern groove (APG): Unidirectional face pattern. The unidirectional advanced pattern groove face represents a breakthrough in dry-running gas seal face technology. The APG incorporates specially designed tapered grooves that become progressively more shallow as they reach the circumferential groove. The APG design outperforms traditional spiral groove designs with lift-off at lower speeds, low-pressure hydrostatic lift, and better film stiffness performance. See Fig. S-4.

- Under static conditions, the APG allows the sealed gas pressure to penetrate deeply across the face. The circumferential groove next to the sealing dam ensures

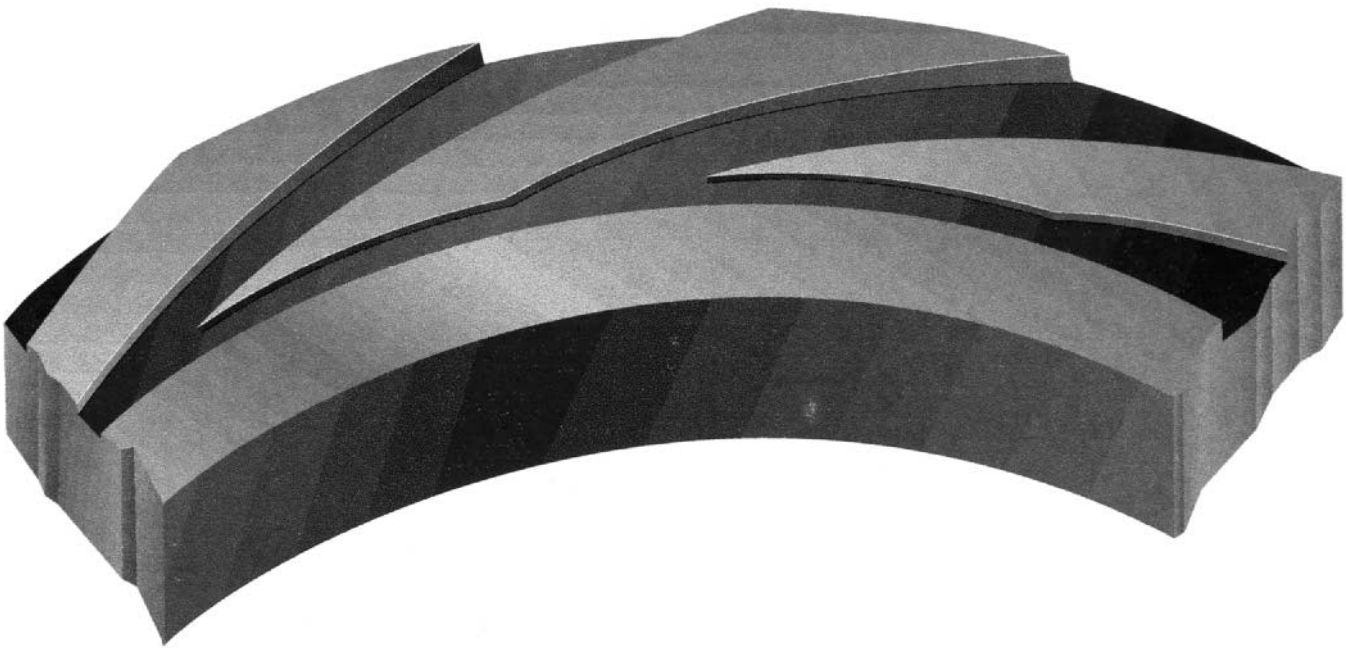


FIG. S-4 Unidirectional face pattern. (Source: Flowserve.)

uniform pressure distribution. The combination of these design features provides extraordinary hydrostatic lift, resulting in low torque at startup.

- During dynamic operation, any face pattern creates additional pressure to separate the faces so they are noncontacting. The APG extends farther across the face than conventional unidirectional patterns, providing early lift-off and better performance at extremely low speeds. The tapered groove depth of the APG allows the faces to rapidly adjust, providing stable operation during changing process conditions. The deeper grooves at the face periphery pump the sealed medium toward the center dam, developing pressure to cause hydrodynamic lift. The APG operates with noncontacting seal faces, thus keeping parasitic horsepower requirements low.

Symmetrical T-groove (SMT): Bidirectional face pattern. The GASPAC seals use bidirectional technology. The SMT also provides increased protection with a unique bidirectional T-groove face design, and can operate in a clockwise or counterclockwise rotation. This attribute provides optimum protection from reverse rotation. It also reduces the need for separate spare seal cartridges for each end of a beam-type compressor, because the same seal can operate in either end of the compressor. See Fig. S-5.

- In static operation, the aerostatic balance of the SMT results in face separation above 30 psig. Optimized face balance allows for stable operation and minimal leakage rates.
- During shaft rotation in either direction, gas flows into the symmetrical T-grooves and is pumped circumferentially toward the edge of the groove. Stagnation of gas flow at the edge builds pressure and results in hydrodynamic lift-off—even at low peripheral speeds above 6.6 ft/sec. The unique design of the SMT also provides lower leakage than conventional patterns. The SMT also operates with noncontacting seal faces, thus keeping parasitic horsepower requirements low.

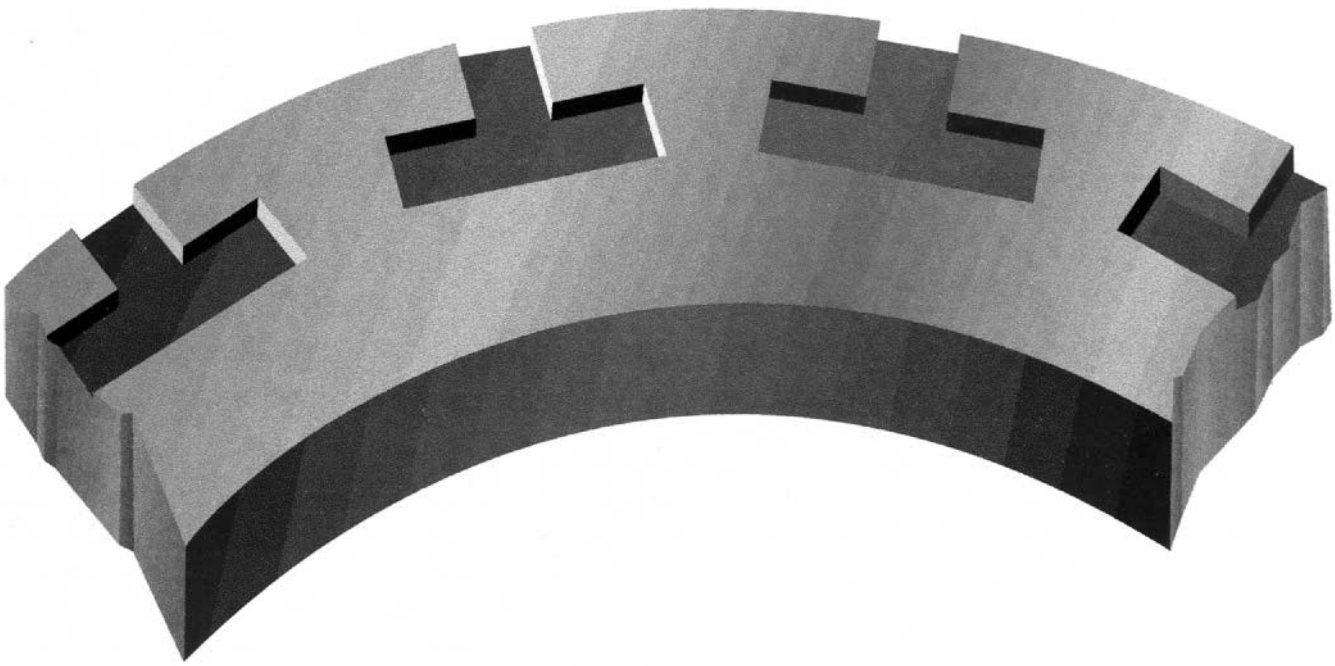


FIG. S-5 Bidirectional face pattern. (Source: Flowserve.)

Dynamic seal options. While choosing the right seal face design is essential, dependable secondary seals are just as important to the reliable operation of any gas seal. This information source offers a spring-energized O-ring seal within regular operating range, and a spring-energized polytetrafluoroethylene (PTFE) J-ring seal for extended operating range. Innovations in secondary seals are designed to eliminate the causes of O-ring hangup, the most common operational challenge in other compressor gas seal designs.

Spring energized standard design: O-ring. A proprietary dynamic secondary seal arrangement does not squeeze the O-ring in a fully confined groove, as in conventional gas seals. Instead, a 90+ Durometer™ (specific to this information source) dynamic O-ring is energized with the aid of a garter spring located on its outside diameter. While closing springs provide the O-ring axial compression between the stator and spring retainer, the garter spring provides adequate radial compression to effectively seal the gas medium. Live loading of the dynamic O-ring provides constant sealing force while compensating for O-ring cross-section irregularities or swelling due to temperature or chemical reaction. The O-ring conforms to the sealing surface when inconsistencies exist on the surface or in the presence of dirt or deposits on the sealing surface. Drag and seal hangup are greatly reduced. See Fig. S-6.

The 90+ Durometer dynamic O-ring provides optimum properties for applications up to 105 bar (1500 psig). The O-ring is designed to resist extrusion. Its high density minimizes the possibility of explosive decompression. Additionally, because the O-ring is not confined radially, reverse pressure is easily handled. Back pressure is relieved by the O-ring and garter spring system, reducing the reverse pressure buildup that can cause other seals to fail.

Polytetrafluoroethylene sealing elements optional design: PTFE J-ring seal. The PTFE dynamic seal, or J-ring seal, extends the operating range to temperatures of –100

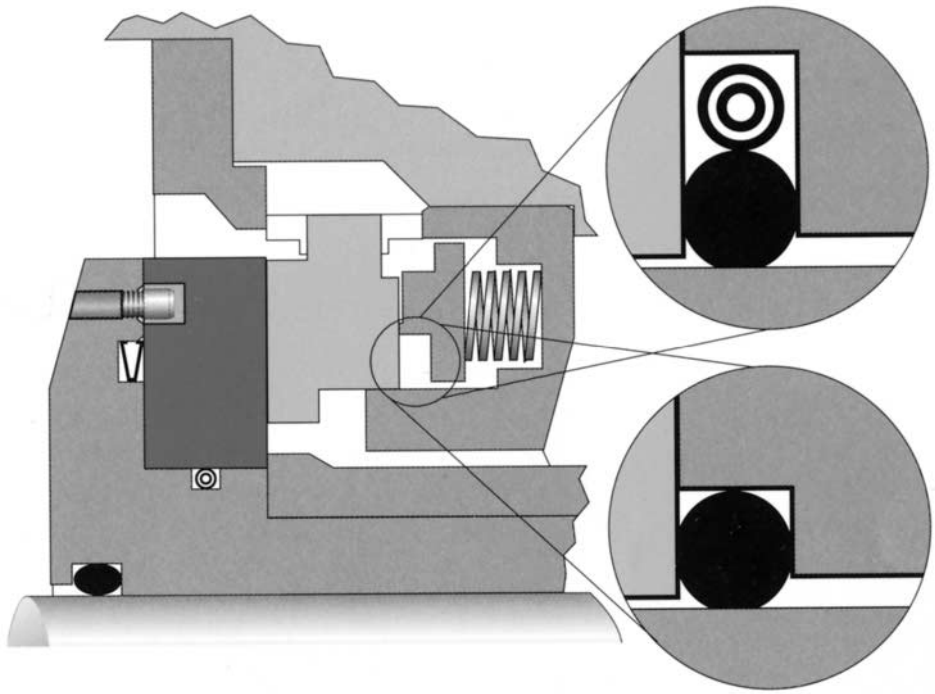


FIG. S-6 O-ring arrangement. (Source: Flowserve.)

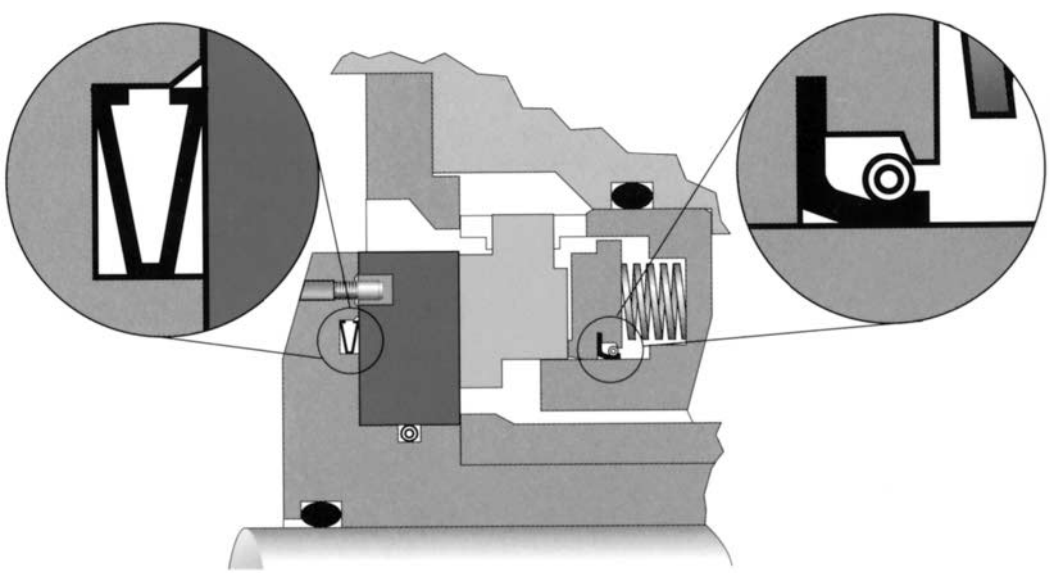


FIG. S-7 J-ring seal. (Source: Flowserve.)

to 230°C (–148 to 446°F) and pressures up to 230 bar (3300 psig). PTFE is also used for its extended chemical resistance beyond that of O-rings. The J-ring has a spring-energized lip that functions similarly to the spring-energized O-ring. The lip maintains contact with the sliding surface to provide a reliable seal. Pressure reversal is also handled effectively. The nonporous nature of PTFE has proven effective at minimizing explosive decompression. The combination of GASPAC's hard-versus-hard face configuration and J-ring design has allowed handling of the highest pressures across a single seal face. See Fig. S-7.

TABLE S-1 Technical Information and Operating Parameters

Pressure	0 to 230 bar, 0 to 3,300 psi
Speed	Up to 200 m/s, up to 650 ft/s
Temperatures	–100 to 230°C 148 to 446°F
Rotating face materials	Silicon carbide Silicon nitride
Stationary face materials	Special grade carbon Silicon carbide

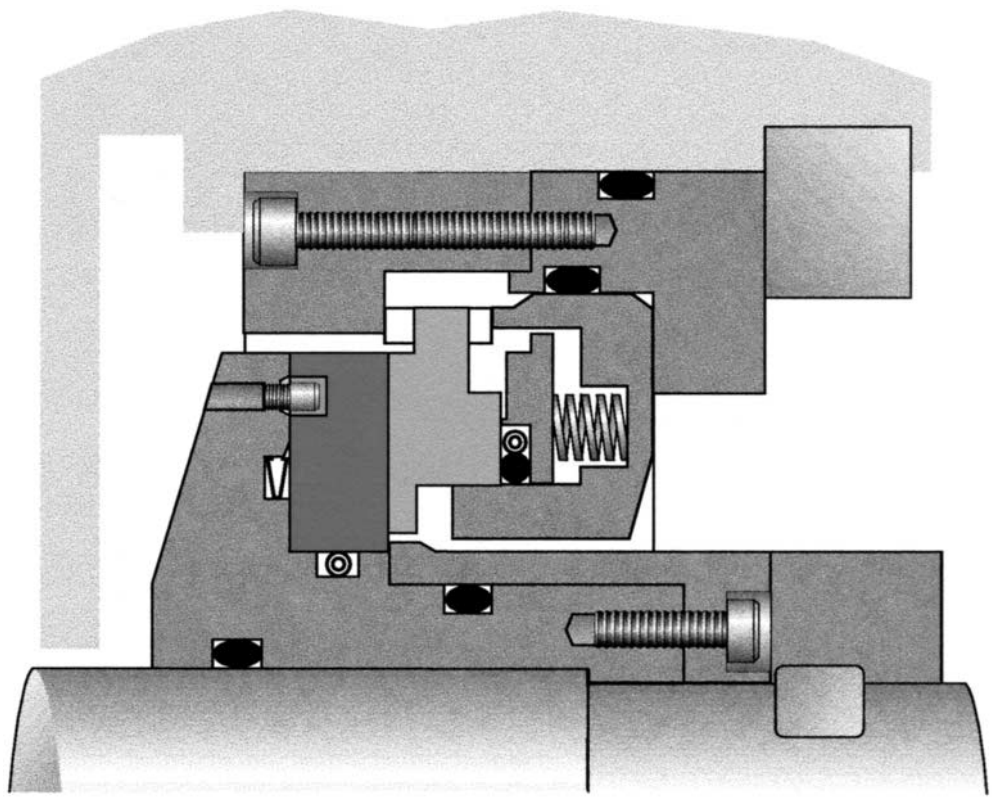


FIG. S-8 Single seal design. (Source: Flowserve.)

Available seal configurations. See Table S-1.

GASPAC S (single seal) 985. A single seal design is suitable for moderate-pressure applications where the gas sealed, such as air, nitrogen, or carbon dioxide, is neither flammable nor harmful to the environment. Seal leakage is to the atmosphere. A labyrinth seal may be integrated to reduce the amount of leakage in the event of a failure. See Fig. S-8.

GASPAC D (double seal) 987. The double opposed seal configuration requires a barrier gas pressure higher than the pressure of the process gas being sealed. This configuration can be used where any leakage of the process gas is not permissible, where consumption of filter buffer gas needs to be minimized, or in dirty gas application. See Fig. S-9.

GASPAC T (tandem seal) 986. The tandem seal provides full pressure breakdown across the primary seal faces. The secondary seal faces normally operate under low

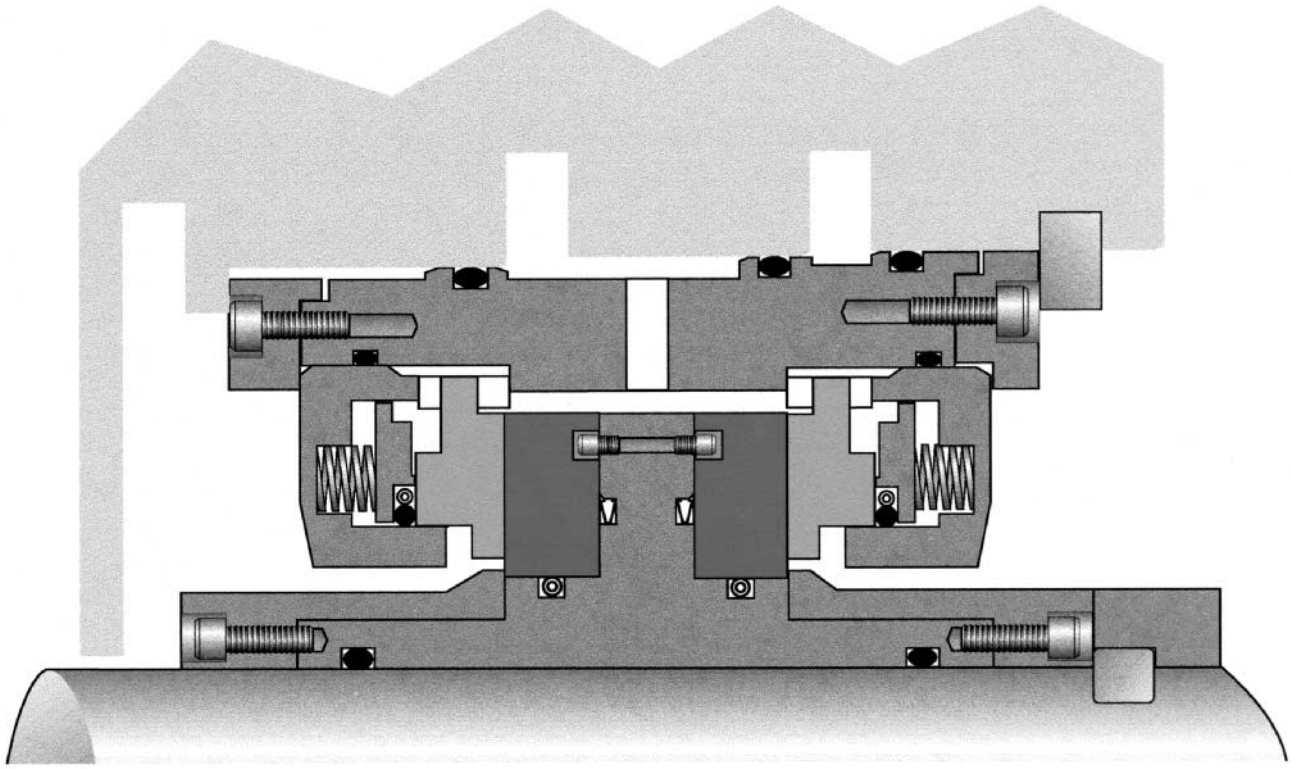


FIG. S-9 Double seal design. (Source: Flowserve.)

pressure. In the event of primary seal failure, the secondary seal acts as an installed spare. The process gas has controlled leakage across both sets of seal faces. The tandem seal has become an industry standard for hydrocarbon or critical applications. See Fig. S-10.

GASPAC L (tandem seal with interstage labyrinth) 984. The tandem seal with interstage labyrinth is used to eliminate process gas leakage to the atmosphere. This is accomplished by introduction of an inert buffer gas to the secondary seal. With a slightly higher inert buffer gas pressure, the labyrinth will keep the process gas from migrating to the secondary seal faces. The interstage labyrinth provides a low-pressure solution to controlling emissions across a gas seal. See Fig. S-11.

Barrier seals. The barrier seal is an integral component of successful compressor gas seal operation. Barrier seals are designed to keep oil within the bearing housing from contaminating the compressor gas seal.

Control panels. The panel distributes filtered gas to buffer the seal and energize the barrier seal. It also monitors gas seal performance.

Retrofit assistance. Converting from an oil seal to a compressor gas seal can pay for itself in improved reliability and lower operating costs. The logistical complexity of a proposed changeover can't be underestimated, however.

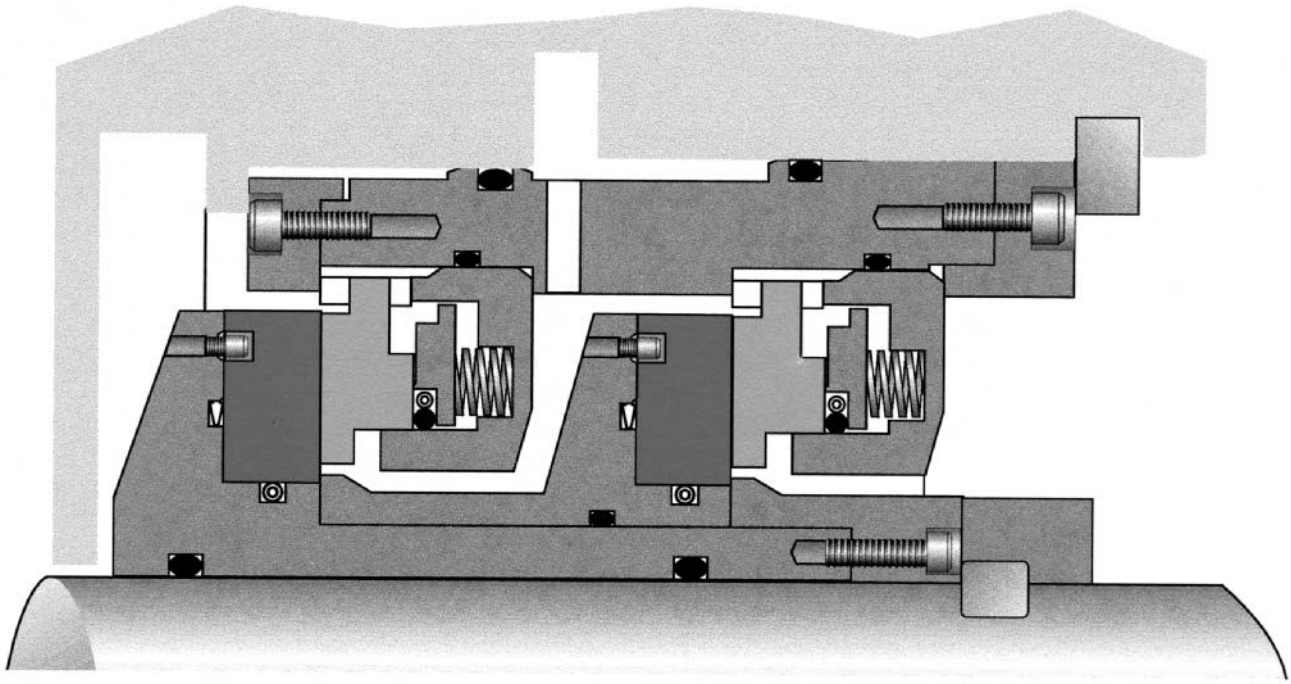


FIG. S-10 Tandem seal design. (Source: Flowserve.)

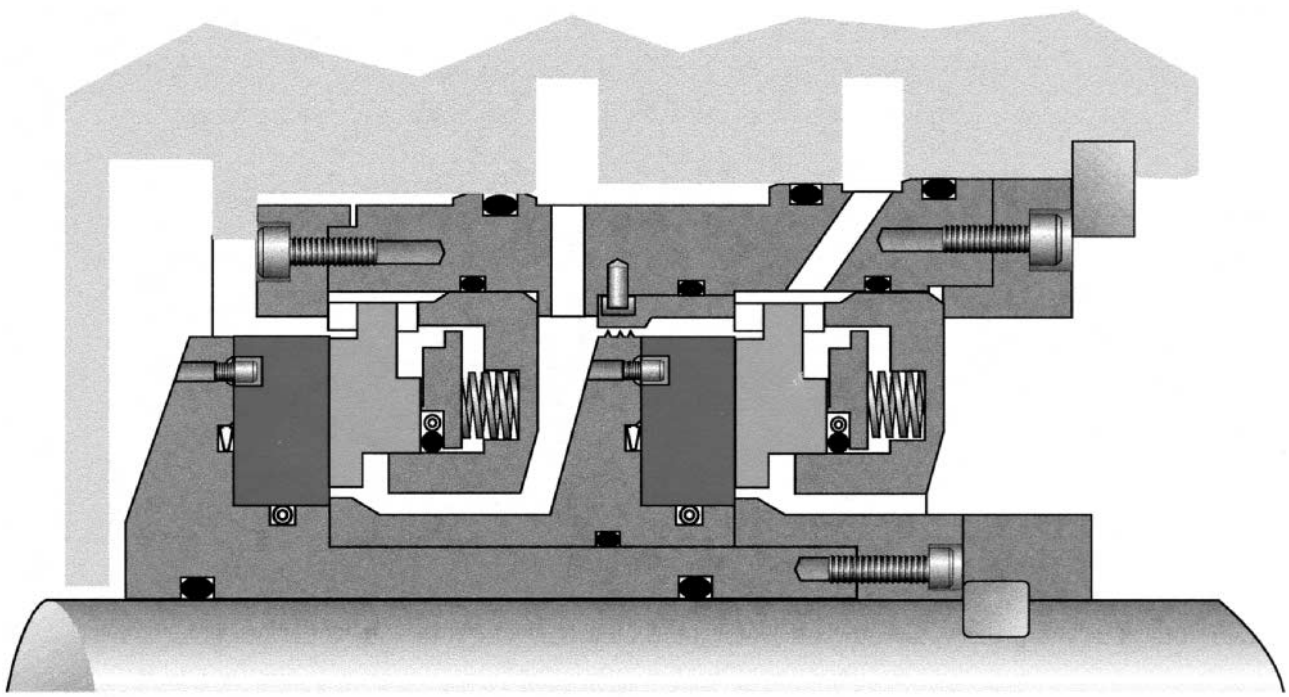


FIG. S-11 Tandem seal with interstage labyrinth. (Source: Flowserve.)

Separators*

Separators might also be called *knockout drums*. They are used to extract liquid droplets entrained in a gas stream and are generally positioned upstream of rotating machinery that either compresses the gas or uses it for fuel. Liquid droplets in gaseous fuel in a turbine designed for being fueled by gas only has resulted in ignition of the droplets and a totally incinerated turbine section. Less severe cases have resulted in localized "hot flashing" of the liquid and consequential heat damage to the intake nozzles. The penalty of not having separators in a steam line supplying a steam turbine has resulted in severe erosive damage to the turbine airfoils.

Sometimes the application involves gas and higher levels of liquid in the process stream and then the separator is required to prevent slugging. Some separators are designed for three-phase operation (gas, liquid, and solid). Different designs are discussed here.

Dry Scrubbers

The dry scrubber is designed to remove liquids and dry impurities at the wellhead, inlet to transmission stations, in distribution systems and in many industrial processes. The use of a dry scrubber has the following advantages:

1. Efficient removal of dust and liquids
2. Cannot be overloaded
3. Uses no scrubbing oil, hence no oil loss
4. Requires no oil filling or settling tanks
5. Virtually maintenance free, requiring only occasional blowdown of collected material
6. Ordinarily smaller and less costly than wet-type scrubbers
7. Efficiently handles wide flow range to allow for future conditions
8. Constant pressure drop regardless of contaminant loading

Principle of operation

The use of centrifugal force to separate solid and liquid particles from a gas is well known. The low pressure, large diameter cyclone separator on the roof of industrial plants to remove scale, sawdust, grain chaff, etc., is a familiar sight.

Due to the unique requirements of the gas industry, which has high pressures, wide flow ranges and requires high efficiency, multiple small diameter cyclones are arranged in parallel to achieve results on small as well as large particles. The cyclone tube is approximately 2 inches in diameter and is described in Fig. S-12 while Fig. S-13 shows the multiple bank of cyclones.

Dirty gas enters the cyclone tube at two points, designated A, and sets up a swirling motion. Solid and liquid particles are thrown outwardly and drop from the tube at point B to a large storage area. The swirling gas reverses direction at the vortex C and rises through the exit portion of the tube, designated D. This tube design results in the most efficient particle removal available today for high pressure gas scrubbing by means of centrifugal force.

---

* Source: Peerless, USA. Adapted with permission.

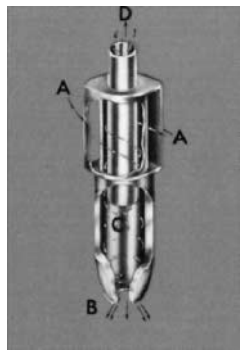


FIG. S-12 Dry scrubber operation. See text for key to components. (Source: Peerless.)

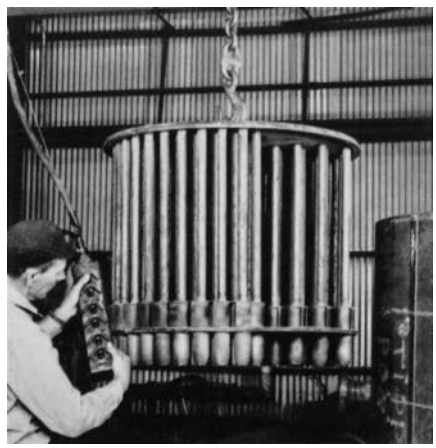


FIG. S-13 Multiple bowl of cyclones. (Source: Peerless.)

The dry scrubber has the following efficiency on pipeline dust:

8 micron size particles and above: 100 percent

6 to 8 micron size particles: 99 percent

4 to 6 micron size particles: 90 percent

2 to 4 micron size particles: 85 percent

The above efficiency is based on a velocity of 20 ACFM per tube. For higher flows, the efficiency will be slightly higher and for lower flows it will be slightly lower.

Liquid removal: In a typical specification the outlet gas will not contain more than 0.10 gallon of entrained liquid per million standard cubic feet of gas passing through the separator.

Design features

The dry scrubber is designed to give high capacity with high efficiency and low pressure drop.

Capacity. Any desired capacity is afforded by providing a sufficient number of tubes in parallel in a suitably sized vessel.

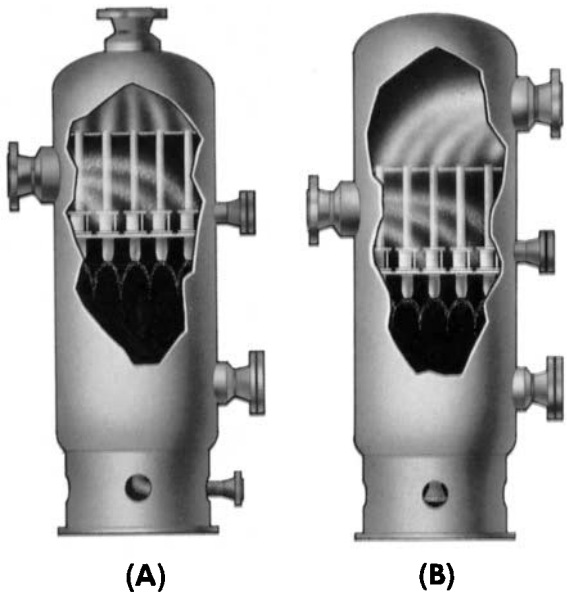


FIG. S-14 Two vertical dry scrubber designs. (Source: Peerless.)

A maximum capacity of 40 ACFM per tube and a minimum of 10 ACFM are recommended. The maximum is recommended to prevent excessive wear of the tubes and the minimum is maintained to provide sufficient centrifugal force to accomplish satisfactory particle removal efficiency. The maximum can be exceeded and greater efficiency is obtained. This practice is recommended in specific applications only.

Pressure loss. Pressure drop is an important consideration in almost any scrubber installation. The dry scrubbers can be designed for pressure drops varying from several inches of H_2O to several psi. Since a minimum actual cubic foot flow/cyclone tube must be maintained to effect good separation, the pressure drop will vary directly as the operating pressure.

Cyclone tube construction. In addition to high efficiency and capacity, an ideal cyclone must be constructed to withstand many years of abrasive wear and be rugged enough to withstand the pounding inherent in compressor station operation. The critical wearing parts of the cyclone tubes are made of a special wear-resistant steel alloyed with nickel and chromium. This material combines abrasive resistance with toughness or shock-resistant qualities.

The cyclone tube construction is more expensive and better suited to the application than its more brittle and less wear-resistant counterparts in other makes of scrubbers.

Optional features

Configuration. Dry scrubbers can be furnished in almost any configuration that the customer desires. Where space is limited, a vertical design with side in and top out, side in and side out, or other piping configuration can be provided (Fig. S-14). In the case of very large volumes such as in gas transmission stations, the horizontal unit might be most advantageous. The horizontal dry scrubber also offers another

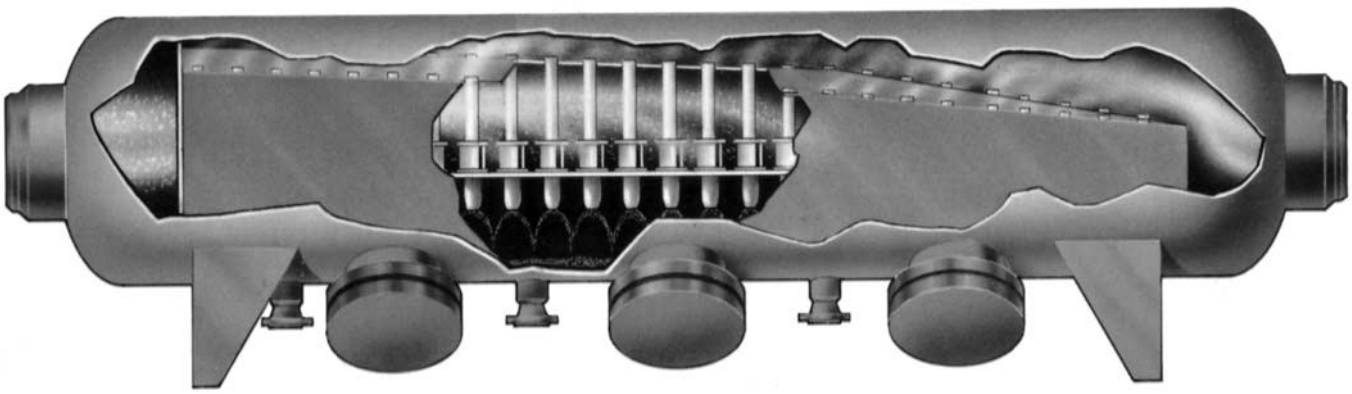


FIG. S-15 Horizontal dry scrubber. (Source: Peerless.)

advantage in helping alleviate inherent vibration problems that might be encountered in vertical piping (Fig. S-15).

Removable tube section. The dry scrubber with a removable tube section that might be necessary in certain applications. This feature has been provided in sizes up to 48 inches in diameter.

Dust storage tanks can also be furnished.

Construction. Dry scrubbers are constructed of carbon steel in accordance with the ASME Code; however, they can be built of almost any weldable material.

The separator/filter

See Fig. S-16.

1. *Inlet chamber.* The abrupt decrease in gas stream velocity inside the inlet chamber drops out entrained material by gravity. At the same time, the "square" tube arrangement (as opposed to the conventional triangular arrangement) allows a free fall of separated material to the bottom of the chamber. This material is forced by gas velocity into the inlet chamber sump, from which it is subsequently removed. This design extends filter cartridge life by reducing the loading of material on the cartridges.

2. *Filter chamber.* The filter cartridges in the filter chamber have two functions—first, to coalesce fine liquid droplets into larger ones that are subsequently separated in the mist extractor chamber, and second, to filter out all remaining solid particles that have not been removed in the inlet chamber.

As with all filters, periodic replacement of the filter cartridges is necessary because of the accumulation of solids. However, in the design, replacement is required less frequently, since large quantities of contaminant are removed in the inlet chamber.

In addition, the arrangement and design of the filter cartridges in a filter/separator make filter replacement a quick and simple operation.

All filter/separators allow the use of a small closure for filter cartridge access. Removal and replacement of the filter cartridges is accomplished through the use of a filter retention mechanism that is constructed of large, easily handled parts. The retention mechanism completely eliminates internal filter supports, small threaded studs, multiple nuts and washers, and complicated lattice works.

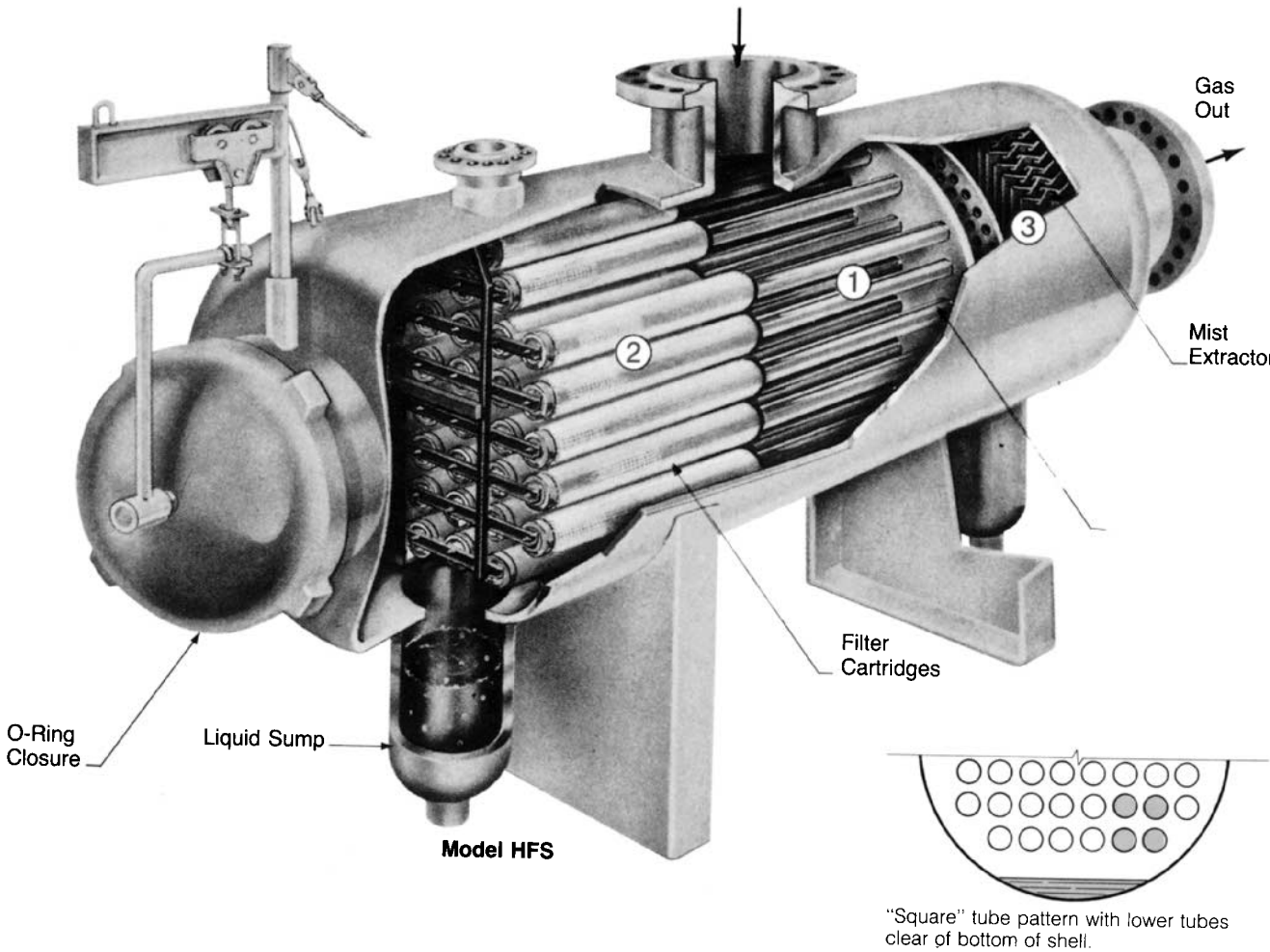


FIG. S-16 Filter/separator. (Source: Peerless.)

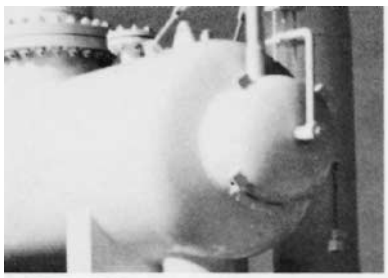
Upon removal of the filter cartridges, the inlet chamber is completely exposed for easy cleaning and removal of built-up solid materials.

See Figs. S-17 through S-19. Figure S-20 shows other configurations of filters/separators.

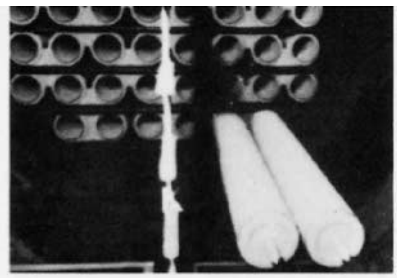
3. *Mist extractor chamber.* This chamber utilizes the vane-type mist extractor to remove with a high degree of efficiency coalesced liquid droplets that have been carried over from the filter chamber. High-capacity vanes are mounted perpendicular to the gas flow, reducing overall vessel length.

Filter/separator advantages

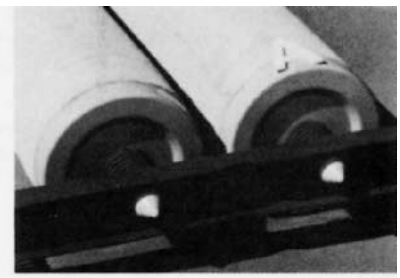
- Quick and simple maintenance
- Small closures
- Maintenance from outside the vessel—not necessary for operator to enter
- Extended filter cartridge life
- No small parts
- No threaded studs



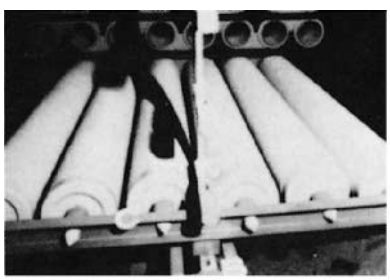
A. Small closure opens quickly.



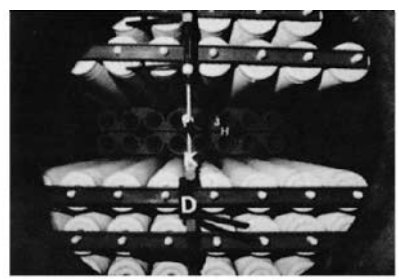
B. Filter cartridges are placed on temporary shelf and rolled into place.



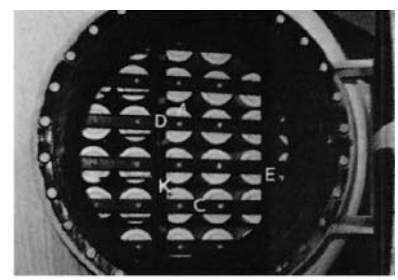
C. Support bars with attached pressure springs are fitted over bullet noses of cartridges.



D. Lower tier of filter cartridges is completed.



E. Additional tiers of filter cartridges are completed.



F. Vertical support bars are in place and large wing nut (shell diameters 42 in and larger) is tightened.

FIG. S-17 Filter cartridge installation system. (Source: Peerless.)

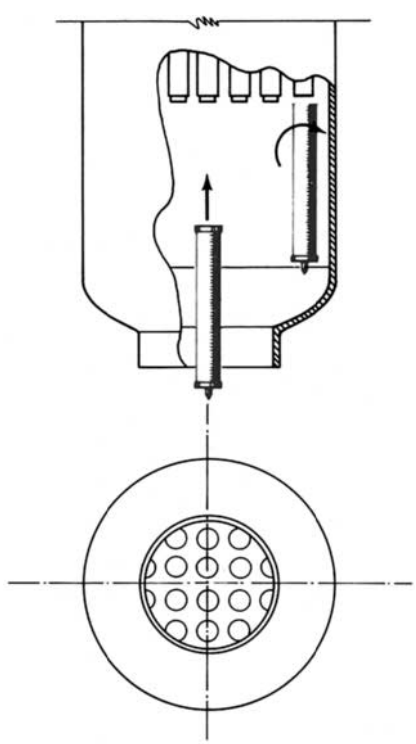


FIG. S-18 Filter cartridges inserted through closure and rolled into position. (Source: Peerless.)

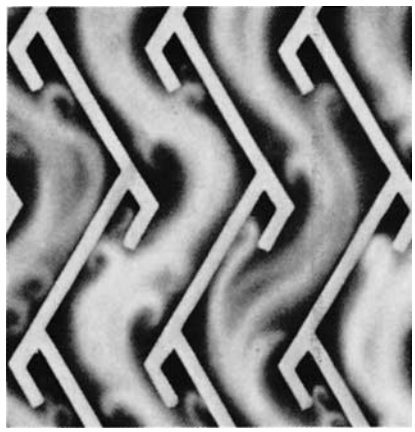


FIG. S-19 Mist extractor vanes. (Source: Peerless.)

- 0 to 100 percent flow range with a constant separation efficiency
- Mist extractor not susceptible to plugging
- Access to filter gasket seats
- Easy cleanout during maintenance
- Collapsed filter cartridges easily removed
- Low maintenance turnaround time
- Compact vessel size
- No filter lattice supports

Vertical absolute separators

Application. The absolute separator was developed for applications where the maximum possible separation of minute liquid particles—varying from mist to submicron in size—from a gas stream is required. The principal applications of the absolute separator are for removal of oil mist upstream of dessicant beds in ammonia and urea plants and other chemical and petrochemical plants. Absolute separators are also effective for removal of mist in process streams such as chlorine, nitrogen, hydrogen, oxygen, carbon dioxide, and air purification.

Operation. The absolute separator is a multistage device utilizing all of the forces employed in mechanical separation—gravity, centrifugal force, inertial impaction, direct interception, and Brownian diffusion.

It consists of a (1) primary section to remove entrained liquids and foreign contaminants ahead of the (2) final, high-efficiency separation section that coalesces and separates the remainder of the liquid. By removing the bulk of the entrained liquid in the primary section, the design increases the life of the final separation elements and holds the pressure drop buildup to a minimum.

Primary separator section. The inlet section has been designed to utilize the forces of gravity and centrifugal force through the use of small diameter cyclones to remove liquid and solid particles. In some applications the vane-type mist extractor will be used.

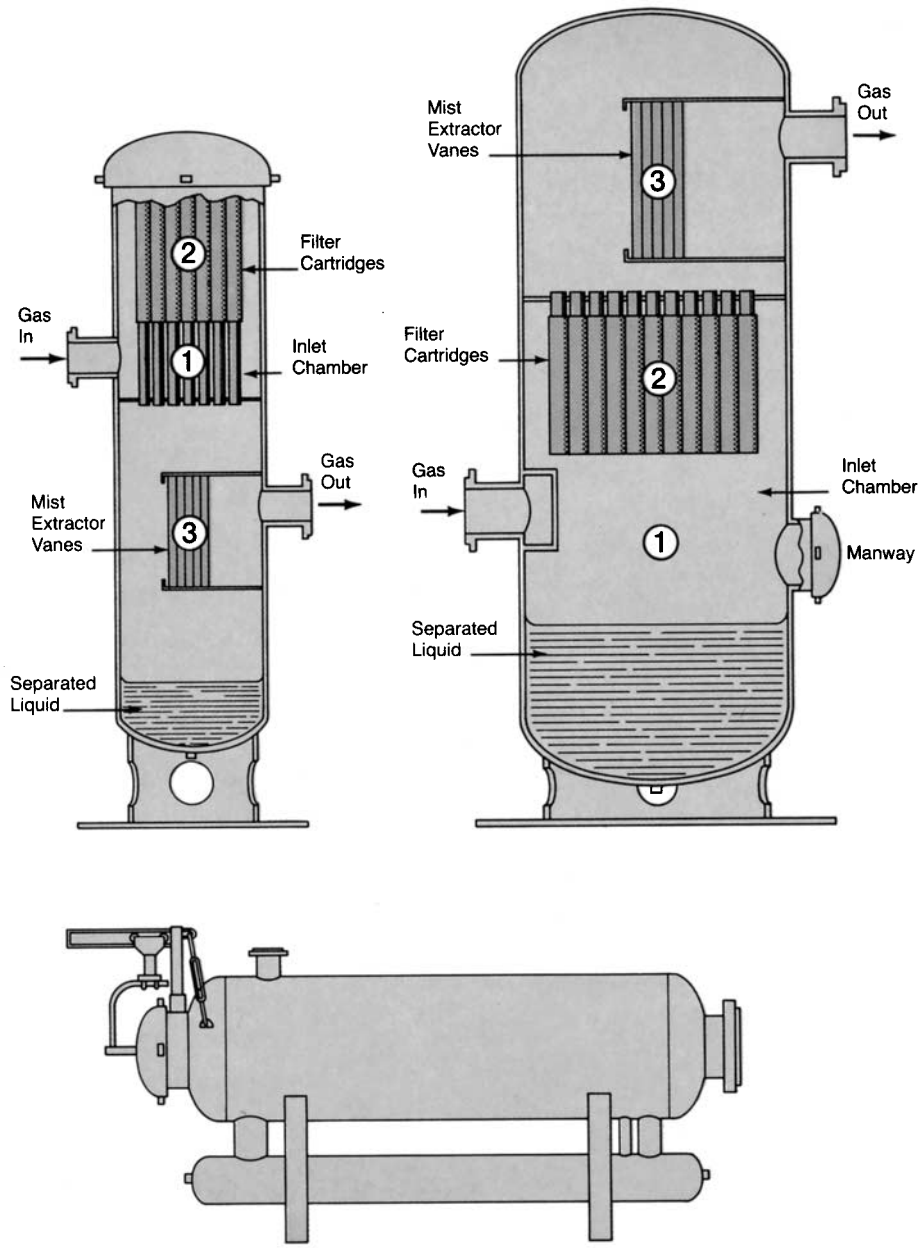


FIG. S-20 Horizontal with horizontal lower barrel and vertical configuration. (Source: Peerless.)

Cyclone inlet section. As the mist-laden gas enters the separator, the entrained liquids and solid particles are subjected to centrifugal force. The gas enters the cyclone tube at two points, designated A, and sets up a swirling motion. Solid and liquid particles are thrown outwardly and drop from the tube at point B. The swirling gas reverses direction at the vortex C and rises through the exit portion of the tube, designated D. See Fig. S-21.

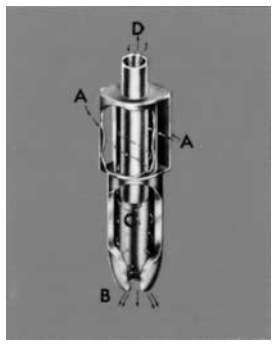


FIG. S-21 Cyclone operation filter/separator. See text for key to components. (Source: Peerless.)

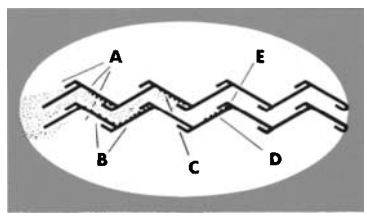


FIG. S-22 Mist extractor section. (Source: Peerless.)

Mist extractor inlet section (alternate for some applications). As the gas enters the vane unit, it is divided into many vertical ribbons (A). Each ribbon of gas is subjected to multiple changes of direction (B) as it follows its path through the vanes. This causes a semiturbulence and rolling of the gas against the walls of the vanes (C). The entrained droplets are forced to contact the vane walls where they impinge and adhere to the vane surface (D). This liquid then moves into the vane pockets (E) and out of the gas stream. It is then drained by gravity into the liquid reservoir. The collected liquid can then be disposed of as desired. See Fig. S-22.

Final separation element. The final separation section consists of one or more cylindrical coalescing elements mounted vertically on support tubes. The gas and fine mists pass from the inside to the outside of the elements. In passing through the coalescing elements, the entrained mist particles diffuse and impinge on the closely spaced surfaces of the element and are agglomerated into larger liquid droplets. The larger liquid droplets emerge on the outer surface of the coalescing element and run down the sides of the element to the liquid collection chamber. The gas, free of liquid particle entrainment, rises and passes out of the separator through the upper gas outlet nozzle.

Design features. Replacement of the final separation elements can be made with a minimum of time and effort through the use of a full opening O-ring or float ring closure. The primary separation elements (vane-type mist extractor or cyclone section) are completely maintenance-free and self-cleaning, with no replacement or moving parts to cause shutdown.

The absolute separator could be guaranteed to remove 100 percent of all liquid particles above 3 microns; and, depending on design conditions, it will remove up to 99.98 percent of all particles less than 3 microns. This efficiency is maintained throughout the entire flow range to design capacity.

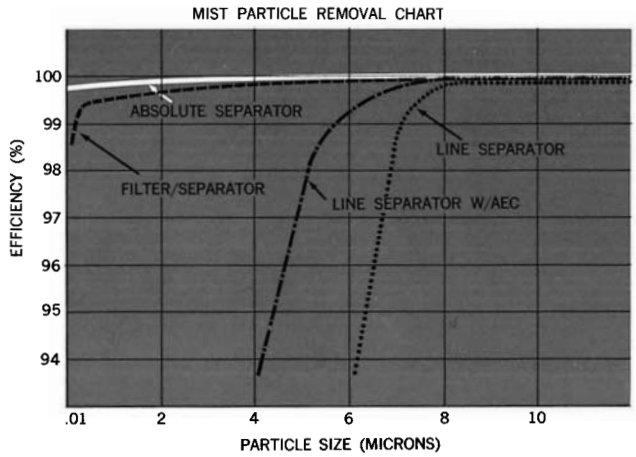


FIG. S-23 Mist particle removal chart. (Source: Peerless.)

TABLE S-2 Vane-Type Separators

	Body Diameter (O.D.)	6⅝″	8⅝″	12¾″	14″	16″	18″	20″
	In and out line sizes	2″	3″	4″	6″	8″	10″	12″
A	Face-to-face	17″	20″	24″	26″	30″	34″	38″
B	Approximate overall height	36″	38″	47″	49½″	52½″	60″	63″
C	Body length, seam-to-seam	29″	30″	36″	38″	40″	46″	48″
D	Top head seam centerline inlet and outlet	4½″	4½″	8½″	9″	9″	11″	11″
G	Lower head seam to bottom of base	12″	12″	18″	18″	18″	18″	18″
-0-	Liquid capacity (gal.)	2¾	3¾	8¼	11 1/16	14⅞	21	21⅞
-0-	Pressure drop (in, H_2O)	12–13	12–13	8–10	8	6–8	6	5

NOTES:
1. All dimensions are ±¼″.
2. Base support is optional.
3. Vessels stocked with design pressure of 275 psig 150# RF and 720 psig 300# RF.
4. Vessels equipped with these accessory fittings:
   A. 2, ¾″ 6,000# gauge glass connections.
   B. 2″ 3,000# equalizer connection.
   C. 2″ 3,000# drain connection (1½″ on 6⅝″ and 8⅝″ O.D. sizes).
   D. 2, 1″ 3,000# high-level shut-down connections on 14″ O.D. and larger sizes.
   E. 2″ 3,000# vent connection (1½″ on 6⅝″ and 8⅝″ O.D. sizes).

The normal pressure drop through the final separation elements is limited by design to 5 in of water column or less. The pressure drop across the primary section will depend on operating conditions and the type of separation elements used.

See Figs. S-23 and S-24.

Vane-type separator

Line separators are designed and fabricated to conform fully to all current ASME requirements and are usually furnished in carbon steel for most industrial applications; however, units can be fabricated in part or entirely from stainless steel, Monel, or other special alloy materials. See Figs. S-25 and S-26. See also Table S-2.

Although standard units are designed for 275- and 720-psi working pressure (412.5- and 1080-lb test pressure), vessels can be furnished in virtually unlimited

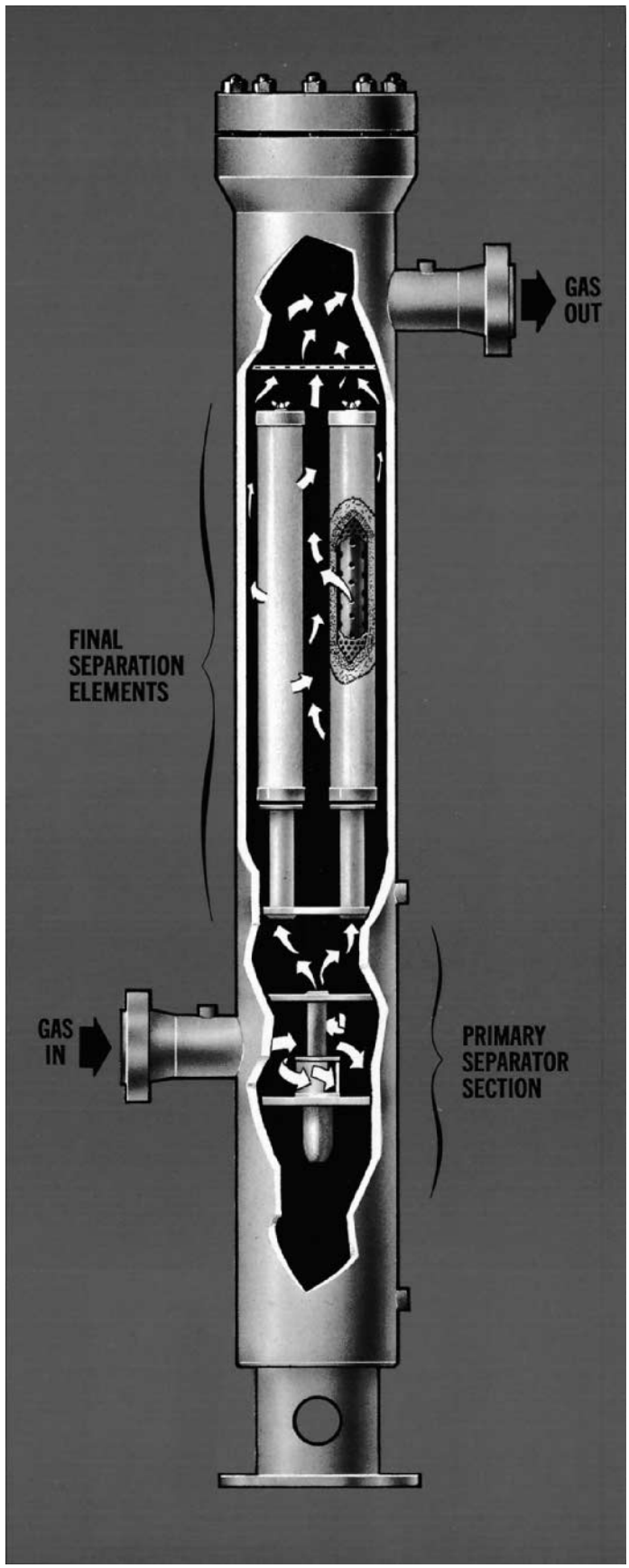


FIG. S-24 Absolute separator. (Source: Peerless.)

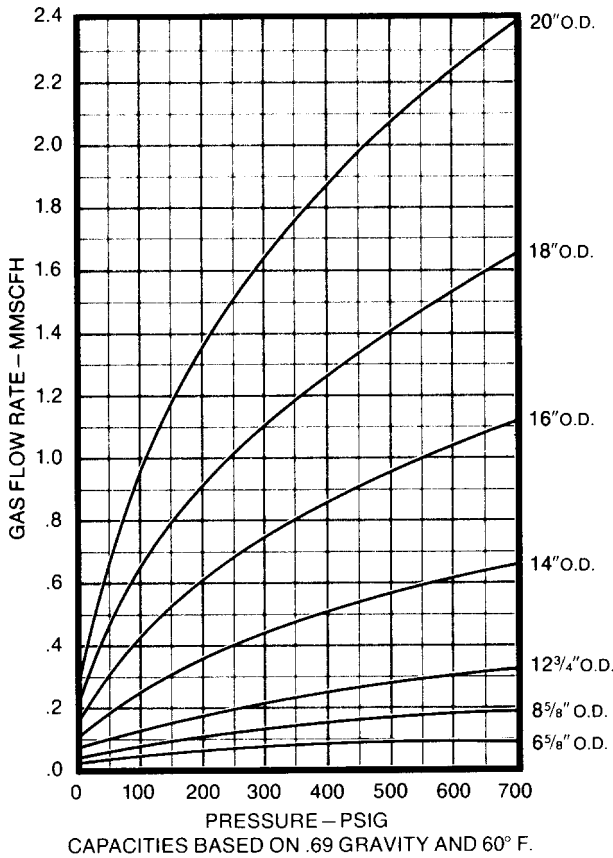


FIG. S-25 Vane-type separator performance curves. (Source: Peerless.)

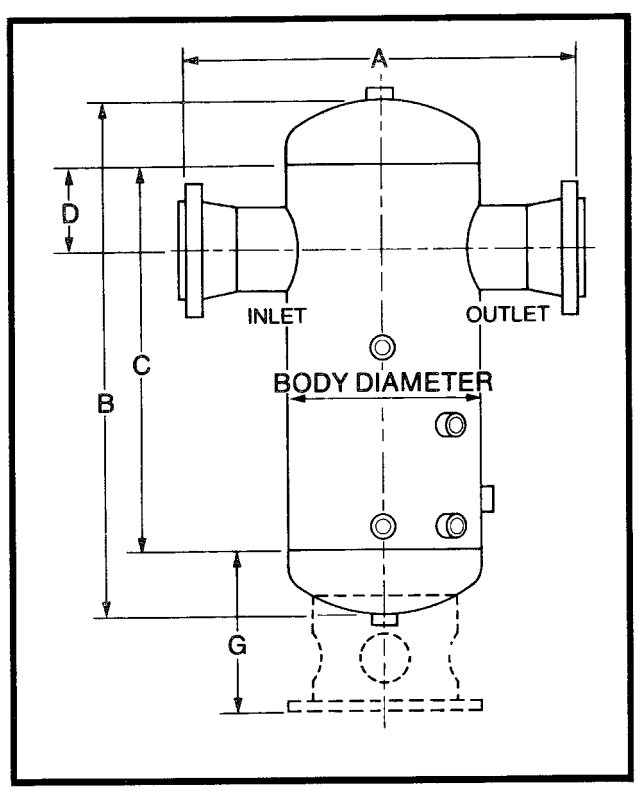


FIG. S-26 Vane-type separator (external). (Source: Peerless.)

ratings for greater pressures if required (pressures in the 20,000-lb range are not uncommon).

Vertical gas separators

These vessels employ many physical means to separate the liquids from gases in addition to the mist extractor. Foremost among these various separation forces are: impingement, centrifugal force, gravitational force, and surface tension. See Fig. S-27.

Inlet baffle. Of prime importance to the separation is the inlet impingement baffle, which acts to eliminate heavy slugging problems set up by excess amounts of liquid in the stream. See Fig. S-28. As the slugs of liquid come into contact with the baffle, they are deflected at an angle and are broken up by a hooked vane attached to the edge of the baffle. This breaking up of the slugs causes them to drop out of the stream and to the bottom of the separator. The baffle is made of extra thick material to protect against excess erosive wear.

Rise to mist extractor. Having removed a majority of the entrained liquid or slugs, the gas flow continues its travel to the mist extractor that is above the inlet baffle. During this travel, a centrifugal and gravitational action takes place that separates more of the entrained liquid. The distance the vapor (gas and liquid) must rise to enter the mist extractor aids in the separation by supplying time necessary to permit coalescing or the forming of small droplets into larger drops that have a greater rate of fall than the upward velocity of the gas. By this method, maximum separation, using impingement, centrifugal motion, and gravity, has been obtained with a minimum pressure drop.

This settling effect, utilized in the vertical gas separator, removes all but a very small portion of the liquid. This remaining liquid continues to rise toward the gas outlet in the form of a fine spray. To solve this final separation problem, the mist extractor is used.

Mist extractor. The mist extractor combines maximum scrubbing area with an absolute minimum pressure drop. It utilizes the forces of impingement, centrifugal motion, and surface tension to obtain its high efficiency. See Fig. S-29.

The path of the gas through the unit is constantly bending, causing the impingement of the liquid droplets against the walls of the vane, separating some of the entrained mist. Centrifugal force aids by throwing the heavier liquid droplets out of the main gas stream and impinging them on the scrubbing surface. The entrained liquid, after coming into contact with the metal surface and other liquid droplets of the vane unit, is coalesced and adheres to the vane surface by utilizing the forces of surface tension. Gravity and impact of the gas stream then drives the droplets into the pockets provided at each turn of the vane where they roll down out of the gas stream. After going through the complete process of scrubbing and separation, the gas finally reaches the outlet opening of the separator clean and dry.

Liquid control. The large liquid reservoir is adequate to store incoming slugs of liquid during the time required for opening the liquid valve. The volume of the vessel is large enough to allow the gas to break out of the solution and to escape the liquid in the bottom of the separator.

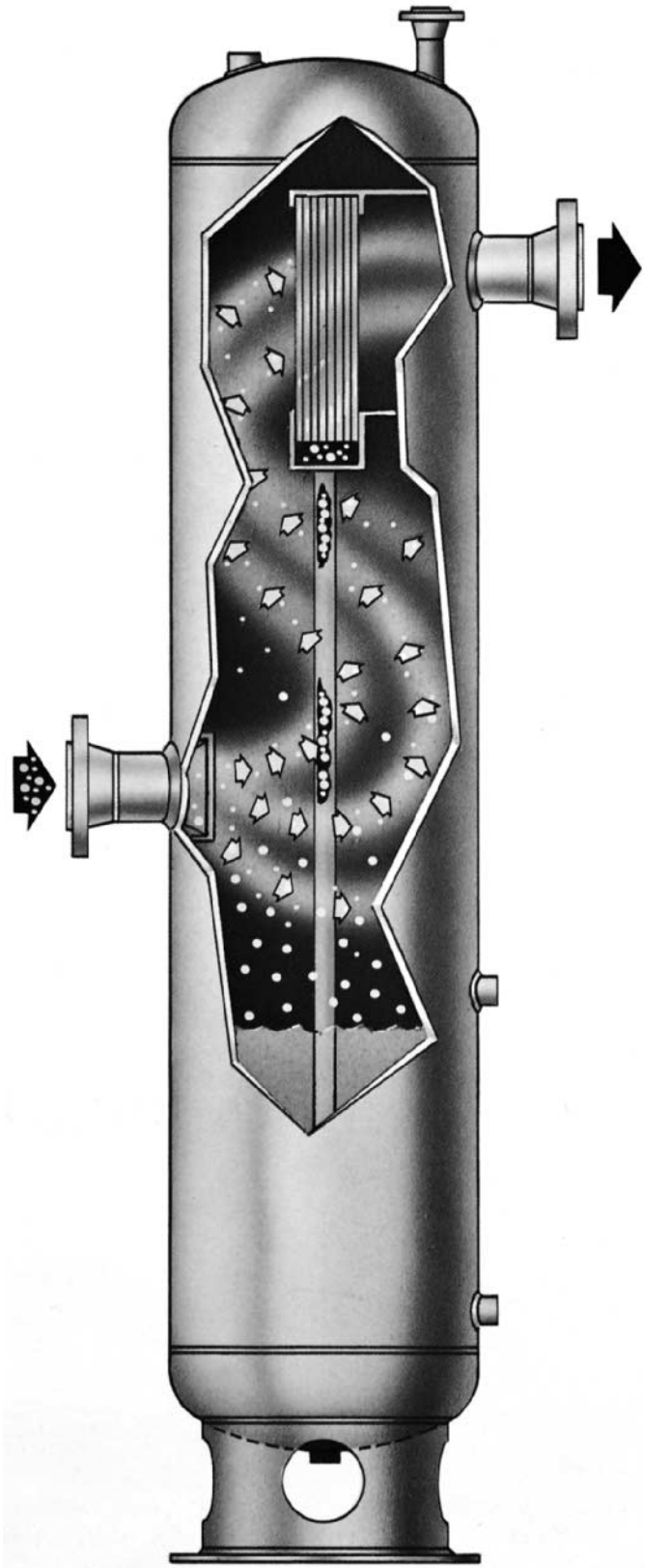


FIG. S-27 Vertical gas separator. (Source: Peerless.)

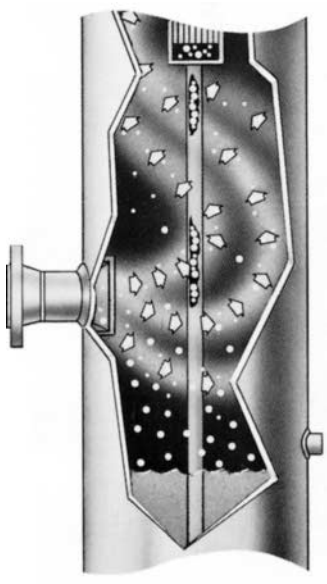


FIG. S-28 Interbuffer—vertical gas separator. (Source: Peerless.)

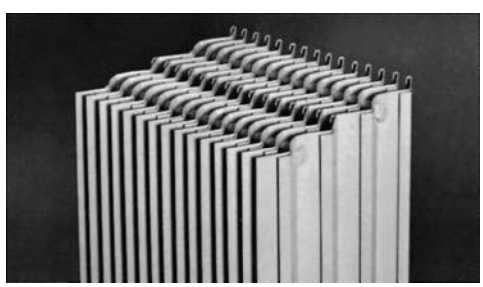


FIG. S-29 Mist extractor. (Source: Peerless.)

Line separators

The vane-type line separator offers efficient separation of entrained liquids from a gas or vapor stream. This separator design has been used successfully for over 25 years in chemical plants, refineries, natural gas pipelines, and all types of industrial processing plants where efficient liquid-gas separation has been required. See Figs. S-30 and S-31.

These separators incorporate the vane-type mist extractor as the separating element. This unit offers a number of operating characteristics not found in other types of separators:

- 100 percent removal of all entrained droplets 8–10 microns and larger
- Extremely low pressure drop—less than 6 in of water column
- Small housing requirement for ease of installation and economy
- Flat efficiency curve with no decrease in efficiency from rated capacity down to zero flow

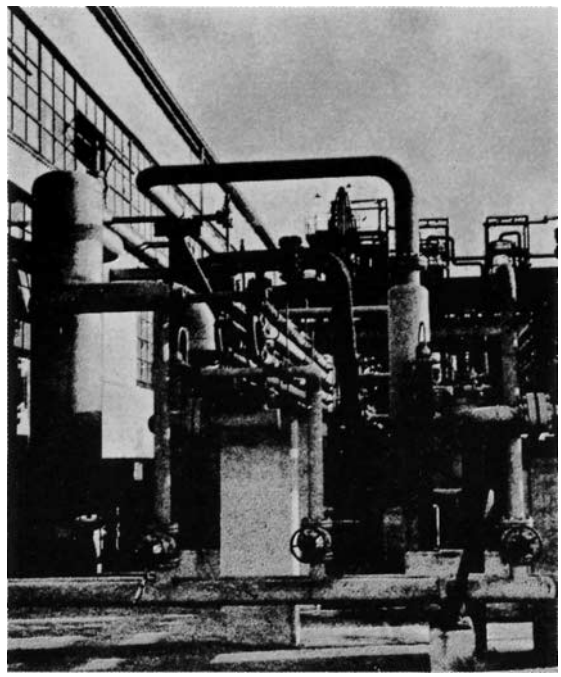


FIG. S-30 Line separator installation. (Source: Peerless.)

Principle of operation. The vane unit is the heart of the separator (see Fig. S-32). As the gas enters the vane unit, it is divided into many vertical ribbons (A). Each ribbon of gas is subjected to multiple changes of direction (B) as it follows its path through the vanes. This causes a semiturbulence and rolling of the gas against the walls of the vanes (C). The entrained droplets are forced to contact the vane walls where they impinge and adhere to the vane surface (D). This liquid then moves into the vane pockets (E) and out of the gas stream where it is drained by gravity into the liquid reservoir. The collected liquid can then be disposed of as desired.

It is significant to note that the liquid drainage in the vane-type mist extractor differs from the drainage in other impingement-type mist extractors, in that vane drainage occurs with the liquid out of the gas flow and at a right angle to the direction of flow through the separator.

The individual vane corrugations, depth and size of the liquid pockets, and the vane spacing are critical features of the vane-type mist extractor. Many years of testing and operating experience eventually arrive at optimum dimensions and spacing. The slightest variation in any one of these three features will materially decrease the capacity and performance of this type of separator.

Efficiency and capacities. The vane-type line separator (see Fig. S-33) will remove all of the entrained liquid droplets that are 8–10 microns and larger. The efficiency of the unit decreases on droplet sizes less than 8 microns as shown on the chart. In order to separate these smaller droplets, the separator must be preceded by an agglomerating or coalescing device to increase the size of the droplets so that they can be removed by the mist extractor. Several types of agglomerating devices are available. Some of these are capable of achieving efficiencies as high as $99^1/_2$ percent removal of 1 micron size droplets.

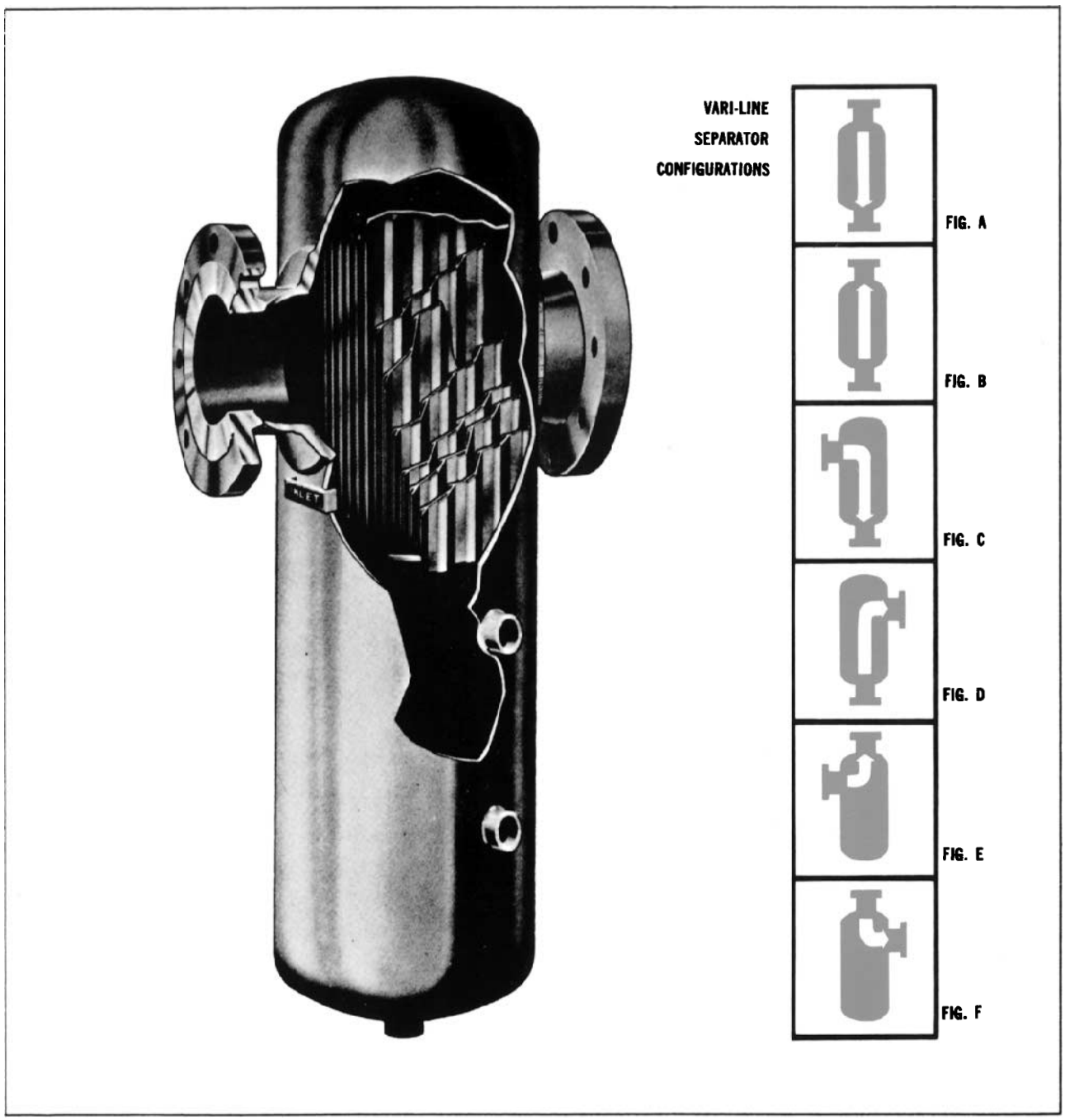


FIG. S-31 Line separator. (Source: Peerless.)

Low-pressure drop. Since the vane-type mist extractor is self-cleaning and contains no small openings that can fill up and restrict the flow—such as are present in wire mesh pads or filter screens—the pressure drop across the separator is very low. The drop is as small as 2–3 in of water in the larger sizes.

Vari-line separators. Vari-line separators are specifically designed for those installations where space is at a premium and piping limitations prevent the use

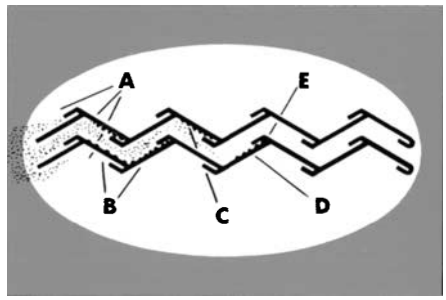


FIG. S-32 Section of vane unit on line separator. (Source: Peerless.)

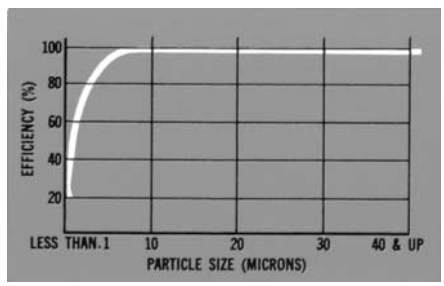


FIG. S-33 Line separator performance curve. (Source: Peerless.)

of a straight pattern line separator, which has the gas inlet and outlet connections on the same horizontal centerline. These units consist of the vane-type mist extractor with internal baffling designed to permit virtually any combination of locations for the gas inlet and gas outlet connections. The principle of operation and performance characteristics for the vari-line separators are the same as those described for the straight pattern line separator.

Typical inlet and outlet connections are: side in–top out, side in–bottom out, top in–side out, top in–bottom out, bottom in–side out, and bottom in–top out.

High-pressure separators. High-pressure separators are designed for pressures in excess of 1500 lb. There is no upper limit on design pressures. Several separators have been fabricated with design pressures in excess of 20,000 lb. These separators contain the vane-type mist extractor and the internal design and principle of operation is the same as discussed for the lower pressure units.

The pressure vessel housing for these separators can be fabricated using either A105 Grade 2 forged steel or A216 WCB cast steel. Virtually all of the separators now used in plant designs are fabricated of forged material.

These separators have application in any service where high-efficiency separation of entrained liquids from a gas is required.

Horizontal gas separators

Principle of operation. The gas is directed into the inlet separator section (A) (see Figs. S-34 and S-35) where most of the liquid is removed. This separated liquid drains into the first downcomer (B). The remaining liquid is then scrubbed by the

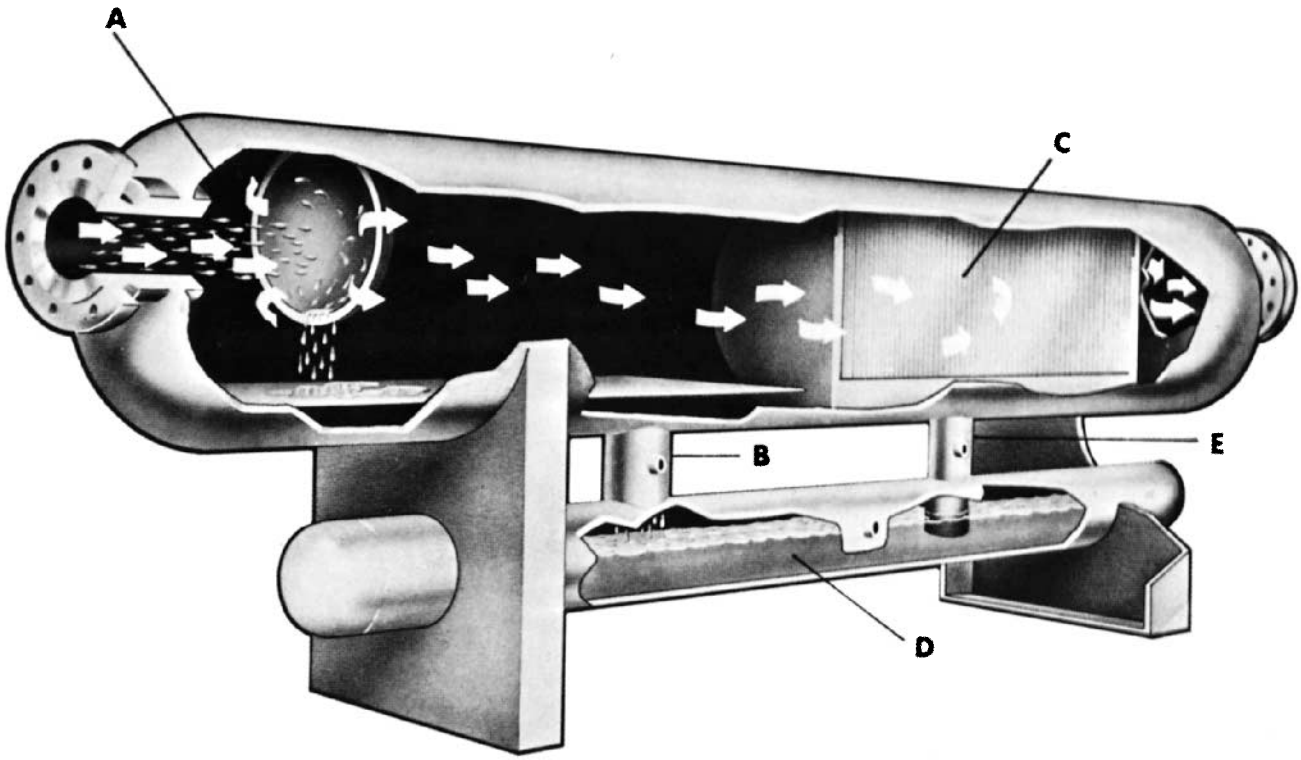


FIG. S-34 Horizontal gas separator. (Source: Peerless.)

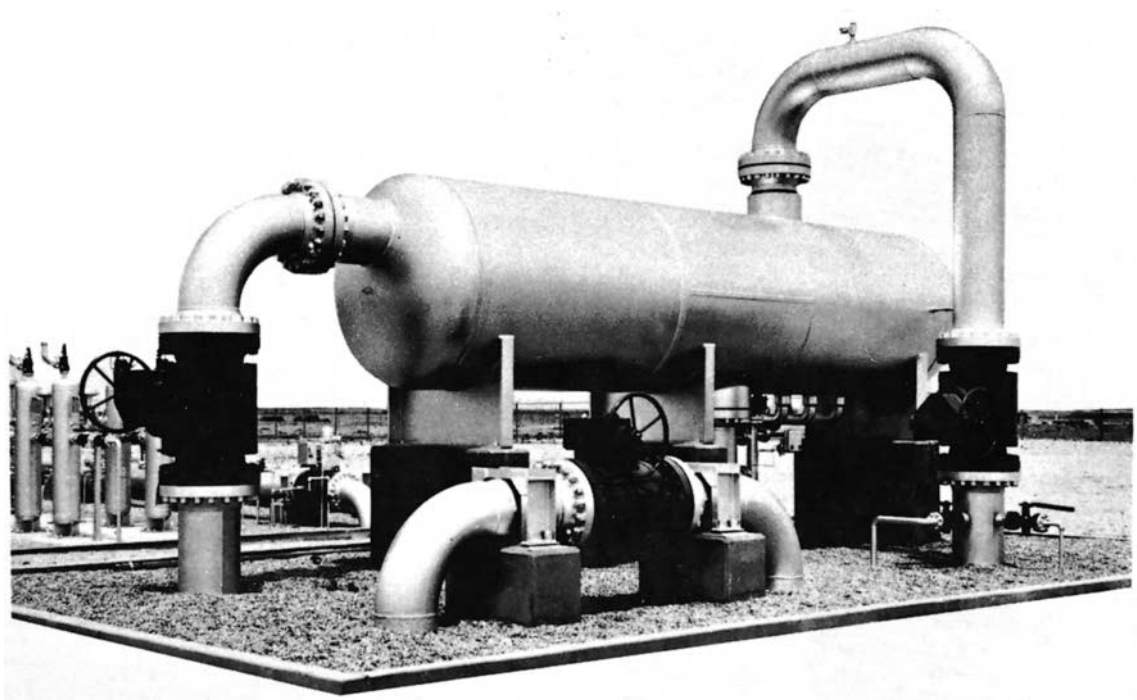


FIG. S-35 Horizontal single barrel design. (Source: Peerless.)

mist extractor (C). This entrainment drains into the lower barrel (D) through downcomer (E). This second downcomer is submerged in liquid, and this liquid seal prevents the gas from following a path through the lower barrel and bypassing the mist extractor.

Advantages of the lower barrel. The lower barrel makes it possible to get the separated liquid away from the gas flowing in the upper barrel, thus eliminating reentrainment. It also makes possible the installation of larger separation elements in the upper barrel, which results in a higher capacity. The lower barrel also provides a quieting chamber for gas to break out of solution to effect a clean separation.

Ordinarily, liquid level in the lower barrel is controlled by torque tube level controls and diaphragm-type valves. Connections for liquid level gauges, pressure gauges, and drains are provided.

Capacities. Horizontal separators have high gas capacities because the mist extractor is installed longitudinally in the vessel. This arrangement permits the use of a large mist extractor inlet area.

Mist extractor design. This mist extractor incorporates a series of closely spaced baffles, which combine impingement, centrifugal force, surface tension, and gravity to effect separation. High capacity and low pressure drop are combined in this design. The high efficiency is maintained over the entire range of flow from practically zero to maximum rated flow.

Snubbers (*see* Pulsation Dampeners)

Stacks

Stacks can be used to conduct gases to be flared. The lit gas flame can be seen from the top of the stack in that case. This kind of stack may be called a *flare stack*.

Another kind of stack is a stack that exhausts the gaseous products of combustion, including water vapor and carbon dioxide to the atmosphere. The most severe application stresswise for a freestanding stack might be its use in an offshore environment due to wind loading and additional stress due to wave and water movement on a platform. Although this book is not intended to be a dedicated design text, it is useful for a process engineer in operations and maintenance to understand what to look for in a stack design. The following illustrates a stack design that helps cope with these stresses in an offshore environment. Note, however, that the design features presented apply to onshore designs as well.

The following material* describes the methods developed to optimize the mechanical design of a freestanding exhaust stack and its supporting structure. These particular methods have been used for the design of three gas turbine exhaust systems on a UK sector offshore platform currently under construction. The driving force behind the choice of a freestanding stack was to save weight and therefore cost. The move toward the development of marginal fields in deeper waters will only increase the need for lighter, and therefore more cost-effective design solutions.

---

* Source: Altair Filters International Limited, UK. Adapted with permission.

Although national standards that cover the basic design philosophy are available, these have serious limitations when applied to this type of structure. The aim of this section is to demonstrate how the limitations may be overcome by undertaking fundamental design analysis, and also to indicate those critical areas that demand special consideration. The detailed design analysis presented here considers the possibility of failure due to local instability, the effect on the dynamic response of the flexible foundation provided by the platform, and the determination of thermal stresses at critical locations.

Although this design has been developed for offshore use, the techniques utilized can be applied equally to onshore applications.

Nomenclature

V_s	design wind velocity, m/s
w	design wind load per unit length of stack, N/m
$L = L_4 - L_3$	unsupported height of stack, m
$L_o = L_3 - L_2$	spacing of upper pinned support, m
D	mean diameter of exhaust stack, m
D_1	spacing of main bearings, m
k_v	vertical elastic stiffness of foundation, kN/m
R_A, R_B and R_C	transverse support reactions, kN
V	vertical reaction at main bearing, kN
t	stack section material thickness, mm
σ	0.1 percent proof strength of stack material, MN/m
α	semiangle subtended by imperfection
n	vortex shedding frequency, Hz
S	vortex shedding coefficient
x	distance along stack axis, m
v	transverse displacement, mm
$w_s = \rho\pi Dt^2$	exhaust stack weight per unit length, kg/m
E	elastic modulus of stack material, GN/m
$I = \frac{\pi}{8} D^3 t$	second moment of area of stack section, m^4
k	elastic stiffness of supporting foundation, kN/m
g	acceleration due to gravity, ms^{-2}
ρ	specific weight of stack material, kg m^{-3}
m	circular frequency for transverse vibration, rad/s
$f = \frac{m}{2\pi}$	fundamental frequency in Hz
u_o	radial displacement at flange/shell intersection, mm
ϕ_o	rotation at flange/shell intersection
t_F	flange thickness, mm

The design of gas turbine exhaust systems on offshore platforms generally falls within well-proven parameters. The gas-carrying duct is suspended inside an external structural steel framework and connected via a system of mounts and guides to allow for thermal growth.

Extending the ductwork or stack above the steelwork such that the stack itself carries structural loads would appear to be a simple extension of proven designs. When the design of such a system was undertaken in practice, this was shown to be a long way from the truth.

Offshore oil and gas production platforms are well known for providing a particularly hostile environment for mechanical equipment operation. The North

Sea is probably one of the harshest examples of an offshore environment. Wind strengths are uncommonly high, with wind speeds of 45 m/s not unusual. In addition to the high wind strength, the rapid changes and gusting make conditions extremely unpredictable. Humidity levels are also high leading to problems with chloride attack by the saliferous atmosphere.

From a mechanical viewpoint major problems are caused to a tall slender structure by the flexibility of the platform. Consideration of its dynamic response compared with the wind-induced excitation is therefore of paramount importance. Thermal effects due to the hot exhaust gases are a further factor for consideration. The design must consequently take account of thermal stresses at critical locations. Choices of construction materials must be carefully considered. In this instance stainless steel grade 316 L was chosen as the most suitable to meet all the project requirements.

Choice of Design Philosophy

A typical design for an offshore gas turbine exhaust system is shown in Fig. S-36. The exhaust ducting is surrounded on all sides by a substantial steel framework. Upright structural members would typically be 254 mm × 254 mm × 73 kg/m universal column and horizontal members 457 mm × 191 mm × 67 kg/m universal beam. This structure would then be diagonally braced with 168 mm × 8 kg/m circular hollow section. A 2-m-diameter exhaust duct would require approximately 9 tonnes of such steelwork for every 5 m of stack height.

The steelwork would support the dead weight of the exhaust system and withstand all dynamic loads due to wind. The mounting system connecting the duct to the steelwork enables the loads to be transferred while allowing for thermal expansion of the system during operation. Normally the fixed support would be near the base of the stack and be capable of supporting the dead weight. Longitudinal thermal growth would then be vertically upward, with a system of guides allowing vertical movement while providing horizontal restraint.

The total exhaust stack length for the design in question is 24.4 m. The weight of platform steelwork required to fully support such a system would consequently be 45 tons. Recent North Sea developments have tended to be on more marginal fields, and therefore consideration of capital costs versus revenue has become crucial. With a typical installed platform cost of £3500 per ton of steelwork weight saving now takes a high profile.

The 45 tons of stack support structure is located at a high level on the platform. For this reason it is necessary to provide a further 45 tons of steel in the topside structure.

A freestanding exhaust system has a significant proportion of the upper ductwork unsupported by steelwork, as shown in Fig. S-36. The main support would be at the base of the freestanding section, with a system of guides for ductwork below this support as necessary.

The design study considered the option of both 10 m and 20 m of freestanding stack. Figure S-37 shows the relationship between the free length of stack and platform steelwork saving for a single stack of 2 m diameter. The design and fabrication costs can then be compared directly with the savings in steelwork to show the net savings. Table S-3 shows the results. The cost savings can be seen to be substantial. With three identical stacks being utilized on the platform in question, savings of £440,000 can be achieved with a platform weight reduction of 156 tons. This equates to a saving of 35 percent when compared with the total installed cost of the fully supported exhaust system.

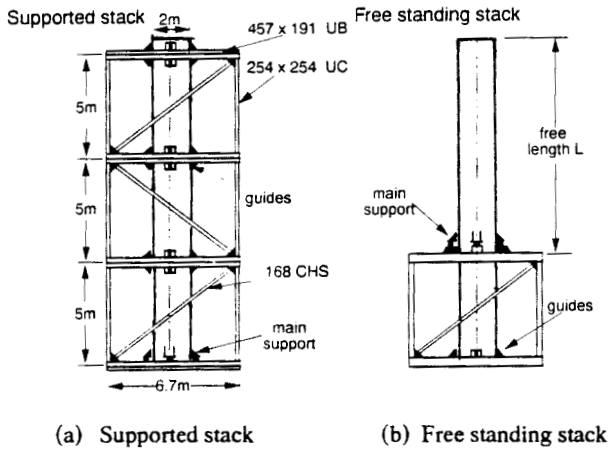


FIG. S-36 Exhaust stack showing alternative support arrangements. (Source: Altair Filters International Limited.)

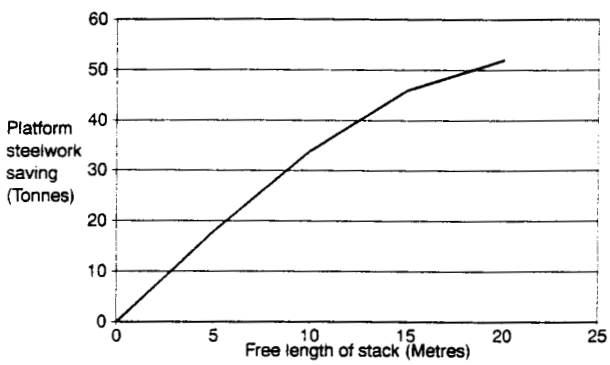


FIG. S-37 Relationship of free stack length to weight saving. (Source: Altair Filters International Limited.)

TABLE S-3 Net Cost Savings for Various Lengths of Freestanding Stack

Free Stack Length (m)	Stack Manufacturing Cost (£)	Platform Steelwork Saving (£)	Net Saving (£)	Percentage Saving
0	110,000	0	0	0
5	125,000	63,000	48,000	11
10	130,000	126,000	106,000	25
15	135,000	161,000	136,000	32
20	145,000	182,000	147,000	35

NOTES
1. The percentage saving relates to the total installed cost of a fully supported system. This is £425,000 made up from a stack cost of £110,000 plus 90 tons of steel at £3,500 per ton.
2. The stack manufacturing cost includes fabrication and design by an acoustic equipment manufacturer.
3. All prices quoted are UK pound sterling, and based on 1990 rates.

Design Parameters

The following base design parameters are defined as an example:

Lower stack connection	59.1 m above sea level
Upper stack termination	83.5 m above sea level
Freestanding stack length	10 m
Turbine type	Coberra 6462
Nominal power output	23 MW
Exhaust gas mass flow	90 kg/s
Power turbine gas exit temperature	475°C
Design wind speed	53 m/s
Material of construction	Stainless steel AISI 316L

From the turbine exhaust flow and system pressure loss limitations the duct internal diameter was sized at 2 m. Under maximum flow and temperature conditions the mean gas velocity in this duct is 60 m/s. Previous work on freestanding exhaust systems had highlighted the lack of a comprehensive standard that covered the structural aspects of such a system. Further investigations for this project confirmed this situation. The most applicable standard is BS4076 (1978). This specification offers guidelines for the design of freestanding chimneys, but has definite limitations. The most serious of these with respect to this design is that it makes no allowance for nonrigid foundations. The empirical formulae used are also based entirely on using carbon steel as the construction material. It is not clear what modifications would be needed to make these formulae applicable to stainless steels or other metallic alloys.

A further consideration highlighted by this standard is the warning on the interaction between pairs, rows, or groups of chimneys. In this case the three systems are positioned in a row at close pitch. The use of aerodynamic devices such as helical strakes to alter the response to gust loading of a single stack are well proven. Deeper investigation into the effectiveness of such arrangements on multiple arrays has again emphasized the serious limitations of available codes. Designers should, however, look closely at the possible impact of aerodynamic effects from nearby structures before finalizing their design. In this section consideration of structural aspects only are considered. The aerodynamic interaction of the three stacks on the platform in question played a major part in the decision to limit the freestanding height to 10 m above the steelwork.

Because of the unique aspects of this design and the considerable uncertainties in available design codes, a decision was made to develop directly applicable analysis methods from fundamental principles.

See Table S-3.

Steady-State Wind Loading

The wind loading on an isolated exhaust stack will depend on its geographic location, height, and the nature of the surrounding terrain. The design wind speed and aerodynamic force on the freestanding stack design have been determined in accordance with the recommendations of BSI (a British standard) CP3, Chapter V, Part 2 (1972). Typical values for an offshore environment are a design wind speed of 53 m/s, which for a 2-m-diameter exhaust stack corresponds to a drag force of $w = 3.6$ kN/m. For a stack mounted on pinned supports at three discrete levels as shown in Fig. S-38 the support reactions are not statically determinate.

It is necessary to integrate the general expression for the bending moment and

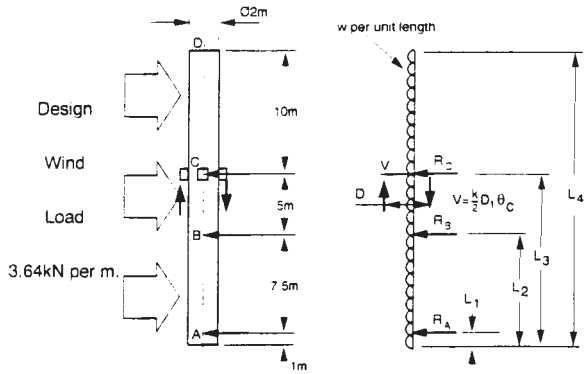


FIG. S-38 Stack idealization for steady wind load. (Source: Altair Filters International Limited.)

introduce the boundary conditions in the resulting displacement function. This yields four simultaneous equations that may be solved to give the individual reactions in the form:

$$R_A = \frac{K_2K_3 - K_1K_4}{K_3 - K_1}w \qquad R_B = \frac{K_4 - K_2}{K_3 - K_1}w \qquad R_C = wL_4 - R_A - R_B$$

and

$$V = \frac{1}{D_1}\left\{wL_4\left(L_3 - \frac{L_4}{4}\right) - R_A(L_3 - L_1) - R_B(L_3 - L_2)\right\}$$

where

$$K_1 = \frac{(L_3 - L_2)^3}{(L_3 - L_1)\{(L_3 - L_1)^2 - (L_2 - L_1)^2\}}$$

$$K_2 = \frac{(L_3^2 + L_1^2)(L_3 + L_1) - (L_2^2 + L_1^2)(L_2 + L_1)}{4\{(L_3 - L_1)^2 - (L_2 - L_1)^2\}}$$

$$K_3 = \frac{L_3 - L_2 + \dfrac{k_v D_1^2}{4EI}(L_2 - L_2)^2}{L_3 - L_1 + \dfrac{k_v D_1^2}{12EI}\{3(L_3 + L_1)^2 - (L_2 - L_1)^2\}}$$

$$K_4 = \frac{L_4\left(L_3 - \dfrac{1}{2}L_4\right) + \dfrac{k_v D_1^2}{48EI}\{4L_3^3 - (L_2^2 + L_1^2)(L_2 + L_1)\}}{L_3 - L_1 + \dfrac{k_v D_1^2}{12EI}\{3(L_3 - L_1)^2 - (L_2 - L_1)^2\}}$$

Figure S-39 shows the bending moment diagram for a stack with a free length of 10 m.

The maximum bending moment of 182 kN occurs at the main support where the transverse reaction R_C = 86 kN. Introducing a typical value k_v = 2 MNm^{-1} for the vertical stiffness of the supporting steelwork at the main bearing, it is interesting to find that the corresponding vertical force (V = 1 kN) is low when compared with the self-weight of the exhaust stack when the design wind load is applied. Consequently this element can be ignored during subsequent analysis.

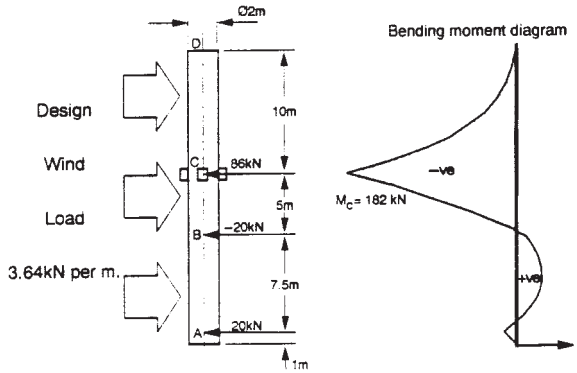


FIG. S-39 External reactions and bending moment diagram for exhaust stack subjected to design wind load. (Source: Altair Filters International Limited.)

The loading conditions have been determined in accordance with the recommendations of BSI CP3, Chapter V, Part 2 (1972), which is appropriate to the proposed location in the North Sea. For locations in the vicinity of North America it might be more appropriate to use the equivalent American National Standard (ANSI A58.1 1982).

Effects of Gust Loading

In addition to steady drag forces the exhaust stack is required to sustain the effects of gust loading. Appendix B of BS (a British standard) 4076 (1978) highlights procedures for avoiding aerodynamic excitation. In particular, the vortex shedding frequency for an isolated cylinder is given by the empirical formula

$$n = \frac{SV_s}{D}$$

For a 2-m-diameter cylinder supported in a turbulent airstream having a mean velocity V_s = 53 m/s, the coefficient S will have a value of 0.25, and the frequency of vortex shedding from the sides of the exhaust stack will be 6.2 Hz.

For practical installations it is known that the flow pattern will be three dimensional with additional vortices being generated by the flow over the top of the stack. The shedding frequency for such vortices may well be different from the value calculated above. To avoid any difficulties associated with aerodynamic excitation it is prudent to ensure that the fundamental frequency for transverse vibration of the exhaust stack is well above the primary vortex shedding frequency.

Unfortunately the formula given in BS 4076 calculates the natural frequency for transverse vibration of a cantilever mounted on a rigid foundation. While this is suitable for most land-based chimney designs it will yield an unduly optimistic estimate for an exhaust stack supported on pinned joints. The error will be increased further when consideration is also given to the flexibility of the supporting structure. It is clear that a more detailed analysis of the vibration behavior is required.

Examination of the bending moment diagram for inertia loading of an exhaust stack supported at three levels suggests that the natural frequency for the fundamental mode of transverse vibration will be almost entirely determined by the geometry of the upper sections. Accordingly for the purpose of frequency calculations only the two upper supports and corresponding stack sections have

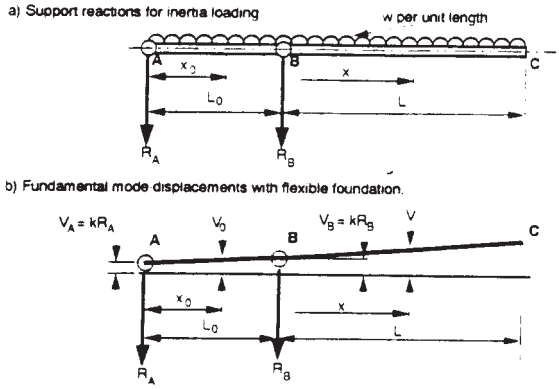


FIG. S-40 Idealization of exhaust stack for vibration analysis. (Source: Altair Filters International Limited.)

been modeled as a uniform beam mounted on pinned supports as shown in Fig. S-40*a*. This simplification has an immediate advantage in that the support reactions are statically determinate with values given by:

$$R_A = \frac{w}{2} KL_o \qquad \text{and} \qquad R_B = \frac{w}{2} KL_o \frac{L_o + L}{L_o - L} \qquad \text{where } K = 1 - \left(\frac{L}{L_o}\right)^2$$

Calculation of the fundamental frequency for transverse vibration of the beam system has been undertaken using Rayleigh's method, in which the maximum kinetic energy of the vibrating system is equated to the maximum bending strain energy. It is necessary to select realistic displacement functions that satisfy the boundary conditions of the problem. The supporting structure is flexible. This will affect the transverse displacement, and hence the natural frequency of the vibrating stack.

Suitable displacement functions for a foundation with stiffness k are illustrated in Fig. S-40. For vibration with a circular frequency m, these may be written

In section AB

$$v_o = \frac{w_s}{24EI}(x_o^4 + 2KL_o x_o^3 + A_o L_o^3 x + B_o L_o^4) \sin mt$$

where $$B_o = \frac{12EIk}{L_o^3} \qquad A_o = B_o \frac{2L}{L_o - L} + 2K - 1$$

and in section BC

$$v_o = \frac{w_s}{24EI}(x^4 - 4Lx^3 + 6L^2x^2 + AL^3x + BL^4) \sin mt$$

where $$B = B_o = \left(\frac{L_o}{L}\right)^4 \frac{L_o + L}{L_o - L} \qquad A = \left(\frac{L_o}{L}\right)^3 \left(B_o \frac{2L}{L_o - L} + 3 - 4k\right)$$

Equating the maximum values of the kinetic and bending strain energies, putting $w_s = \rho\pi Dt$ and $I = \frac{\pi}{8} D^3 t$, the fundamental frequency for transverse vibration is obtained as:

$$f = \frac{1}{2\pi} \sqrt{\frac{3.6ED^2 g}{\rho L^4} \frac{X}{Y}}$$

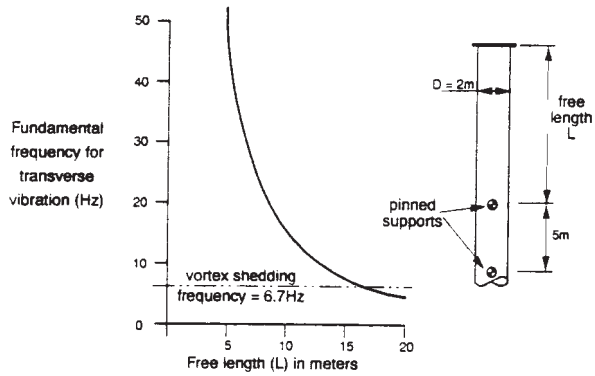


FIG. S-41 Variation in transverse vibration frequency with free length of exhaust stack. (Source: Altair Filters International Limited.)

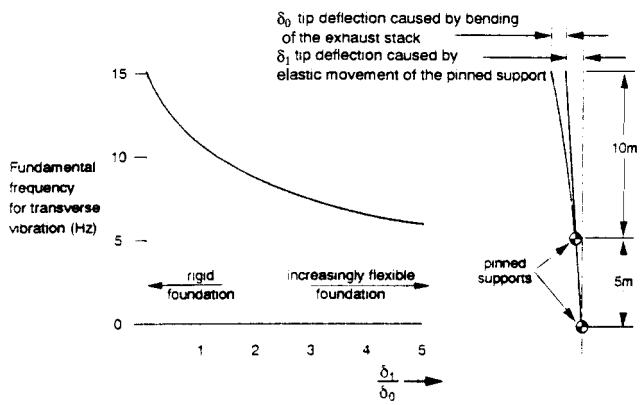


FIG. S-42 Effect of foundation flexibility in reducing natural frequency of transverse vibration. (Source: Altair Filters International Limited.)

where

$$X = 1 + \frac{1}{6}\left(\frac{L_o}{L}\right)^5 (6 - 15K + 10K^2) + B_o \frac{5}{12} K \left(\frac{L_o}{L}\right)^5 \left[1 + \left(\frac{L_o + L}{L_o - L}\right)^2\right]$$

and

$$Y = \frac{104}{45} + \frac{26}{15}A + \frac{12}{5}B + \frac{1}{3}A^2 + AB + B^2 + \left(\frac{L_o}{L}\right)^9 \left\{\frac{1}{9} - \frac{1}{2}K + \frac{4}{7}K^2 + \frac{5 - 12K}{15}A_o + \frac{2 - k}{5}B_o + \frac{1}{3}A_o^2 + A_oB_o + B_o^2\right\}$$

For a fixed foundation $k = 0$ and these expressions can be simplified with the fundamental frequency for transverse vibration being given by

$$f_o = \frac{1}{2\pi}\sqrt{\frac{3.6ED^2g}{\rho L^4} \cdot \frac{1 + \frac{1}{6}\left(\frac{L_o}{L}\right)^5 (6 - 15K + 10K^2)}{\frac{104}{45} + \frac{26A}{15} + \frac{1}{3}A^2 + \left(\frac{L_o}{L}\right)^9 \left(\frac{1}{9} - \frac{1}{2}K + \frac{4}{7}K^2 + \frac{5 - 12K}{15}A_o + \frac{1}{3}A_o^2\right)}}$$

The variation in f_o as the free length of a 2-m-diameter exhaust stack is increased from 5 to 20 m is shown in Fig. S-41. There is no doubt that the natural frequency

of the freestanding stack with a height of 20 m is too low for this configuration to be an acceptable design solution. The natural frequency of a stack with a free length of 10 m, mounted by pinned supports on a rigid foundation, is calculated to be 15.1 Hz.

It is interesting to note that the corresponding natural frequency for a 10-m-high cantilever mounted on a rigid foundation is 19.6 Hz. The difference between these values emphasizes the importance of ensuring that the recommendations of the relevant standard have been interpreted correctly.

For the pin-jointed configuration with a free length of 10 m the reduction in natural frequency with support flexibility is shown in Fig. S-42. It is convenient to express the flexibility of the support in terms of the tip deflection obtained with a rigid foundation.

For the proposed design it is important to note that even for a relatively stiff foundation, with a flexibility $^1/_k$ = 145 MN/m, which is typical of the support structure, the natural frequency falls to 10.9 Hz. It follows that the stiffness of a conventional support structure will be sufficiently low to have an adverse effect on the ideal dynamic response of the stack assembly.

The significant commercial benefits available from the correct choice of the design for offshore gas turbine exhaust systems and their supporting structures is evident. By using a freestanding exhaust the cost savings are shown to be substantial.

No comprehensive standard is available to assist the designer for this application. Should a design analysis be undertaken without appreciating the limitations of existing standards, the consequences could be disastrous.

These limitations have been highlighted, and a number of analysis procedures presented to illustrate how the deficiencies may be overcome. Use of the proposed procedures will allow a detailed and comprehensive design analysis to be completed for a freestanding stack. The method is simple to apply and allows parameter and optimization studies to be carried out within commercially viable time scales.

Freestanding stacks of even greater length can be achieved by prudent selection of configuration, material type, thickness, and support arrangement.

Steam Generator and Steam Supply

A steam generator consists of a boiler (*see* Some Commonly Used Specifications, Codes, Standards, and Texts, at the back of the book, for boiler specifications) its fuel system, and all controls and accessories. Steam supply is now a sophisticated science in itself, especially with supercritical steam now increasingly used to boost efficiency on steam-turbine cycles and other operations. Service factors on steam valves, lines, and other accessories are consequently more severe.

T

Tanks

Tanks are used for storage in process plants and refineries. Tanks for petrochemicals, oil, and other potentially explosive products are highly specialized.

A more common kind of tank is used for bulk dry storage of products in agriculture and a variety of other process industries. A variety of designs exist. The principle behind one of these types (trade name TecTank SealWeld) follows.

General Storage Tanks*

TecTank SealWeld tanks (see Fig. T-1) from Engineered Storage Products Company are designed for simple precision assembly. The critical flange-to-shell fabrication process incorporates positioning clamps and automated welding. Dual weld seams are used to increase the joint strength and to avoid warpage and distortion. During jobsite panel fit-up, flange connections are secured with structural bolts, not rivets. (See Fig. T-2.)

The interior welding of the assembled silo can proceed without the problems that plague traditional field-welded silos. Location costs are reduced, and the welding can be accomplished under controlled conditions. The welded interior of the product zone is free of ledges and gaskets.

Carefully controlled tank coating processes ensure the product within the tanks is not contaminated. Some examples of products stored are:

ABS Pellets
Alfalfa (Dehydrated)
Alumina Ore
Ammonium Nitrate
Barium Carbonate
Barium Sulfate
Bark Ash
Barley
Bauxite
Bentonite
Bisphenol “A”
Blood (Dried)
Bonemeal
Borax
Boric Acid
Brewers Grits
Burnt Lime
Calcium Carbonate
Calcium Chloride
Calcium Silicate

* Source: A.O. Smith Engineered Storage Products Company, USA. Adapted with permission.

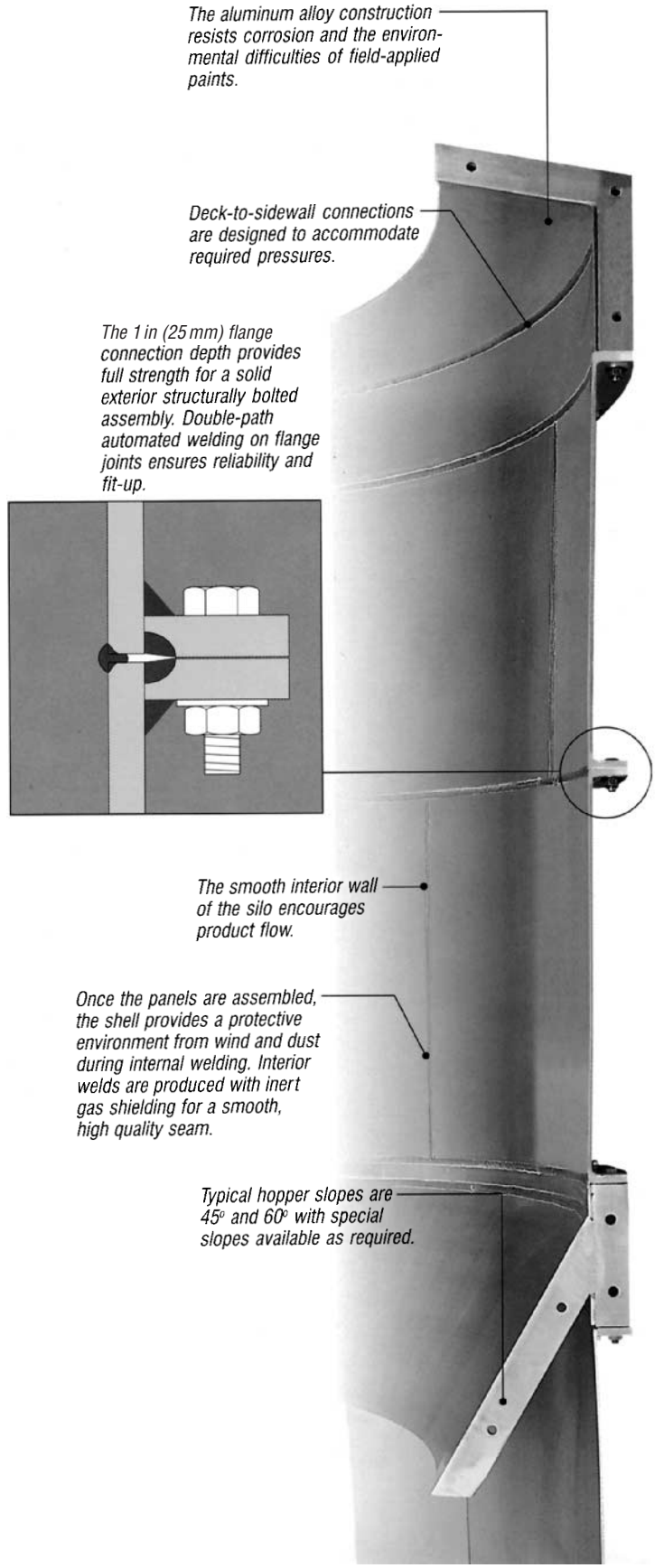


FIG. T-1 Generic storage tank (TecTank SealWeld tank). (Source: Peabody TecTank.)

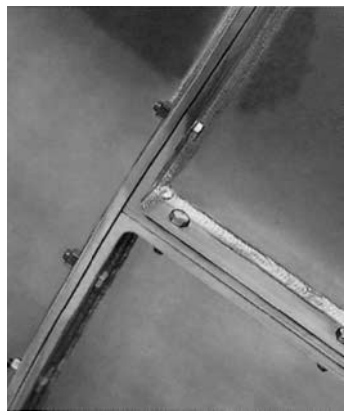


FIG. T-2 Panels are butted together and secured with bolted fasteners on the exterior of the silo. Exterior aluminum structural flanges are $^{1}/_{2}$ in (13 mm) thick and smooth with no ribs or grooves to retain debris. (Source: Peabody TecTank.)

Carbon Black
Cat Litter
Caustic Prills
Cellulose
Cement
Coffee Beans
Corn and Corn Cones
Cottonseed Meal
Diatomaceous Earth
Fish Meal
Flour
Gelatin Powder
Guar Beans
Gypsum, Calcined
Hydrated Lime
Inert Chemicals
Iron Ore
Isophthalic Acid
Kaolin Clay
Malt
Mill Feed
Para Formaldehyde Powder
Peanuts
Pebble Lime
Perlite
Pet Food
Phosphate Rock
Polyethylene Pellets
Polypropylene Pellets
Polystyrene Pellets
Polyvinyl Acetate
Polyvinyl Chloride
Portland Cement
Potash
Potassium Bicarbonate

Rice
Roofing Granules
Sewage Sludge (Dried)
Soap Powder
Soda Ash
Sodium Bicarbonate
Sodium Chloride
Sodium Sulfate
Sodium Sulfite
Sodium Tripolyphosphate
Soybean Meal
Starch
Sugar (Cane and Beet)
Sulfur
Terephthalic Acid
Tetra Sodium Pyrophosphate
Urea
Vermiculite
Vinyl Plastic Resins
Wheat
Wood Chips
Wood Waste
Zinc Oxide Dust

Information on typical welded tank interior and exterior coatings follows.

Gravity blending

Welded tank interior coatings

AR-Lon 6100. FDA and USDA excepted, the properties of AR-Lon 6100 markedly improve abrasion resistance and product flow. The chemically resistant epoxy primer has excellent adhesion and flexibility. The super-slick properties of polytef are combined with a urethane topcoat that provides additional impact protection and abrasion resistance. It is baked on at over 200°F (93°C).

Dry film thickness: 5 mils (125 microns) minimum

Abrasion resistance taber: 18 mg loss (CS17, 1000 g, 1000 cycles)

Falling sand, ASTM D-968: 60 liters/mil

Corrosion resistance: less than $\frac{1}{8}$ in creepage after

ASTM B-117: 1000 hr salt spray

Impact resistance: 100 in-lb

Heat resistance, dry: 250°F (120°C) continuous

FDA epoxy. The welded coating is a two-component, amine adduct epoxy coating with excellent adhesion, flexibility, and chemical resistance. FDA and USDA excepted, it's an excellent lining for most dry products.

Polyamide epoxy primer. This two-component, rust inhibitive primer outperforms standard *shop* primers. It's an excellent coating where FDA approval is not required.

Welded tank exterior coatings

"System 3." A premium factory-applied exterior coating system combines the long-term corrosion protection of the inorganic zinc primer with a polyamide epoxy intermediate coat and an acrylic urethane topcoat. The ultimate in welded tank exterior systems, System 3 provides *off shore* protection and extended color and gloss retention. This coating system is further enhanced by factory baking at 200°F (93°C).

Acrylic urethane. This two-component aliphatic acrylic urethane enamel retains its gloss and color even in severely corrosive and marine environments. Its very tough, abrasion-resistant film not only resists corrosion, but also salt spray, dirt, stains, acids, alkalies, solvents, crudes and petroleum products. Acrylic urethanes can provide more than double the life expectancy of conventional finishes.

Acrylic baking enamels. These true thermosetting acrylics provide excellent weathering, color and gloss retention characteristics, even in industrial environments. Using an automotive-type resin, acrylic baking enamels provide years of dependable service. It is an exclusive baking process that enables the use of these acrylic enamels on welded tanks.

Inorganic zinc primer. A primer that provides unequaled protection from coastal and marine exposures, inorganic zinc offers resistance to fumes and spillage of alcohols, petroleum products, and salt water. The galvanic action of its high zinc loading offers increased protection. Inorganic zinc primer is the base coat for TecTank System 3, and may be used as a primer-only system or topcoated for improved appearance and performance.

Polyamide epoxy primer. TecTank two-component polyamide epoxy primer is a lead and chromate free, rust inhibitive primer that provides long-term protection from corrosion. Polyamide epoxy outperforms standard *shop* primers and is an excellent primer for enamel, acrylic urethane, or field-applied topcoats.

Gravity blending in tanks is quite common in the process industries. For illustrative purposes a specific blending silo/tank combination has been selected (trade names Gravi-Merge Blending Silo* and Peabody TecTank*). See Figs. T-3 through T-5.

The Gravi-Merge Blending™ Silo minimizes the effects of batch variables for powders, granules, and pellets. It may be used to blend a wide range of materials, including plastics, foods, and chemicals. The blender produces a final product with consistent performance and/or color characteristics. The blender discharges a homogeneous mix that results from efficient internal recirculation and blending that occurs during filling, after filling (postfill recycle when required), and at discharge.

Conventional gravity fed blenders are limited by their "first in, first out" operation and by mixing "dead zones," both of which contribute to low blending performance. Internal recirculation via the lift pipe in blending silos corrects these deficiencies. The standard blend factor, the ratio of inlet to outlet standard deviations, of the blender is designed to exceed performance requirements. This reduces the effects of "off-spec" materials and delivers a premium product.

The lift pipe design makes the system more versatile than conventional gravity blenders. Both top-fill and energy-efficient bottom-fill silos are available for

---

* Trademark registered to Peabody TecTank.

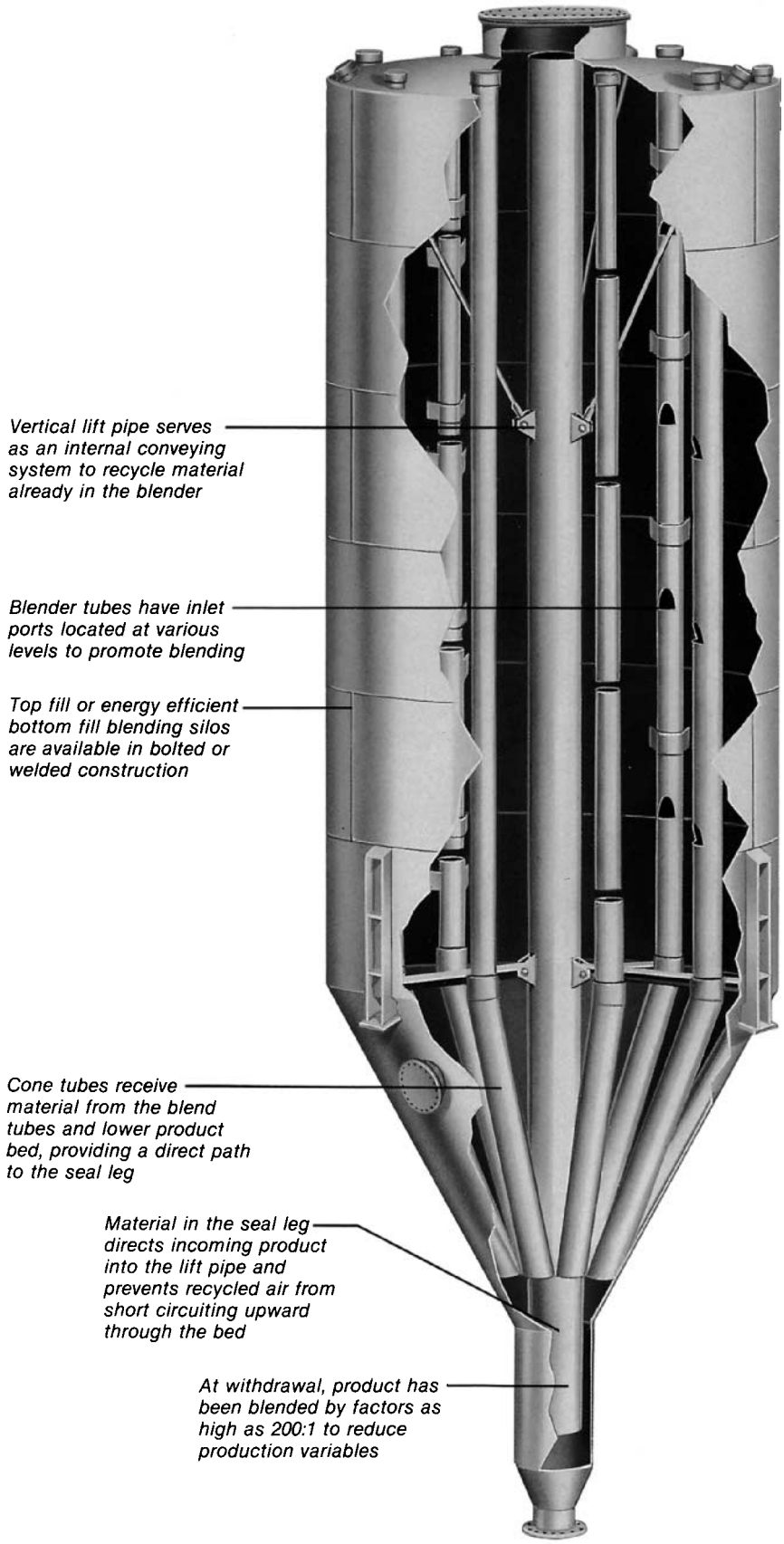


FIG. T-3 Gravity blending in a tank. (Source: Peabody TecTank.)

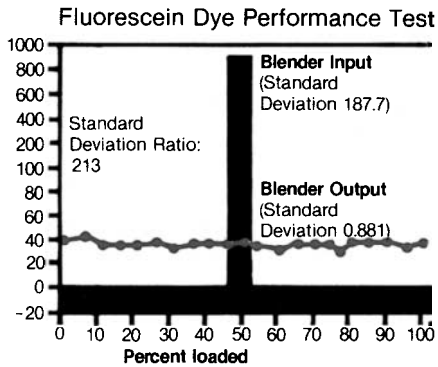


FIG. T-4 A concentration of fluorescein dye was added to the product during the filling cycle. The blender was emptied after filling was complete, with no additional blending. Samples were analyzed during withdrawal. (Source: Peabody TecTank.)

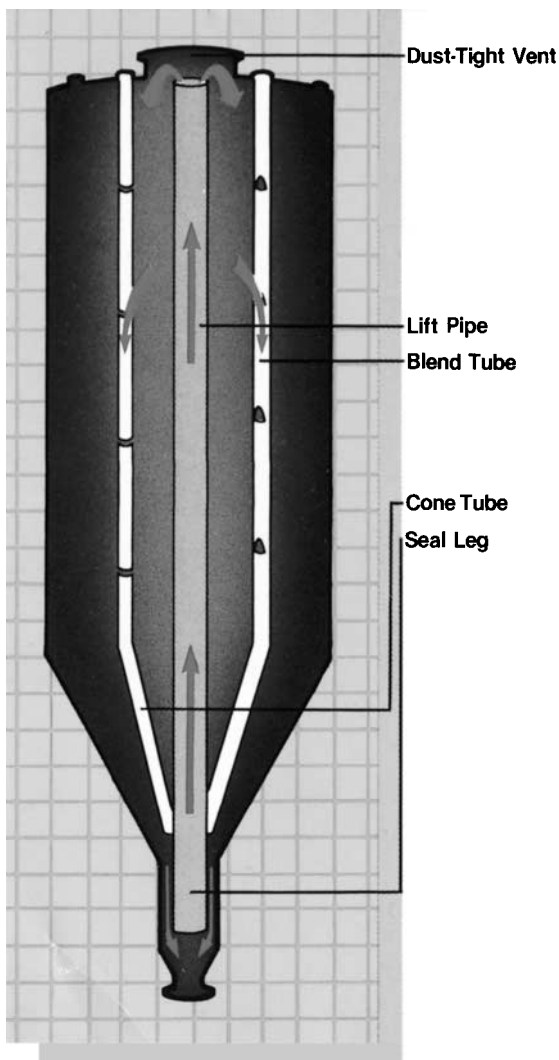


FIG. T-5 Pneumatic conveying fill system. (Source: Peabody TecTank.)

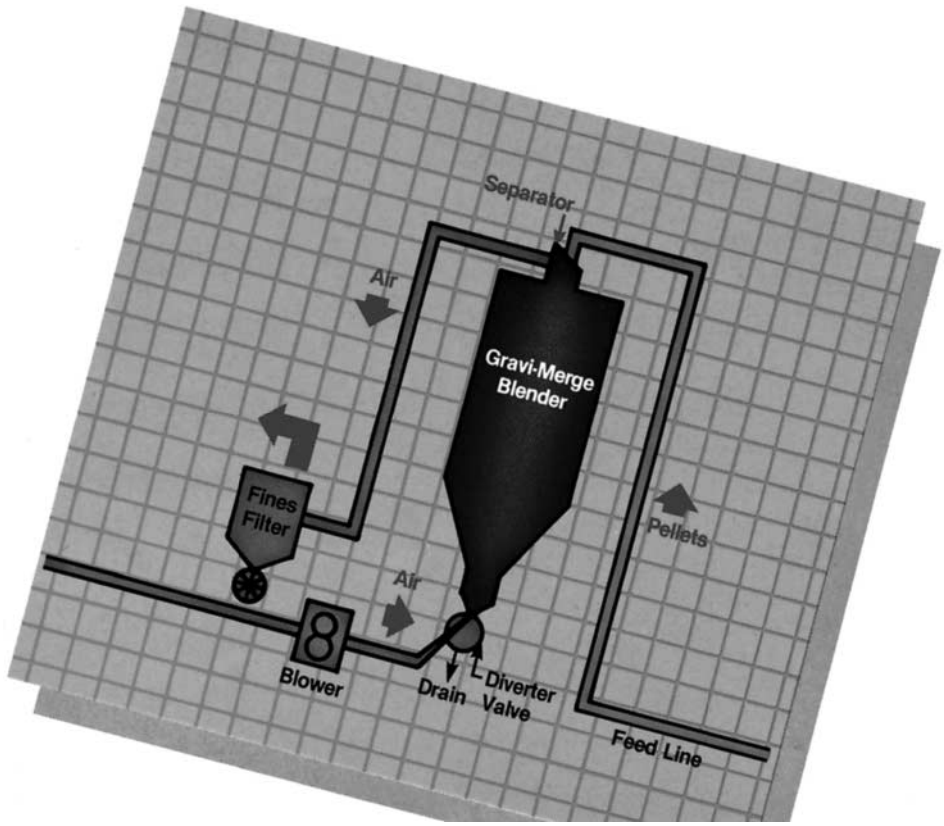


FIG. T-6 "Gravi-Merge" Blender. (Source: Peabody TecTank.)

blending granulars, powders, or pellets. The versatility of design frequently allows existing silos to be retrofitted with a blender.

Efficient internal recirculation blends while filling. Utilizing a hopper bottom tank, a lift pipe, and a series of channel pipes, the blender has four distinct mixing locations. (1) The lift pipe combines new material with the recirculating existing inventory during the fill cycle. (2) The gravity flow blend tubes simultaneously combine inventory from different levels. (3) Product within the main body of the tank moves downward by gravity. The difference in velocities between materials in the main body and the blend tubes promotes further blending. (4) Final blending is accomplished on withdrawal of material from the seal leg. All of these features result in the uniform flow of a consistent product from the blender. See Fig. T-6.

The silo is the first gravity blending system to offer efficiencies high enough for the processing market. Blend factors have been achieved that are comparable to those of high-intensity mechanical blenders. Blending silos may also be designed for continuous blending applications. Because the pneumatic conveying filling system is used in combination with the force of gravity, blending is cost efficient.

The assembly can incorporate an optional fines separator in either the top-fill or bottom-fill configuration. An integral wash system may also be specified as a component of the blending silo.

Storage Tanks for Liquids*

Specification: Factory coated, bolted steel for potable and process water storage

I. General

A. Scope of Work

Design and supply factory-coated, bolted steel water/wastewater storage tank(s), complete with assembly hardware, gasket, sealant, and accessories as shown on the contract drawings and described herein.

B. Tank Supplier

1. The engineer's selection of the products herein specified are predicated on a thorough examination of design criteria, construction methods, and comparative extended life-cycle analysis. Deviations from the specification will not be permitted except as noted in I.B.3 below.
2. Tank(s) to be used in the contract is (are) to be 100 percent new material, and is (are) manufactured by A.O. Smith Engineered Storage Products Company (ESPC) of Parsons, Kansas, USA.
3. Alternate tank products will be considered for approval by the engineer. Without exception, manufacturer shall maintain a current ISO-9001 certification. Approval submittals shall include:
   a. A reference list of five tanks presently in service of similar size and application operating satisfactorily for a minimum of five years.
   b. Technical information covering tank materials, designs, and coatings.
   c. Copy of manufacturer's current active ISO-9001 Quality System Certificate.
   d. Due to the potential aggressive corrosive nature of the products being stored, submittals will be required to be accompanied by statements of suitability for the specific intended purpose of this project covering coatings, gaskets, sealants, and hardware protection.

C. Submittal Drawings

The tank supplier shall furnish for the approval of the engineer, at no increase in contract price, three complete sets of construction drawings. When approved, one set of such prints will be returned to the bidder marked "Approved," and these drawings will then govern the work detailed thereon. The approval by the engineer of the supplier's drawings shall be an approval relating only to their general conformity with the project requirements and shall not guarantee detail dimensions and quantities.

II. Design Criteria, Codes, and Standards

A. Tank Capacity

Tank shall have a (nominal) (minimum) capacity of _____ (U.S. Gallons) (Imperial Gallons) (Cubic Meters) (Liters).

B. Tank Dimensions

The tank shall have a nominal diameter of _____ (feet) (meters) with a shell height of _____ (feet) (meters).

C. Tank Design Standards

1. The tank shall be designed as an atmospheric unit that operates on an equalized pressure, internally and externally. It shall be circular, constructed of carbon steel, and shall be designed in accordance with American Petroleum Institute (API) specification 12B principles (Specification for Bolted Tanks for Storage of Production Liquids) or American Water Works Association

---

* Source: A.O. Smith Engineered Storage Products Company, USA.

specification ANSI/AWWA D103-87 (Factory-Coated Bolted Steel Tanks for Water Storage). Bolted joints shall conform to API standard 12B. High-strength steel may be used in accordance with III.A.1 below.

2. The tank shall be provided with (*select one*):
   - an integral steel bottom, both sides of which shall be coated in accordance with Section 10.5 of the AWWA D103 specification.
   - a reinforced concrete floor (by others) with the tank walls embedded in the concrete.
3. If a deck is required, it shall be an integral steel deck, the underside of which shall be coated in accordance with Section 10.5 of the AWWA D103 specification. At the engineer's discretion and specification, a geodesic aluminum dome may be supplied.

D. Design Loads

1. Specific gravity: _____ (Minimum 1.00).
2. Maximum wind velocity: _____ (mph) (kph).
3. Roof Live Load: _____ (psf) (kg/m^2).
4. Earthquake (*select one*):
   - Seismic zone (1) (2) (3) per AWWA D103 using Fixed Percentage Method.
   - Seismic zone 4 (nonessential) (essential) per AWWA D103 using pseudodynamic approach.
5. Tank to store _____.
6. Stored product temperature (degrees F): maximum _____ minimum _____ normal _____.
7. Stored product pH range: maximum _____ minimum _____ normal _____.

III. Materials Specifications

A. Plates and Sheets

1. Steel plate shall conform to, or at a minimum shall be equal to, the requirements of ASTM A36 with a minimum yield strength of 36,000 psi. High-strength plate shall conform to, or at least be equal to, the requirements of ASTM A572-Gr 50 or 60, ASTM A607-Gr 50, 60, or 70.
2. Steel sheet shall conform to, or at a minimum shall be equal to, hot-rolled quality per ASTM A570 Gr 40 with a minimum yield strength of 40,000 psi. Minimum thickness shall be 12 gauge (nominal 2.65 mm).

B. Rolled Structural Shapes

1. Rolled structural shapes shall conform to ASTM A36.

C. Horizontal Wind Girders

1. When intermediate wind girders are required, the girders shall be either rolled structural shapes or a truss design of equivalent strength, coated the same as the tank exterior.

D. Hardware

1. Bolts used in tank joints shall be 1/2-in diameter and shall meet the minimum requirements of API-12B, Appendix A, and AWWA D103, Section 2.2.
2. Bolts shall conform to ASTM A307, ASTM A325, ASTM A490, or API-12B, as required by the tank design.
3. Bolts shall be mechanically galvanized to Class 50 of ASTM B695, or hot dip galvanized to ASTM A153. Nuts shall be hot dip galvanized to ASTM A153.

4. All bolts in contact with the stored liquid shall be provided with synthetic gasket-backed steel washers for placement between the nuts and the steel sheets. Other joints shall have steel flat washers under the nuts to protect the external coatings. Shell hardware exposed on the interior of the tank shall be plastic covered to protect against corrosion. If the tank is located in a coastal or corrosive industrial area, exterior hardware shall be plastic covered.
5. Gaskets and/or sealants shall conform to AWWA D103 Section 2.10.

IV. Coatings

A. Cleaning

1. Following fabrication, parts shall be thoroughly cleaned by a detergent wash/clean water rinse process followed immediately by hot air drying.

B. Surface Preparation

1. Following the wash/rinse and drying, the parts shall be steel grit blasted to a near-white finish SSPC-SP 10 (SA 2.5).
2. The surface anchor pattern shall not be less than 1.0 mil (25 microns).

C. Coating

1. All parts shall be factory coated after blasting; no shaping, bending, punching, flanging, or grinding may be done on the steel after blasting and before coating. Field coating, except for touchup, will not be permitted.
2. Interior coatings shall be Thermo-Thane 7000™ in accordance with AWWA D103 Section 10.5 and shall be NSF approved. Exterior coating system shall be (*select one*):
   - Polyamide epoxy primer with acrylic enamel topcoat.
   - Polyamide epoxy primer with acrylic urethane topcoat.
   - System 4™ consisting of an inorganic zinc prime coat, two intermediate coats of polyamide epoxy, and a finish coat of acrylic urethane.
3. Exterior color shall be (white) (light blue) (light green) (tan) (light gray).

D. Inspection

1. All coated parts shall be inspected prior to shipment, and shall be marked with a part number that shall correspond to the appropriate tank erection drawings to clarify and simplify tank assembly.
2. All coated sheets and parts shall be inspected for color uniformity.
3. A representative sampling of coated sheets shall be inspected in accordance with AWWA D103, Section 10.5.3 to verify minimum coating dry film thicknesses.

V. Appurtenances

A. The tank supplier shall furnish the appurtenances as shown on the engineer's drawings or as approved equal.

B. Unless otherwise noted, appurtenances shown on drawings shall be as follows:

1. **Hatch.** The tank roof hatch shall have a curbed, upward opening 24-in square manway. The curb shall extend at least 4 in above the tank. The hatch cover lip shall be hinged and provisions made for locking. The hatch cover lip shall extend for a distance of 2 in down on the outside of the curb. The hatch and cover shall be hot dip galvanized to ASTM A123.
2. **Inlet and outlet connections.** Inlet, outlet, and overflow connections shall conform to the sizes and locations specified on the plan sheets. All pipe

connection openings are to be cut into the steel panels in the factory before coating preparation and application unless otherwise specified by the engineer.

3. **Vent.** The free air vent shall be a mushroom-screened vent of sufficient size to accommodate normal inlet and outlet water flow (flow specified in plan sheets). The overflow pipe shall not be considered to be a tank vent. The vent shall be so designed and constructed as to prevent the entrance of birds or animals.
4. **Outside tank ladder.** An outside ladder with lockable hoop shall be hot dip galvanized and shall meet or exceed OSHA requirements.
5. **Deck perimeter guardrail.** A deck perimeter guardrail shall be galvanized and shall meet or exceed OSHA requirements. Guardrail shall have top rail, midrail, and toeboard.
6. **Shell manhole.** The shell manhole shall be circular, 24 inches in diameter, and shall include an extended neck for ease in cover removal and replacement. If the manhole cover weighs more than 50 lb, a cover hinge shall be provided.

VI. Shipping

A. All plates, supports, members, and miscellaneous parts shall be packaged for shipment in such a manner as to prevent abrasion or scratching of the finished coating.

1. Wall sheets shall be placed in disposable racks to separate adjacent sheets.
2. Deck and bottom sheets shall be stacked and bolted together.
3. Structural steel members shall be skidded and banded.
4. Miscellaneous small parts and hardware shall be boxed and crated.
5. Odd shaped parts, if not boxed or skidded, shall be individually secured.

B. For ocean shipments, if shipped as break-bulk cargo, all items shall be full box export crated. For containerized shipments, the following shall apply:

1. Wall sheet racks shall be blocked and braced.
2. Deck and bottom sheet stacks shall be full box crated and blocked and braced.
3. Structural steel members shall be full box crated and blocked and braced.
4. Miscellaneous small parts and hardware shall be crated, blocked, and braced.
5. Odd shaped parts not conducive to packing shall be individually blocked and braced.

VII. Erection

Field erection of factory-coated bolted steel tanks shall be in strict accordance with the manufacturer's recommendations. Particular care shall be exercised in handling and bolting of tank panels, supports, and members to avoid abrasion or scratching of coating. Touchup coating shall be done in accordance with manufacturer's recommendations.

VIII. Testing

1. Following completion of erection and cleaning of tank, it shall be tested for liquid tightness by filling the tank to its overflow elevation.
2. Any leaks disclosed by this test shall be corrected by the erector in accordance with the manufacturer's recommendations.

3. Water required for testing shall be furnished by owner without change at the time of erection completion.

IX. Foundations

1. The tank foundation is not a part of the tank supply contract.
2. The tank foundation shall be designed by the owner's engineer to safely sustain the structure and its live loads.
3. The foundation is to meet the requirements of AWWA D103. The top of the foundation shall be a minimum of 6 in (150 mm) above the finished grade, unless specified otherwise by the purchaser. Tanks that require anchor bolts shall be supported on a concrete ringwall or slab. Where steel floor sits directly on concrete, tank pad (1/2-in-thick cane-fiber joint filler to ASTM D1751) shall be supplied by tank manufacturer. The tank foundation shall be (*select one*):
   - Type 1. Tanks supported on ringwalls.
   - Type 2. Tanks supported on concrete slabs.
   - Type 3. Tanks within ringwalls.
   - Type 4. Tanks supported on granular berms.
   - Type 5. Tanks supported on granular berms with steel retainer rings. Steel retainer rings to be supplied by tank manufacturer.
   - Type 6. Tanks without steel floors supported on a concrete slab. Wall-to-slab connection details shall be in accordance with the manufacturer's recommendations. Installation of the foundation, bottom tank ring, and sealing of the tank wall to the slab shall be the responsibility of the general contractor or owner.

X. Warranty

The tank manufacturer shall warrant the tank against any defects in workmanship and materials for a period of one year from the date of shipment. In the event any such defect should appear, it shall be reported in writing to the manufacturer during the warranty period.

Tank Gauges*

What is tank gauging?

Tank gauging is the generic name for the static quantity assessment of liquid products in bulk storage tanks.

Two methods are recognized:

- A volume-based tank-gauging system: quantity assessment based on level- and temperature measurement.
- A mass-based tank-gauging system: quantity assessment based on hydrostatic pressure of the liquid column measurement.

Whatever method is used, a high degree of reliability and accuracy is of paramount importance when data are used for inventory control or custody transfer purposes.

---

* Source: Enraf, UK.

Refineries, chemical plants, terminals, and independent storage companies make use of bulk storage tanks for storage of liquid or liquefied products:

- Common bulk storage tanks are aboveground vertical cylindrical or spherical tanks.
- Vertical cylindrical tanks can be categorized as fixed roof tanks, with either a cone or dome roof construction, or floating roof tanks.
- Underground storage facilities such as caverns are used in areas where the soil structure permits this.

In order to reduce the vapor losses of fixed roof tanks they can be fitted with internal floating roofs or screens.

Liquefied gases are stored under pressure in spherical tanks, cylindrical vessels, or under refrigerated or cryogenic conditions in specially designed, well-insulated tanks.

Typical capacities of bulk storage tanks range from 1.000 m^3 (6300 bbl) to more than 120,000 m^3 (755,000 bbl). The value of the products stored in those tanks amounts to many millions of dollars.

A level uncertainty of only 1 mm (0.04 in) or 0.01 percent in a 10 m (33 ft) tall, 50.000 m^3 tank (315,000 bbl), equals 5 m^3 (31 bbl). Hence accuracy is a prime requisite for good inventory management; however, it is only one of the many aspects involved in tank gauging. Reliability to prevent product spills and safety of the environment and personnel are equally important.

The following listings show a number of requirements for tank-gauging systems.

General requirements for a tank-gauging system

- Safety
- Accuracy and repeatability
- Reliability and availability
- Compatibility with operations
- Stand-alone capabilities
- Operator-friendly
- Low maintenance
- Easy to expand

Additional requirements

- First-order failure detection
- Accepted for custody transfer and legal purposes (duties, royalties)
- Compatible with standards (API, etc.)
- Interface to host computer
- Software support
- Upgradability
- Service and spares support
- Acceptable price/performance ratio
- Vendor's quality assurance procedures (ISO 9000)
- Manuals and documentation

Why tank gauging?

Tank gauging is required for the assessment of tank contents, tank inventory control, and tank farm management. System requirements depend on the type of installation and operation.

The following types of operation, each having its own specific requirements, can be categorized:

- Inventory control
- Custody transfer
- Oil movement and operations
- Leak detection and reconciliation

Inventory control. Inventory control is one of the most important management tools for any refinery, terminal, or storage company. Inventory represents a large amount of assets for each company. Tank inventory control is either based on volume or mass. However, neither volume nor mass is the sole solution for accurate and complete inventory control. Products received, internal product transfers, and delivered products of refineries, chemical plants, and terminals are quite commonly measured in often incompatible volumetric or mass-based units.

Conversions from volume to mass and vice versa have to be frequently made, so that all measuring parameters such as product level, water interface, density, and temperature measurements are equally important.

The combination of volume and mass as realized in hybrid systems provides the most attractive solution.

In-plant accuracy requirements for inventory control are often noncritical. The measurement uncertainties do not result in direct financial losses. Reliability and repeatability are much more important.

Independent storage companies and terminals that strictly store and distribute products, owned by their customers, cannot operate without an accurate inventory control system. Such systems should be very reliable and accurate and provide all inventory data.

Custody transfer. Many installations used their tank-gauging system for the measurements of product transfers between ship and shore and/or pipeline transmission systems. A tank-gauging system is a very cost-effective and accurate solution compared to flow metering systems, especially when high flow rates are present and large quantities are transferred. When flow measuring systems are used, however, the tank-gauging system offers a perfect verification tool.

Where custody transfer or assessment of taxes, duties, or royalties are involved, the gauging instruments and inventory control system are required to be officially approved and certified for this purpose. In countries where such legal certification does not yet apply, verification of the measurements is often carried out by surveying companies. They generally use dip tapes, portable thermometers and sampling cans to measure level, temperature, and density prior to and after the product transfers. This is labor intensive and requires considerable time.

Surveyors use the same procedures to calculate volumes or mass as do modern tank-gauging systems. Hence the presence of a reliable, certified accurate tank-gauging system facilitates their surveys and will reduce the turnaround time. Another advantage is that in those cases where the quantity of product transferred is determined on the basis of opening and closing tank measurements, some systematic errors are canceled out. Hence the uncertainty of such transfer

measurements is better than can be expected on the basis of uncertainties specified for tank inventory.

Oil movement and operations. Generally tank content measurements for day-to-day operational use, for scheduling purposes, and for blending programs do not require the same accuracy as custody transfer operations. However, measurement reliability and repeatability are important. Reliable level alarms are also a must to operate safely. A high degree of accuracy and reliability will allow operations to safely use the maximum tank capacity. Past experience indicates that a 5 percent storage capacity gain can be achieved.

Oil movement and operations generally have very strict equipment requirements. They specify compatibility with their supervisory control and management systems. Operations will use availability and easy maintenance as the main criteria for selection of equipment. Cost of ownership calculations, however, can provide excellent insights into the selection or evaluation of alternative instrument and measurement techniques. Still, the user of these types of calculations should be careful to use only correct and valid arguments. For example, including the price of a stilling well in a comparative study for level gauges can be inappropriate if such a well is already part of the tank construction. Additionally, better performance, in terms of higher accuracy and lower maintenance, needs to be valued.

For oil movement and operations, either mass or volume measurement techniques can be used. Volume can be derived from level only; mass can be measured directly by means of pressure transmitters. Additional information can be obtained by measuring vapor temperature and pressure. Density measurement can also be added, with accuracies from 0.5 percent up to 0.1 percent. Whichever technique is selected, it should be compatible with the operations of all parties using the data from the tank-gauging system.

As stated earlier, plant management and control systems can facilitate oil movement and operations. Maintaining data integrity from the field to the receiving system is essential. A high degree of integration of the transmission of field instruments is a prerequisite. However, as long as a worldwide standard for digital communications is missing, different protocols will be in use.

Leak detection and reconciliation. For many decades the oil industry has been concerned with the financial consequences of oil losses. In recent years, there has also been an increased awareness of the industry's environmental impact. Pollution, caused both by liquid spills and atmospheric emissions, is an area of increased concern, and the industry has initiated programs to reduce the risks of environmental damage. Maintaining an accurate leak detection and reconciliation program is a necessity for any environmentally conscious tank farm owner.

At the fourth Oil Loss Control Conference in 1991, organized by the Institute of Petroleum in Great Britain, several leading authorities presented papers on nearly every aspect of loss control.

Dr. E. R. Robinson, consultant to the IP Refining Loss Accountability Committee, showed with a survey of 11 major UK refineries that an "average" refinery could have yearly losses of 0.56 percent of the total input quantity.

An accurate, reliable tank-gauging system helps to quantify and identify the source of these losses and offers the tools to prevent losses, or at least reduce them.

Another paper presented by Dr. J. Miles (SGS Redwood Ltd.) formulated an interesting approach to loss uncertainty assessment. Stock is mainly determined on the basis of tank measurement; however, inputs and outputs can also be assessed via flow (either volume or mass) and weighing bridge. Reconciliation of both



FIG. T-7 Manual gauging. (Source: Enraf.)

measurements holds the key to reliable inventory control and effective loss control.

A hybrid inventory measurement system (HIMS) combines mass- and volume-based inventory systems, improving the reliability and reducing uncertainties of the overall balance.

Tank-gauging techniques

Tank gauging has a long history. Since each user and every application has its own specific requirements, several measurement techniques and solutions to gauge tank contents are currently available.

Manual gauging. Tank gauging started with manual gauging (Fig. T-7), using a graduated diptape or dipstick. This technique is still used worldwide, and is today still the verification for gauge performance calibration and verification.

The typical accuracy of a diptape used for custody transfer measurements is often specified as $\pm (0.1 + 0.1\,L)$ mm [equal to $\pm (0.004 + 0.0012\,L')$ in] for the initial calibration of new diptapes. In the metric formula, L is the level in meters, and in the ft and inch formula, L' is the level in ft. For tapes in use, the recalibration accuracy applies. This accuracy is twice the uncertainty of a new tape.

But the tape uncertainty is not the only cause of error. Accurate hand dipping is a difficult task, particularly with high winds, cold weather, during night, or when special protection equipment has to be used. Additionally, a human error, of at least ±2 mm (±0.08 in), has to be added to the tape readings. API Standard 2545 is dedicated completely to manual tank gauging.

Another disadvantage of manual tank gauging is that employees are often not allowed to be on a tank because of safety regulations, resulting in costly, long waiting times.

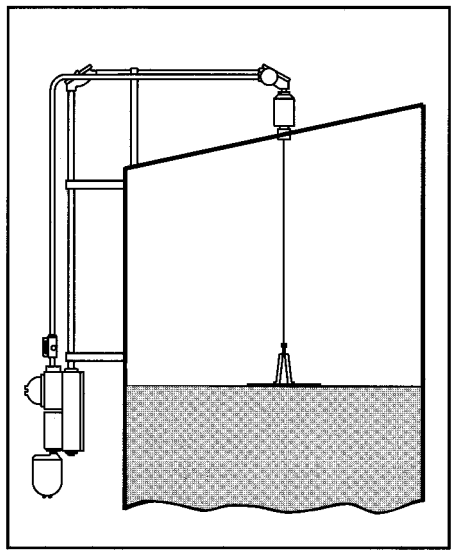


FIG. T-8 Float and tape gauge. (Source: Enraf.)

Float and tape gauges. The first float and tape gauges, also called automatic tank gauges, were introduced around 1930. These instruments use a large, heavy float in order to obtain sufficient driving force. Initially the float was connected via a cable to a balance weight with a scale and pointer along the tank shell indicating the level. Newer versions had the float connected, via a perforated steel tape, to a "constant" torque spring motor. The perforations drive a simple mechanical counter that acts as a local indicator. Typical accuracy of a mechanical gauge is in the range of 10 mm (1/2 in). Due to the mechanical friction in pulleys, spring motor, and indicator, the reliability is poor.

Remote indication is possible via an electronic transmitter coupled to the indicator. However, this will not improve the reliability or accuracy of the mechanical gauge.

One of the major disadvantages with float-driven instruments is the continuous sudden movement due to the turbulence of the liquid gauged. These movements, which can be rather violent, cause a continuous acceleration and deceleration of the drive mechanism, resulting in excessive wear and tear of the local indicator, transmitter, and other devices coupled to the gauge. The reversing motions and accelerations cannot be followed by the indicating system and transmitter. Often the gear mechanism, driving the indicator and transmitter shaft, disengages, resulting in erroneous readings and desynchronization of the transmitter. This leads to considerable maintenance and lack of measurement reliability. In light of the present worldwide concern to prevent product spills, these gauges should no longer be used. Because of their low price, however, a large share of the world's tanks are still equipped with these instruments. See Fig. T-8.

Servo gauges. Servo tank gauges (Fig. T-9) are a considerable improvement over the float-driven instruments. They were developed during the 1950s. In this gauge, the float is replaced by a small displacer, suspended by a strong, flexible measuring wire. Instead of a spring motor, servo gauges use an electrical servo motor to raise and lower the displacer. An ingenious weighing system continuously measures the



FIG. T-9 Servo gauge. (Source: Enraf.)

weight and buoyancy of the displacer and controls the servo system. The motor also drives the integral transmitter.

Mechanical friction in the servo system, transmitter, local indicator and alarm switches has no effect on the sensitivity and accuracy of the gauge. Also, turbulence has no direct effect. An integrator in the servo control system eliminates the effects of sudden product movements. The gauge not only produces an average level measurement under turbulent conditions, but it also eliminates unnecessary movements and reduces wear and tear, greatly extending the operational life of the instrument.

The original servo gauge does not look much like today's modern version. The instruments have evolved into highly reliable mature products, and are gradually replacing mechanical float gauges, cutting down on maintenance and improving on inventory results. Modern intelligent servo gauges have very few moving parts, resulting in long-term reliability and accuracy. They also have a high degree of data processing power.

The instruments do not merely measure the liquid level but are also capable of measuring interface levels and product density.

Accurate, programmable level alarms are standard. Accuracies of better than 1 mm (1/16 in) over a 40-m (125-ft) range can be attained.

The exceptional accuracy and reliability has resulted in the acceptance of the measurements and remote transmission, by Weights & Measures and Customs & Excise authorities in many countries.

Radar gauges. The use of radar to measure product levels in storage tanks is one of the most recent techniques. Radar level gauges were developed in the mid-sixties for crude carriers. The majority of these ships were equipped with mechanical float-



FIG. T-10 Radar level gauge of free space measurement. (Source: Enraf.)

driven gauges. The level gauges were only used when the ship was ashore, loading or unloading. New safety procedures for tank washing with closed tanks during the return voyage, and the necessity to fill the empty tank space with inert gas, made nonintrusive measurements preferable. Accuracy was less important for the level measurement of the cargo tanks, since custody transfer and fiscal measurements used the certified level gauges or flow meters of the shore installation.

Radar level gauges do not have moving parts and only an antenna is required in the tank. This results in very low maintenance cost. Although the investments costs are higher when compared to float gauges, the cost of ownership will be considerably lower.

The radar instruments use microwaves, generally in the 10 GHz range, for the measurement of the liquid level. The distance the signal has traveled is calculated from a comparison of transmitted and reflected signals. With tank gauging, relatively short distances have to be measured. Electromagnetic waves travel with nearly the speed of light. Because of the short distances ranging from some centimeters (inches) to, e.g., 20 m (66 ft), and the required resolution, a measurement based on time is almost impossible. The solution is to vary the frequency of the transmitted signal and measure the frequency shift between transmitted and reflected signal. The distance can be calculated from this frequency shift.

Now radar level gauges are available for product storage tanks found in refineries, terminals, chemical industries, and independent storage companies. The absence of moving parts, their compact design, and their nonintrusive nature result in low maintenance costs and make them very attractive. In order to achieve an accuracy ten times better than for use in marine applications, specific antennas and full digital signal processing have been applied.

Older radar instruments were equipped with large parabolic or long horn antennas, whereas the modern radar level gauges use planar antenna techniques. These antennas are compact and have a much better efficiency, resulting in excellent accuracy.

Several antenna types are available to suit virtually every tank configuration:

- Free space propagation is the most common method and is used if the gauge is installed on top of a fixed roof tank (Fig. T-10).



FIG. T-11 Radar level gauge for stilling well measurement. (Source: Enraf.)

- On floating roof tanks, the radar gauge can be installed on the guide pole. A specific radar signal (circular mode signal) is than guided via the inner shell of the guide pole or support pipe (Fig. T-11).
- Radar gauges can also be used on high-pressure storage vessels. An isolation valve can be installed between the vessel and the instrument. Verification and calibration is possible while the instrument remains in service.

Accurate measurement on products with very low vapor pressures is possible with the latest radar gauging technique.

Radar gauges are also a logical choice for tanks containing highly viscous products, such as blown asphalts, contaminating products, and liquids that are very turbulent.

Hydrostatic tank gauging. Hydrostatic tank gauging (HTG) is one of the oldest techniques to measure the tank contents. In the process industry, level measurement using differential pressure transmitters is very common. Normally this method uses analog pressure transmitters, with a 1 percent accuracy. Inventory measurement requires a much better accuracy; therefore, analog transmitters are not suitable for this purpose.

Specially calibrated smart digital pressure transmitters are now available to provide much better accuracy. The onboard microprocessor allows compensation for temperature effects and systematic transmitter deviations. HTG makes use of these accurate pressure transmitters for a continuous mass measurement of the tank contents (Fig. T-12).

Various HTG configurations are available:

- A simple HTG system can be built with only a single transmitter near the tank bottom (P1). The total mass can be calculated by multiplying the measured pressure by the equivalent area of the tank.
- By adding a second transmitter (P2) at a known distance from P1, the observed density (dens. obs.) of the product can be calculated from the pressure difference P1 – P2. The level can be calculated from the density and the P1 pressure.

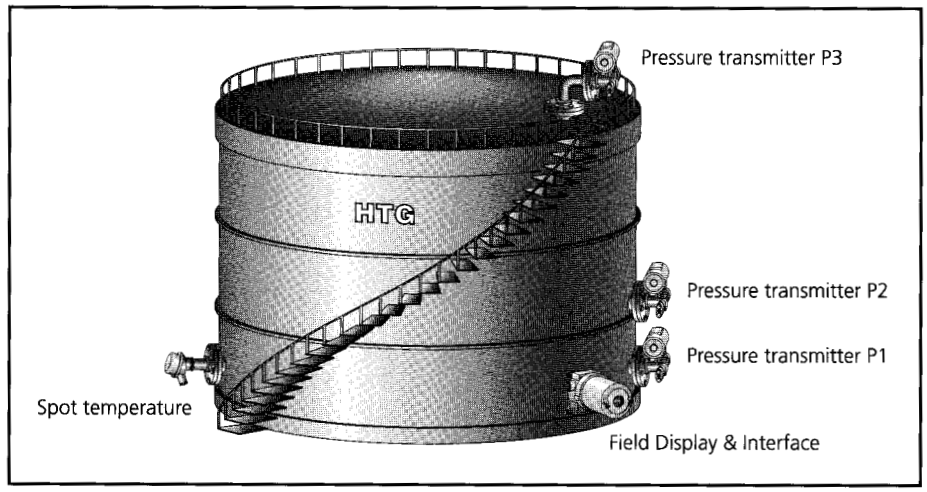


FIG. T-12 Hydrostatic tank gauging system. (Source: Enraf.)

- A P3 or top transmitter can be added to eliminate the effect of the vapor pressure on the P1 and P2 transmitters.

For pressurized tanks, HTG is less suitable. The large difference between the storage pressure and small hydrostatic pressure variations (turndown ratio) causes inaccurate results. Also, the fitting of the transmitter nozzles on spheres is costly and often unacceptable.

On atmospheric tanks, HTG systems offer a 0.5 percent uncertainty or better for the mass measurement. The accuracy of the HTG level measurement, although sufficient for the determination of the equivalent area, is 40 to 60 mm ($1\frac{1}{2}$ to 2 in) and totally unacceptable for custody transfer or inventory assessment. Hence many companies require the addition of a dedicated level gauge.

A drawback of the HTG system is that its density measurement is only over a limited range near the bottom of the tank. If the liquid level is above the P2 transmitter, the calculated value is based on active measurements. However, if the level is under P2 there is no differential pressure measurement. This will be the case when the level is only 1.5 to 2.5 m (6 to 8 ft) above the tank bottom. With many tanks, the density in the heel of the tank will be different from the density at higher levels. This density stratification has a devastating effect on the calculated values for level and volume. Since the level measurement of a HTG system is very inaccurate, it becomes worthless for any form of overfill protection. Secondary high level alarms are essential.

Hybrid inventory measurement system. The hybrid inventory measurement system (HIMS) combines the most modern level gauging techniques with hydrostatic tank gauging (Figs. T-13 and T-14). It utilizes an advanced radar or servo level gauge for accurate level measurement, with a smart pressure transmitter (P1) and a temperature measurement instrument. On nonatmospheric tanks a second transmitter for the vapor pressure compensation is required.

The level measurement is the basis for an accurate volume inventory calculation. The pressure measurement, combined with the level, provides a true average density measurement over the entire level height. This average density is used for

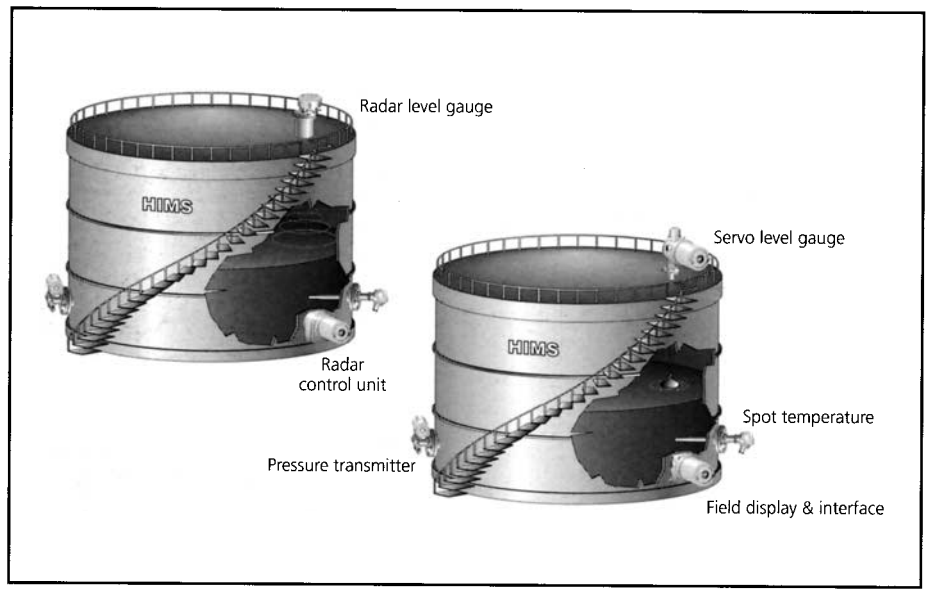


FIG. T-13 HIMS systems with radar or servo level gauge. (Source: Enraf.)

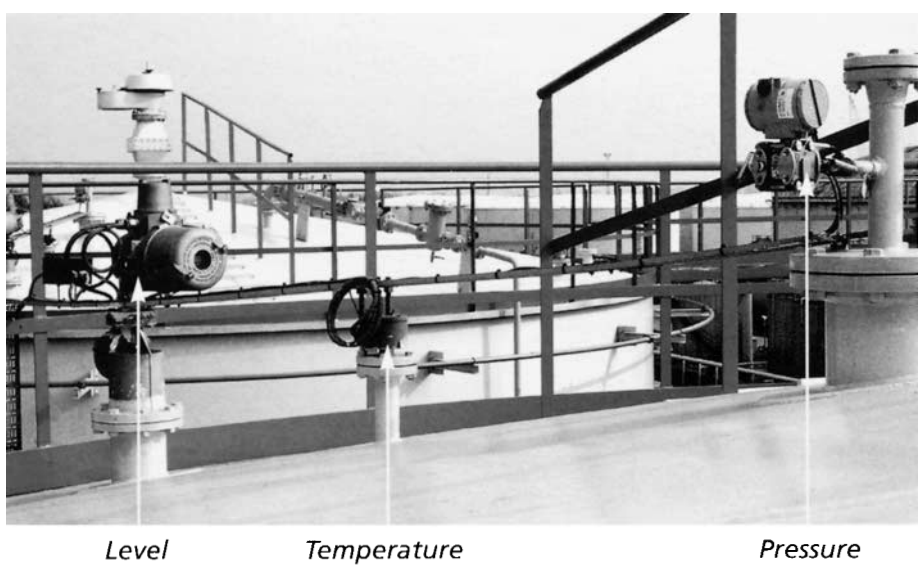


FIG. T-14 Upper part of a HIMS installation. (Source: Enraf.)

the mass assessment. The temperature is used to calculate standard volumes and densities at reference temperatures.

Advanced servo gauges and radar gauges can be provided with an interface board that communicates directly with the smart pressure transmitter. The result is a unique and very complete measurement providing level, interface levels, product-water interface levels, average density, average temperature, vapor temperature, and alarms.

Existing installations with advanced radar or servo level gauges can, in most cases, easily be extended to become a HIMS system.

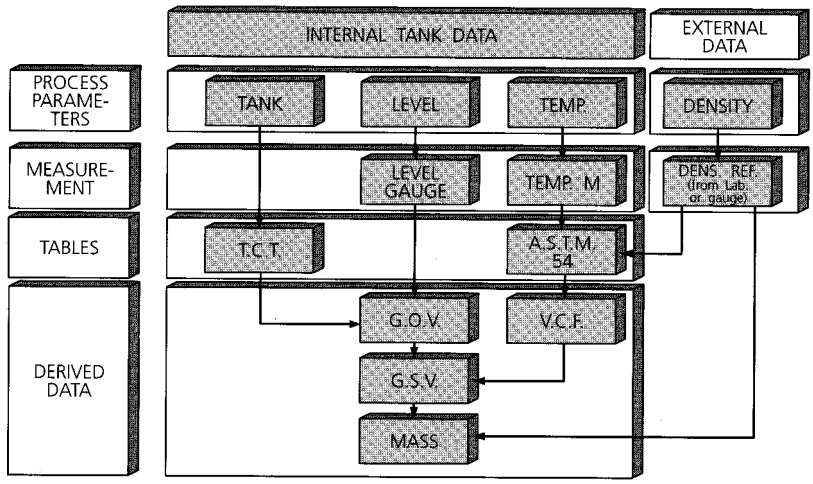


FIG. T-15 Level-based quantity assessment. (Source: Enraf.)

HIMS is often called "The best of both worlds," providing the best of level gauging combined with the best of hydrostatic gauging.

Quantity assessment in tank gauging

The uncertainties of quantity assessment of a tank-gauging system depend on the measuring uncertainties of the installed instruments, tank capacity table (TCT) and installation.

Level gauging instruments measure the liquid level in the tank. Pressure transmitters measure the hydrostatic pressure of the liquid column. Both level and pressure are primary functions for the calculation of volume and mass, respectively. Hybrid systems, such as HIMS, use both inputs in one system. Conversions from volume to mass or vice versa are made using density and temperature as secondary inputs. The density input may be obtained from an outside source, such as a laboratory, or may be measured in the tank by using pressure transmitters or servo density. The temperature input is obtained from a temperature-measuring system in the tank.

How the individual errors influence a mass or volume uncertainty depends on the type of quantity assessment.

Level-based quantity assessment. Figure T-15 shows how the quantity assessment in a conventional-level (volume)—based system is accomplished.

The tank references, liquid level, liquid temperature, and liquid density are the relevant parameters.

- Level is measured using a radar or servo level gauge.
- Temperature is measured using a spot or average temperature sensor.
- Density at reference temperature is obtained from a laboratory analysis of a grab sample.

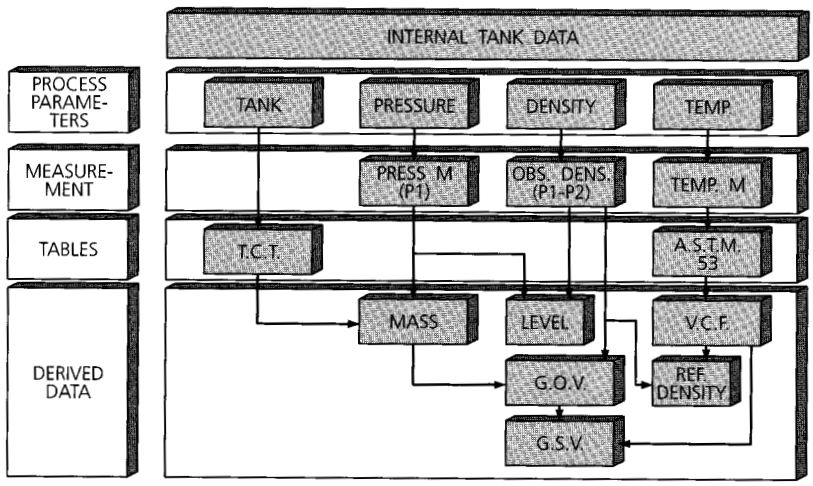


FIG. T-16 HTG-based quantity assessment. (Source: Enraf.)

- The gross observed volume (GOV) is derived from level and the TCT.
- The gross standard volume (GSV) is calculated from the GOV, corrected with the volume correction factor (VCF).
- The VCF is derived from the temperature measurement using ASTM Table 54 and the density at reference temperature (DENS. REF.).
- The total MASS is calculated from the GSV multiplied by the DENS. REF.

The MASS of the product can also be calculated from the net standard volume as the GOV minus sediment contents and water.

Major causes for uncertainties are the temperature assessment and the TCT. Additional functionality can be added to enhance the total performance, e.g., vapor pressure and water interface measurement.

Hydrostatic-based quantity assessment. The quantity assessment of a HTG-based system is shown in Fig. T-16. The tank references, hydrostatic liquid pressure, liquid density, and liquid temperature are the relevant parameters.

- Pressure M is measured via pressure transmitter P1.
- The DENS. OBS. is measured using pressure transmitters P1 and P2.
- Temperature can be measured for GSV calculations with a temperature sensor.
- The MASS is directly calculated from the equivalent area and the P1 (PRESS. M) transmitter. The equivalent area is obtained from the TCT.
- The GOV is derived from mass and the observed density.
- The observed density is derived from the differential pressure measurement of P1 – P2 and the distance between both transmitters.
- The GSV is calculated from the GOV, corrected with the VCF.
- The VCF is derived from the temperature measurement using ASTM Table 54 and the DENS. REF.

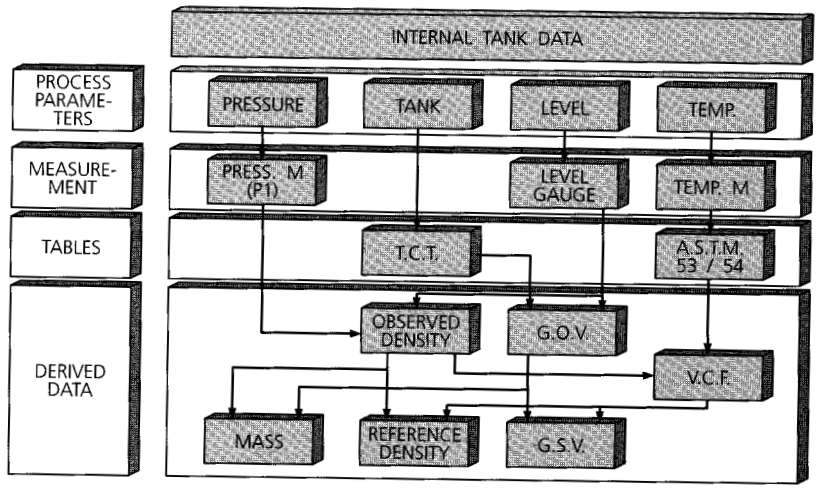


FIG. T-17 HIMS-based quantity assessment. (Source: Enraf.)

- The level is derived from the PRESS. M and DENS. OBS. measurement obtained from P1 and P2.
- The DENS. REF. is derived from the DENS.OBS. corrected with the VCF.

Major uncertainties in a HTG system are caused by the TCT, the pressure transmitters, and calculations using an incorrect density value as a result of nonhomogeneous products.

Variations of the temperature do not influence the mass accuracy. The temperature is required for the calculation of the density under reference conditions and GSV.

Hybrid-based quantity assessment. The quantity assessment of a HIMS-based system is shown in Fig. T-17.

The hydrostatic liquid pressure, tank references, liquid level, and liquid temperature are the relevant parameters.

- The hydrostatic pressure is measured using pressure transmitter P1.
- Level is measured by an advanced radar or servo level gauge.
- Temperature is measured using a spot temperature sensor or average temperature sensor.

The system is basically the same as the level-based system; however, the density is derived from the hydrostatic pressure (PRESS. M) measured by P1 and the height of the liquid column on P1.

- The GOV is derived from level and the TCT.
- The GSV is calculated from the GOV and corrected with the VCF.
- The MASS, however, is directly calculated from the GOV and DENS. OBS. from PRESS. M measured by P1.
- The DENS. REF. is calculated from DENS. OBS. corrected with the VCF.

- The VCF in this case is derived from the temperature measurement using ASTM Table 54 and the DENS. OBS.

HIMS provides, as an additional benefit, a highly accurate continuous average density measurement.

The average observed density is determined over the entire level height. This is a unique feature because all other systems determine the density at one or more specific levels or over a limited range of 2 to 3 m (6.6 to 10 ft) only.

Uncertainties in tank gauging

In order to compare the different quantity assessment systems, it is necessary to analyze all parameters affecting the final uncertainty of each gauging system.

Instrument data sheets usually only state accuracies under reference conditions. Mass and volume accuracies derived from these data are often too optimistic. For correct interpretation of data sheets and justification of the choice of instruments, errors caused by the installation should also be taken into account. This can be difficult. Even within international organizations dealing with standardization, much time is spent to establish the correct way to calculate or determine final uncertainties.

An uncertainty analysis for tank gauging was developed in order to get a better understanding of the mechanisms and parameters involved. On the basis of this analysis, a number of graphs and data tables have been produced, illustrating the uncertainties of the measurement systems dealt with in this document. Analysis was done both for inventory and batch transfers. All uncertainties are expressed as relative values, i.e., as percentages of the inventory or the quantity transferred, as is customary in loss control and custody transfer.

The comparison makes use of generic specifications of uncertainties for tank-gauging equipment, storage tanks, and installation. The data used are assumed to be manufacturer independent.

Sources of errors. The overall uncertainty in the quantity assessment is the combined result of all uncertainties of each single parameter in the calculation. In order to obtain the optimal accuracy of a specific gauge, careful installation is required. This applies to all types of gauges. Figure T-18 shows the major sources of errors in tank gauging.

- Bulk storage tanks are not designed to serve as measuring vessels. Their actual shape is influenced by many factors. Computerized compensation for some of these effects is possible, provided the effects are known and reproducible. For the best accuracy obtainable with level measuring devices, a stable gauging platform is a prerequisite. The use of a support pipe is an available and known technique and is already present on many tanks, with and without floating roofs. The presence of such a pipe is an advantage that makes the best accuracy possible when choosing instruments in a revamp project.

  For radar gauges, existing pipes can be used to provide mechanical stability. Circular mode antennas are required when installation on a pipe is foreseen. On high-pressure tanks, installation of an insert with reference pins is recommended.

- Temperature is an often underestimated measuring parameter. An accurate average temperature measurement is essential to achieve accurate inventory calculations. Spot measurements are not useful when the product temperature is stratified.

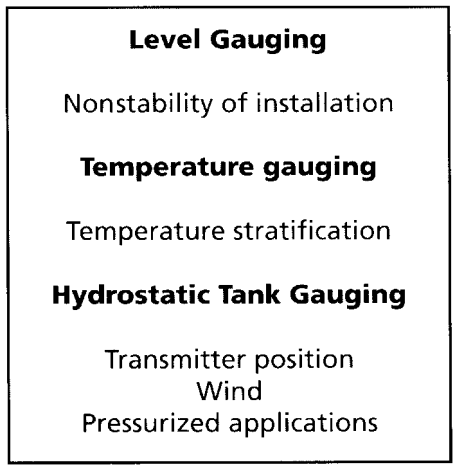


FIG. T-18 Major sources of errors. (Source: Enraf.)

- Equipment used in HTG systems are installed external to the tank. With existing tanks hot tapping, an installation method while the tank remains in service may be the solution when company regulations permit. This technique is fully developed, but there are different opinions on the safety aspects. The P1 transmitter must be installed as low as possible, but above maximum water and sediment level. It is important to realize that the product below the P1 nozzle is not actually measured. This restriction severely limits the minimum quantity that can be measured for custody and tax purposes.

A study performed by the Dutch Weight & Measures showed that wind can cause errors as much as 0.2 percent on a 10-m-high (33-ft) tank. On fixed roof tanks, compensation for this error can be accomplished with an external connection between P1 and P3. High nominal operation pressures encountered in spheres and bullet-type vessels require specially developed transmitters. The measurement of the small signal superimposed on the high pressure reduces the accuracy.

Overview of error sources. Tables T-1 and T-2 show respective overviews on uncertainties on inventory and batch transfer for level-based systems (servo/radar), HIMS and HTG systems.

Safety

Hazards of fire and explosions. The majority of tank-gauging instruments are installed on tanks containing flammable products. The instruments on such tanks or in the surrounding hazardous area must be explosion proof. Circuits entering the tank atmosphere, like temperature-measuring systems, should be intrinsically safe. In the past, each country had its own safety standards, but an international harmonization of standards has become a reality.

The European Cenelec standards and the American NFPA standards are acceptable in many countries. Safety, i.e., the fact that the explosion-proof or intrinsically safe construction meets the standards, must be certified by an

TABLE T-1 Overview of Inventory Uncertainties

Inventory		Level Servo/Radar		HIMS		HTG	
(m)	(ft)	Mass	G.S.V.	G.S.V.	Mass	Mass	G.S.V.
20	66	0.12	0.06	0.06	0.04	0.04	0.43
10	33	0.12	0.07	0.07	0.08	0.08	0.41
2	6.5	0.13	0.08	0.08	0.40	0.40	0.34

Inventory uncertainties in (%)

TABLE T-2 Overview of Batch Transfer Uncertainties

Transfer		Level Servo/Radar		HIMS		HTG	
(m)	(ft)	Mass	GSV	GSV	Mass	Mass	GSV
20–18	66–60	0.31	0.30	0.30	0.28	0.28	3.09
4–2	13–6.5	0.14	0.10	0.10	0.28	0.28	0.61
20–26	66–6.5	0.11	0.04	0.04	0.03	0.03	0.47

Batch transfer uncertainties in (%)

NOTE: For level-based systems (servo/radar) the density is obtained from the laboratory analysis of a grab sample; the uncertainty is assumed to be ±0.1 percent.

independent approval institute. Well-known institutes are PTB (Germany), Factory Mutual Research (USA), SAA (Australia), JIS (Japan), and CSA (Canada).

The better tank-gauging instruments do not just meet the safety standards but exceed them by anticipating future safety requirements as well. Such requirements include the exclusion of aluminum inside storage tanks (zone 0), the limitation of the kinetic energy of moving parts of a gauge to values far less than could cause ignition.

Lightning and tank gauging. Lightning can cause hazardous situations, and measures should be taken to protect the tank installation and tank-gauging system against these hazards. Modern tank-gauging systems contain many electronic circuits. Their position on top of storage tanks makes this equipment more vulnerable to lightning damage than any other type of industrial equipment.

Today's communication systems linking all field equipment via one network increase the probability of possible damage to the equipment as the networks spread over increasingly larger areas. With high reliability and availability one of the prime requirements of modern tank-gauging equipment, there is a need for well-designed, field-proven lightning protection methods. Figure T-19 shows a tank gauge under high voltage test.

In tank farms, lightning causes a direct potential difference between the gauge, grounded to the tank at one end, and the central receiver, at the other end. This results in a potential difference between cable and gauge or cable and receiver. This difference between equipment and cable tries to equalize itself and searches a low

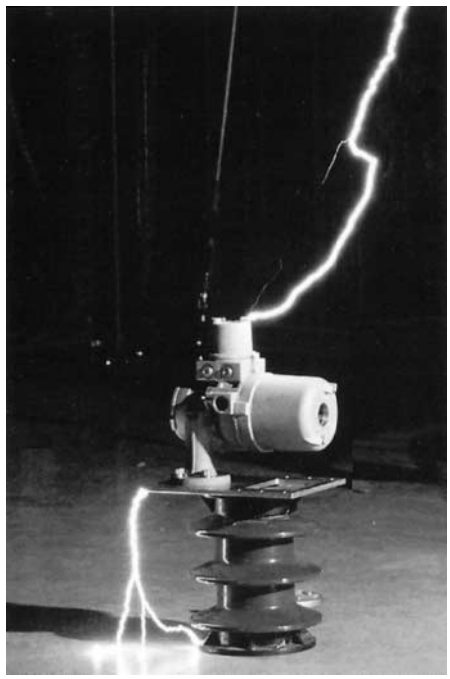


FIG. T-19 Tank gauge under high voltage test. (Source: Enraf.)

impedance path between the circuitry connected to the cable and the ground. As soon as the potential difference exceeds the isolation voltage, a breakdown occurs between the electronics and the ground. Additionally, transient currents will be induced in adjacent components and cabling.

The currents flowing through the electronics cause disastrous effects. Every semiconductor that is not sufficiently fast or capable of handling the currents for even a short period will be destroyed.

Two basic techniques are used for minimizing the damage due to lightning and transients: suppression and diversion.

Suppression. By means of special circuits on all incoming and outgoing instrument cables it is possible to suppress the magnitude of the transient appearing at the instrument (Fig. T-20). A gas discharge tube forms the kernel. Gas discharge tubes are available for voltage protection from 60 V up to more than 1000 V and react in several microseconds, after which they form a conducting ionized path. They provide no protection until they are fully conducting.

A transzorb or varistor, in combination with a resistor and preferably an inductor, can be added to improve the protection. These semiconductors react within a couple of nanoseconds and limit the voltage. A major problem is that each time a transient suppressor reacts, it degrades. Reliability is therefore poor, rendering this type of device unsuitable for critical applications such as tank gauges.

Diversion. Diversion (Fig. T-21) is a much more reliable technique and better suited for lightning protection of electronic tank-gauging instruments. Modern protection uses diversion combined with screening and complete galvanic isolation. It is a technique in which the high-voltage spikes are diverted rather than dissipated. Specially developed isolation transformers are used for all inputs and outputs. They have two separate internal ground shields between primary and

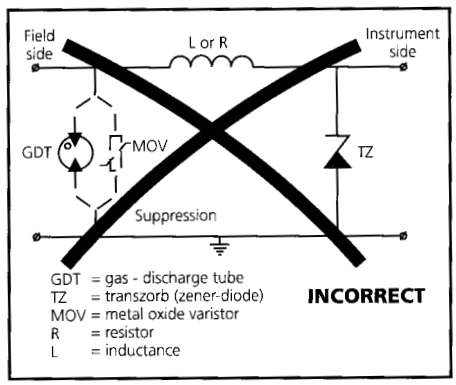


FIG. T-20 Suppression circuit. (Source: Enraf.)

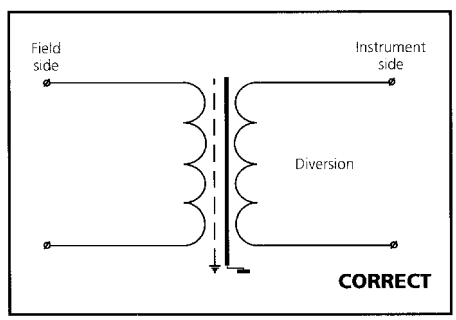


FIG. T-21 Diversion circuit. (Source: Enraf.)

secondary windings and the transformer core. External wiring is physically separated from internal wiring and ground tracks are employed on all circuit boards to shield electronics. Unfortunately this protection method is not suitable with DC signals. In this case a conventional transient protection, enhanced with an additional galvanic isolation, is used.

Grounding and shielding. Proper grounding and shielding will also help protect instruments and systems connected to field cabling against damage by lightning. The possible discharge path over an instrument flange (e.g., of a level gauge) and the corresponding mounting flange should have a nearly zero resistance to prevent buildup of potential differences. A poor or disconnected ground connection may cause sparking and ignite the surrounding product vapors.

Field experience. The diversion method described for internal lightning protection has been in use for more than 15 years, with approximately 50,000 installed instruments. Almost 100 percent of this equipment is installed on top of bulk storage tanks, and interconnected via wide area networking.

A large number of installations are situated in known lightning-prone areas. To date, only a few incidents in which lightning may have played a decisive role have been experienced. The amount of damage was always limited and could be repaired locally at little expense. Before this protection method was applied, more extensive lightning damage had been experienced.

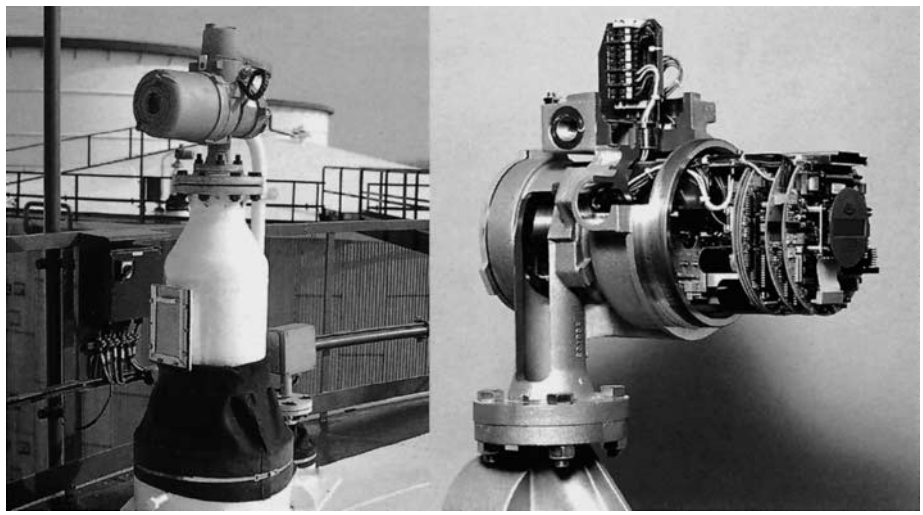


FIG. T-22 Advanced technology servo gauge. (Source: Enraf.)

Developments in tank-gauging technology

Servo gauges. Modern servo gauges are already members of the sixth generation (Fig. T-22). They use modern embedded microcontrollers, minimizing the total amount of electronics. Advanced software development tools and higher order programming languages provide reliable operation. Fuzzy control algorithms improve interaction of mechanics and electronics, reducing the number of mechanical parts.

Current advanced servo tank gauges (ATG) have less than five moving parts.

The main features of an advanced technology servo gauge are:

- Low operating cost
- Typical MTBF of more than 10 years
- Low installation cost, especially when used to replace existing servo gauges
- A standard accuracy of better than 1 mm (0.04 in)
- Software compensation for hydrostatic tank deformation, making support pipes no longer a must for accurate measurement
- Full programmability for easy setup and simple maintenance without having to open the instrument
- Compact and lightweight construction requiring no hoisting equipment
- Possibilities for installation while the tank stays in full operation
- Continuous diagnostics to provide maximum reliability and availability
- Water-product interface measurement for time-scheduled water measurement
- Spot and average product density measurement
- Interfacing to other smart transmitters, e.g., for product and vapor temperature, and pressure via a digital protocol, including average density support

The German legislation currently accepts advanced servo gauges as a single alarm for overfill protection.

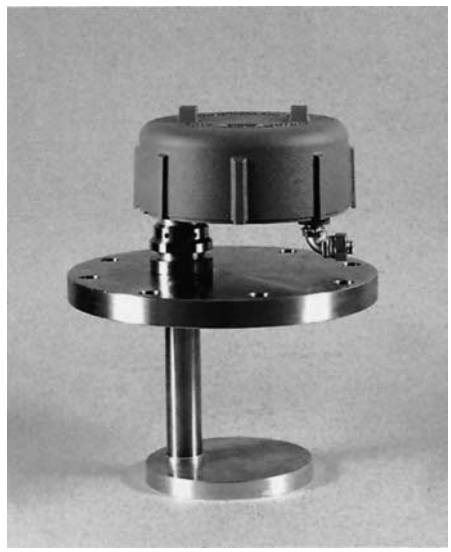


FIG. T-23 Radar level gauge with Planar antenna technology. (Source: Enraf.)

Radar gauges. Radar gauges play an important role in tank gauging (Figs. T-23 and T-24). Their nonintrusive solid-state nature makes them very attractive. The accuracy of the newest generation radar gauge meets all requirements for custody transfer and legal inventory measurements.

Reliability is high and maintenance will be further reduced. The onboard intelligence allows for remote diagnosis of the total instrument performance. The compact and lightweight construction simplifies installation without the need for hoisting equipment. Installation is possible while the tank stays in full operation. Current developments are aimed at more integrated functions. Improved antenna designs, full digital signal generation, and processing offer better performance with less interaction between tank and radar beam.

The main features of the new generation radar level gauge are:

- No moving parts
- Very low maintenance cost
- Low operational cost
- Nonintrusive instrument
- Low installation cost
- Typical MTBF of more than 60 years
- Low cost of ownership
- Modular design
- A standard accuracy of ±1 mm (0.04 in)
- Software compensation for the hydrostatic tank deformation, making support pipes no longer a must for accurate measurement
- Full programmability for easy setup and verification facilities
- The compact and lightweight construction eliminating the need for hoisting equipment



FIG. T-24 Radar level gauge for high-pressure applications. (Source: Enraf.)

- Installation possibilities while the tank stays in operation
- Continuous diagnostics providing a maximum of reliability
- Water-product interface measurement using digital integrated probe
- Density measurement via system-integrated pressure transmitter (HIMS)
- Interfacing to other transmitters, e.g., for product and vapor temperature, and pressure via digital protocol

Temperature gauging. Accurate temperature measurement is essential for level-based tank-gauging systems.

Spot temperature elements are widely accepted for product temperature assessment on tanks with homogeneous products. Installation is simple and the reliability is good. The graph of Fig. T-25 shows that spot measurements are unsuitable to accurately measure the temperature of products that tend to stratify. The effects of temperature stratification can be neglected only for light products, mixed frequently.

In general, average temperature-measuring elements are used in case of temperature stratification. The latest development is the multitemperature thermometer (MTT) shown in Fig. T-26 that utilizes 16 thermosensors evenly distributed over the maximum possible liquid height. A very accurate class A Pt100 element at the bottom is the reference. Accuracies of better than 0.05°C (0.08°F) are possible. The elements can also be individually measured to obtain temperature profiles and vapor temperatures. MTTs are available with both nylon and stainless steel protection tubes. It provides a rugged construction suitable for the harsh environments of a bulk storage tank.

Another type of average temperature measuring element is the multiresistance thermometer (MRT). Its operation is based on a number of copper wire temperature sensing elements of different lengths. Average temperature measurement is achieved by measuring the longest fully immersed resistance thermometer chosen

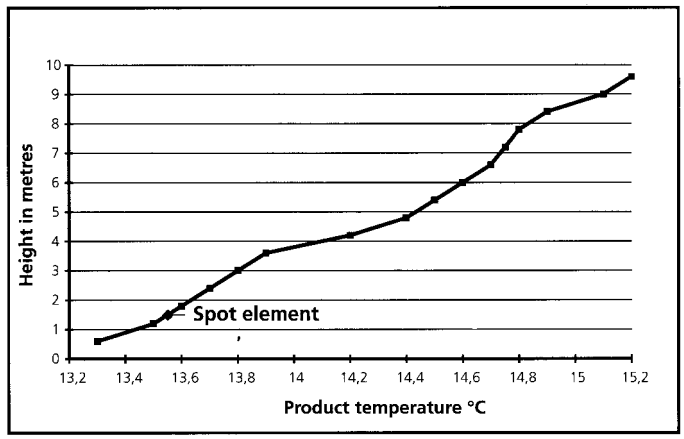


FIG. T-25 Temperature stratification in a storage tank. (Source: Enraf.)

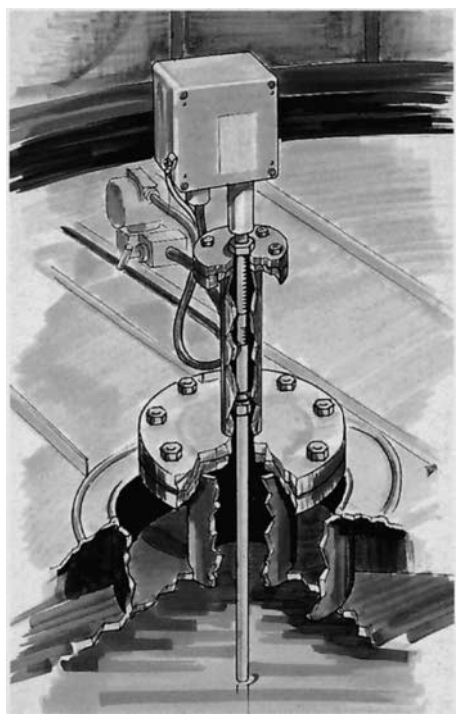


FIG. T-26 Average temperature sensor with selector/interface unit. (Source: Enraf.)

by a solid-state element selector. A drawback of MRTs is the delicate construction of the elements. The very thin copper wire used makes the device susceptible to damage, especially during transport and installation.

Hydrostatic tank gauging. Recent developments of smart transmitters opened a new era for HTG. The development of smart pressure transmitters with microcomputers



FIG. T-27 Central inventory management system. (Source: Enraf.)

made HTG feasible. Only a couple of years ago, high-accuracy pressure transmitters were still rare and quite expensive. Several manufacturers now offer 0.02 percent accuracy transmitters. Digital communication by means of de facto standards, as the HART™-protocol, permits simple interfacing to almost any transmitter. This wide choice simplifies selection for specific applications and allows the user to choose his own preferred transmitter. The inherent standardization for the end user reduces the cost of maintenance.

Hybrid inventory measurement system. HIMS are also based on the integration of smart pressure transmitters. Modern level gauges, either servo or radar, provide the possibility for direct interfacing to smart pressure transmitters. HIMS opens the ideal route to total tank inventory systems, measuring all tank parameters via one system.

Central inventory management system. The interface to the operators and/or the supervisory control and management system is the tank-gauging inventory management system (Fig. T-27). These high-speed systems collect the measurement data from all tank-gauging instruments, continuously check the status of alarms and functional parameters, and compute real-time inventory data such as volume and mass. The hardware used is generally off-the-shelf personal or industrial computers loaded with dedicated inventory management software. It is this software, together with the reliability and integrity of the field instrumentation that determines the performance and accuracy of the inventory management system. All field instruments, regardless of age or type, should communicate via the same transmission bus.

Product volumes and mass should be calculated the same way as do the owner-appointed authorities and surveyors. The system software should store the tank table parameters, calculate observed and standard volumes, correct for free water and, if applicable, correct for the floating roof immersion. The GSV calculations must be in accordance with API, ASTM, and ISO recommendations implementing tables 6A, 6B, 6C, 53, 54A, 54B, 54C, and 5.

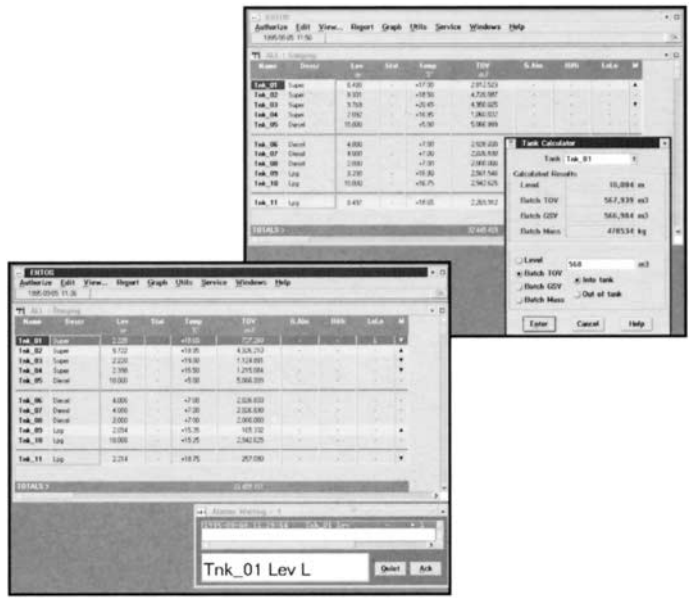


FIG. T-28 Tabular screens of an inventory management system. (Source: Enraf.)

The quality of the inventory management system can be evidenced from the availability of Weights & Measures or Customs & Excise approvals. Inventory management systems can have their own display consoles or can make all data available for a supervisory system.

Networked systems are available when required. Apart from a large number of inventory management functions, the system can also control inlet and outlet valves of the tanks, start and stop pumps, display data from other transmitters, provide shipping documents, provide trend curves, show bar graph displays, perform sensitive leak detection, calculate flow rates, control alarm annunciation relays, perform numerous diagnostic tasks, and much more. For examples of display formats of an inventory management system see Fig. T-28 for tabular displays and Fig. T-29 for graphical displays.

The operator friendliness of the system is of paramount importance. The better and more advanced systems have context-sensitive help keys that make the proper help instructions immediately available to the operator.

Interfacing to host systems. The receiving systems can also be equipped with host communication interfaces for connection to plant management systems, e.g., Distributive Control Systems (DCS), Integrated Control Systems (ICS), oil accounting systems, etc. (Fig. T-30). Protocols have been developed in close cooperation with the well-known control system suppliers.

These are needed in order to transmit and receive the typical tank-gauging measuring data. Standard protocols as Extended MODBUS, Standard MODBUS, and others are also available for smooth communication between tank inventory systems and third-party control systems. Modern DCS or other systems have sufficient power to handle inventory calculations, but often lack the dedicated programming required for a capable inventory management.

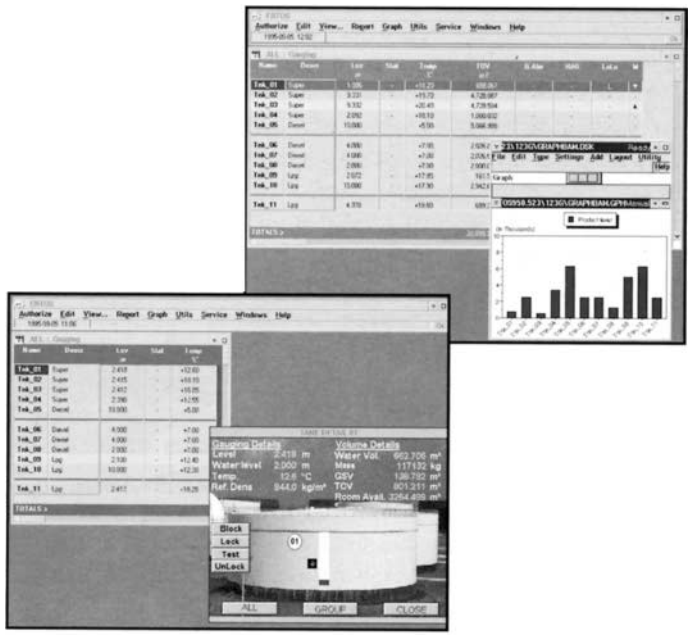


FIG. T-29 Graphical screens of an inventory management system. (Source: Enraf.)

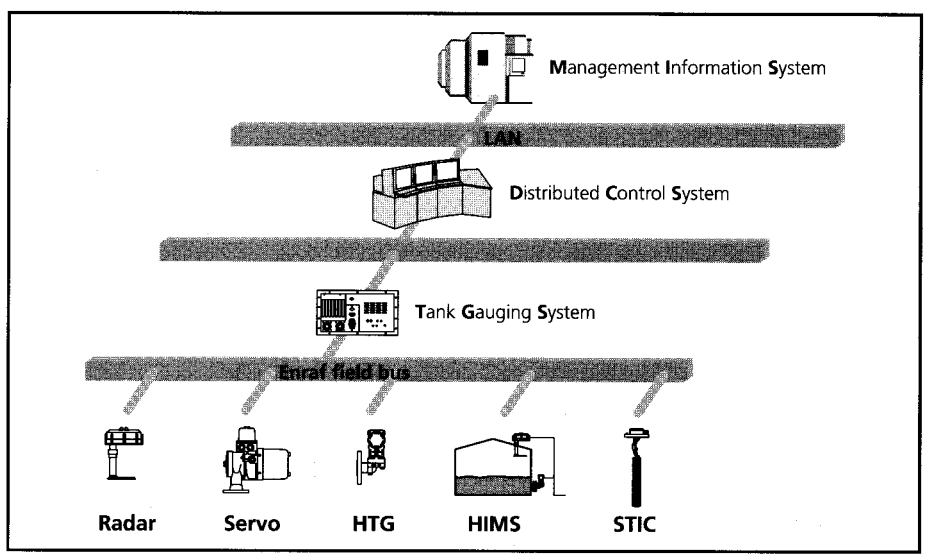


FIG. T-30 Interfacing to distributed control system and management information system. (Source: Enraf.)

Tank inventory management systems, specially developed for tank farms and equipped with suitable host links, will have distinct advantages.

- It frees the host system supplier from needing detailed knowledge of transmitter and gauge specific data handling.
- Maintaining a unique database, with all tank-related parameters in one computer only, is simple and unambiguous.

TABLE T-3 Suitability of the Different Gauging Techniques

	Servo	Radar	HTG
Asphalt	–	++	–
Fuel oil, crude	+	++	+
Black products	+/–	+	+/–
White products	++	+	+
LPG/LNG	++	+/–	—

- Inventory and transfer calculation procedures outside the host system are easier for Weight and Measures authorities.
- Implementation of software required for handling of new or more tank gauges can be restricted to the tank-gauging system. This will improve the reliability and availability of the host system.

Connecting all field instruments via one fieldbus to the supervisory system, DCS or tank-gauging system is advantageous for operations. It simplifies maintenance and service, and allows fast replacement of equipment in case of failure.

Future trends in tank-gauging technology

Combining static and dynamic measuring techniques provides a possibility for continuously monitoring physical stock levels on a real-time basis. By reconciling recorded changes in stock levels against actually metered movements, the system can detect and immediately identify any product losses.

Unexpected product movements can then be signaled to the operator by an alarm. Statistical analysis of static data from the tank-gauging system and dynamic data from flow meters could also be used to improve the accuracy of the tank capacity table. Cross-correlation of gauges versus flow meters could further reduce measurement uncertainties. With high-accuracy tank-gauging instruments combined with powerful computing platforms, automatic reconciliation becomes realistic.

Interfaces to multiple supplier systems, ranging from tank gauging to loading and valve control systems, will be feasible via internationally accepted communication standards.

In summary, a wide range of different tank-gauging instruments is available. The employed techniques are more complementary than competitive as each measuring principle has its own advantages. See Table T-3. Modern servo and radar gauges have improved considerably. They hardly need any maintenance and can provide trouble-free operation if applied correctly. The possibility of mixed installations with servo, radar, HTG, and HIMS provides optimal flexibility and utilizes the capability of each gauging technique.

HTG is to be preferred if mass is the desired measurement for inventory and custody transfer.

The costs of any tank-gauging system are mainly determined by the cost of installation including field cabling. In upgrading projects, costs depend very much on the possibility of retrofitting existing facilities.

Because of worldwide commercial practice, volume measurement will continue to play an important role.

The combination of volume and mass offers great advantages. A globally accepted

measurement standard will probably not be published for several years. Implementation of volume and mass calculations outside the management information, DCS, or host systems remains preferable. Integrity requirements for volume and mass calculations imposed by the Weight and Measurement authorities are easier to fulfill externally and justify the additional hardware.

Standard field buses may play a decisive role in the direct interface between dedicated tank-gauging systems and other systems. However, the quality of the measurements should never be sacrificed for the sake of bus standardization.

Temperature and Pressure Sensors (*see* Measurement)

Thermal Insulation (*see* Commonly Used Specifications, Codes, Standards, and Texts)

Thin-Film Processors (*see* Chillers)

Torque Converters, Measurements, and Meters (*see* Power Transmission)

Towers and Columns

Towers and columns are heat- and mass-transfer devices in which reactions may occur. Reactors frequently are large enough to require the structural design techniques used with towers. The term "reactor tower" might be used to describe a tower that does reactor functions.

The tower may be dealing with fractional distillation or component content change(s) (two substances mutually insoluble, but where one contains a dissolved substance that needs to be transferred to solution in the other). A *scrubber* is the term given to a tower where the solute is transferred from a gas to a liquid phase. In a *stripper* (or *regenerator*) the reverse occurs.

When towers are tall, wind loading factors become severe. Towers a few hundred feet high are not that uncommon now.

Tray-type reactors. Internally, a variety of different tray types may be used. The descriptive terms for these trays include: bubble cap trays, "flexitrays," ballast trays, float tray, sieve trays, "turbogrid," and "kittel" trays. They use a variety of techniques, including sieve slots and holes, as well as caps or fitted mini "skirts," to alter the residence time of the fluid that passes over them, thereby enabling a more complete reaction.

Packed reactors. A packed reactor is more popular with very corrosive applications. A designer needs to allow for good distribution and avoid overly large packing and bed depth. Packing types include various kinds of rings and saddles.

Toxic Substances (*see* Pollutants, Chemical)

Transportation, of Bulk Chemicals, of Large Process Equipment

For regulations and guidelines covered for these items as well as spills during transportation of same, consult the appropriate government protection agency. In the United States, this would be the EPA; in Canada, Environment Canada; in the United Kingdom, DOE. If traveling across borders, one needs to look at all the specifications from different countries and pick the most stringent one to work toward.

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.

Triple Redundancy

Triple redundancy is a term generally associated with aircraft engine control. However, as with other aspects of gas turbine system design—-such as metallurgy, where Rolls Royce RB211 and Trent flight engines lend their metallurgical selections to their land-based counterparts in power generation and mechanical drive service—-the technology is starting to move to "ground level." The governing factor, as always, is economics.

Triple redundancy means a more reliable, available system that is prone to fewer failures. When that translates into money, the more sophisticated technology is adopted. At this point, many power plants in Asia have unused capacity and are not always hurt financially if there is an interruption in availability in one of their power modules. On the other hand, some of them have power purchase agreements (PPAs) that guarantee them income if they run (the YTL and Genting IPP power plants in Malaysia are in this category). They may not be in as bad an income-lost-in-the-event-of-failure situation as some of their mechanical drive counterparts, however. (On critical mechanical drive applications, 24 h of downtime on a critical compressor, pump, or blower could mean $250,000 to $500,000 in lost income.) At this point, triple redundancy technology is more popular with these users, but power operators need to take notice. As other more elementary problems, such as transmission-line losses, are brought under control, they have to look elsewhere for further optimization. Triple redundancy or *triple modular redundancy* (TMR) is an expensive option, so foreknowledge is important.

Software-implemented fault tolerance is the most common TMR technique in use. This method involves three processors that run asynchronously. This guards against transient errors. Each processor waits for the other two to "cast their vote" at certain points in the program cycle (at least once per input/output scan). The processors vote about:

- Input values
- Output values
- Data results
- Condition codes
- Condition interrupts
- Memory locations
- Diagnostics

The processors communicate during the vote so that they have high test coverage. Communication occurs through read-only isolated links.

Choice of TMR Control

TMR is generally selected if:

1. Shutdown/malfunction/loss of availability might endanger operators.
2. Shutdown/malfunction/loss of availability might hurt overall plant economics and cost per running hour.

3. Shutdown/malfunction/loss of availability might violate contracts (such as PPAs)/legislation (such as environmental laws).
4. There are remote control situations.
5. Damage to the overall system may result.

The Effect on Life-Cycle Costs

- Improved safety: A safer operation is a less costly one. Fewer faults/unwanted shutdowns put operational staff under less pressure and make them more inclined to contribute to safety.
- A TMR can keep a system operating even if there are one or more system faults (electronic or field equipment). Backup machinery can be accurately cued and started. The system can be run, albeit imperfectly, until optimized timing for taking the turbomachinery units and/or accessories can be arranged. This cuts down on lost production losses.
- Unscheduled outages are costly. They can occur due to
  - Machinery and hardware failures
  - Control system failures
  - Operator or maintenance personnel errors

  If there is no redundancy (simple or simplex system) or dual redundancy (duplex systems), these types of failure are likely to result in shutdown. TMR eliminates control system-caused shutdowns. It cuts down on the number of operator-caused shutdowns, as much because of the speed of TMR's diagnostics as anything else.
- Maintenance costs per fired hour: With optimized diagnostics, the system is likely to run "unknowingly" with faulty components that would have gone undetected in a simplex or duplex system.
- System component longevity: Smooth (bumpless) synchronization of generators reduces wear factors on generators, couplings, and turbines. Smooth transfer of fuel types in a dual (gas/liquid) or tri (gas/liquid/gas and liquid mixed) fuel system greatly compensates for the temperature bursts that take a severe toll on hot-section component lives.
- Efficiency: Total system thermal efficiency is influenced by many factors including
  - Improved NO_x control (combustion stability)
  - Automatic operation, starting, loading, or synchronizing
  - Integration of control systems of multiple plant units. This is additionally significant in many power development projects where modules or cogeneration or waste-heat recovery schemes are commissioned after the main unit has run for a while (add-ons).

Operation of the TMR

The TMR operates according to a two-out-of-three algorithm that is generally configured for fail-safe operation. If there is one input/output failure, the system continues to operate. If there is another component failure, the system can be configured to continue running (3:2:1 mode) or shut down safely. This meets all the safety codes that are required for plant operation as well as international standards for system management. It gives full fault tolerance between input and output

terminals. PC controllers with Windows software can be used to monitor the overall system, so this high degree of sophistication is quite user-friendly.

Control Features of TMR

For turbines (power generation and mechanical drive applications) and driven equipment, the TMR system provides:

- Automatic startup sequencing
- Starter control
- Load and load-sharing controls (using temperature or speed variables)
- Alarms and shutdowns
- Dual/triple fuel changeovers
- Turning gear controls
- Automatic compressor module washes
- Safety interlock control
- Monitor and diagnostics of condition monitoring systems (CMS)
- Antisurge control systems

For generators, the TMR provides:

- Synchronization to local transmission bus systems, controllers, and exciters
- Control of the main breaker
- Control of safety interlocks
- Monitoring diagnostics of CMS

In summary, TMRs are worth the investment when a system must remain running. They provide availabilities of 99.999 percent. These applications in the power industry will grow in number as Asia's demand growth accelerates. While simplex control may be acceptable as long as a turbine and generator are essentially all the system consists of, the justification for TMR rises with the addition of other modules, other system complexities, or increased demands on availabilities.

Turbines, Gas

Gas Turbine: Basic Description*

The gas turbine is a heat engine, i.e., an engine that converts heat energy into mechanical energy. The heat energy is usually produced by burning a fuel with the oxygen of the air. In that way the engine converts the potential chemical energy of the fuel first to heat energy and then to mechanical energy. However, in a gas turbine, as well as in other types of heat engines, only a part of the released heat energy can be converted into mechanical energy. The remaining heat energy will be transferred to the atmosphere. See Fig. T-31.

The efficiency of the energy conversion tells the portion of the input energy converted into useful energy and is generally designated h. In a gas turbine 25–40

* Source: Alstom.

terminals. PC controllers with Windows software can be used to monitor the overall system, so this high degree of sophistication is quite user-friendly.

Control Features of TMR

For turbines (power generation and mechanical drive applications) and driven equipment, the TMR system provides:

- Automatic startup sequencing
- Starter control
- Load and load-sharing controls (using temperature or speed variables)
- Alarms and shutdowns
- Dual/triple fuel changeovers
- Turning gear controls
- Automatic compressor module washes
- Safety interlock control
- Monitor and diagnostics of condition monitoring systems (CMS)
- Antisurge control systems

For generators, the TMR provides:

- Synchronization to local transmission bus systems, controllers, and exciters
- Control of the main breaker
- Control of safety interlocks
- Monitoring diagnostics of CMS

In summary, TMRs are worth the investment when a system must remain running. They provide availabilities of 99.999 percent. These applications in the power industry will grow in number as Asia's demand growth accelerates. While simplex control may be acceptable as long as a turbine and generator are essentially all the system consists of, the justification for TMR rises with the addition of other modules, other system complexities, or increased demands on availabilities.

Turbines, Gas

Gas Turbine: Basic Description*

The gas turbine is a heat engine, i.e., an engine that converts heat energy into mechanical energy. The heat energy is usually produced by burning a fuel with the oxygen of the air. In that way the engine converts the potential chemical energy of the fuel first to heat energy and then to mechanical energy. However, in a gas turbine, as well as in other types of heat engines, only a part of the released heat energy can be converted into mechanical energy. The remaining heat energy will be transferred to the atmosphere. See Fig. T-31.

The efficiency of the energy conversion tells the portion of the input energy converted into useful energy and is generally designated h. In a gas turbine 25–40

* Source: Alstom.

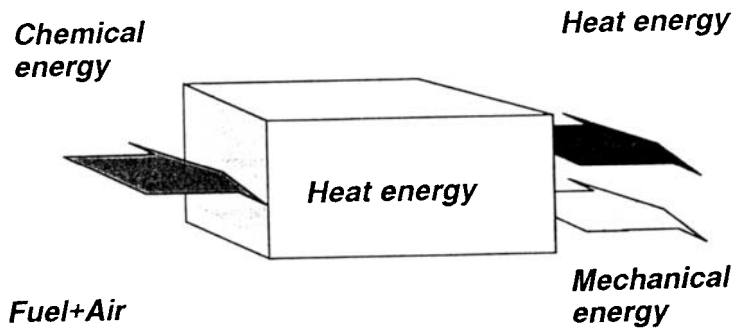


FIG. T-31 Energy exchange for a gas turbine. (Source: Alstom.)

percent of the input energy is transformed into mechanical energy. The remaining 60–75 percent will be transferred to the atmosphere in the form of waste heat (exhaust losses). The efficiency is consequently 15–40 percent. Where a part of the waste heat can be recovered, e.g., in a waste heat recovery boiler, the efficiency increases correspondingly.

$$\text{Efficiency} = \eta = \frac{\text{output energy}}{\text{input energy}}$$

Operating cycle main parts

In a gas turbine the operating medium is air and gas, and the flow runs through the cycle COMPRESSION—HEATING—EXPANSION.

In an *open gas turbine cycle*, ambient air is sucked in, compressed in a COMPRESSOR, heated in a COMBUSTION CHAMBER by injection and burning of a fuel and then expanded through a TURBINE back to the atmosphere. The operating medium of an open gas turbine cycle consequently is air and a mixture of air and combustion gases.

In a *closed gas turbine cycle* an enclosed gas, which cannot be air, runs through the same phases as in the open cycle, but the heating takes place in a heat exchanger and the gas expanded through the turbine must be cooled before it is led back to the compressor. See Fig. T-32.

In practice the open gas turbine cycle is completely dominating and the further description is fully concentrated on the open gas turbine cycle.

Function principle

As mentioned in the previous part, the gas turbine consists of three main parts: compressor, combustion chamber, and turbine. How heat energy, by the operating medium flowing through these main parts, is converted into mechanical energy can be explained by means of the simple model shown in Fig. T-33.

A tube is in either end equipped with a simple fan. One of the fans is named compressor and the other fan is named turbine. An external power source, or starter, is through a coupling connected to the compressor.

Through the tube an airflow is created that will speed up the turbine. Energy is supplied to the compressor and is transferred to the airflow. From the airflow energy flows to the turbine, which through its rotation gives off a mechanical output. The

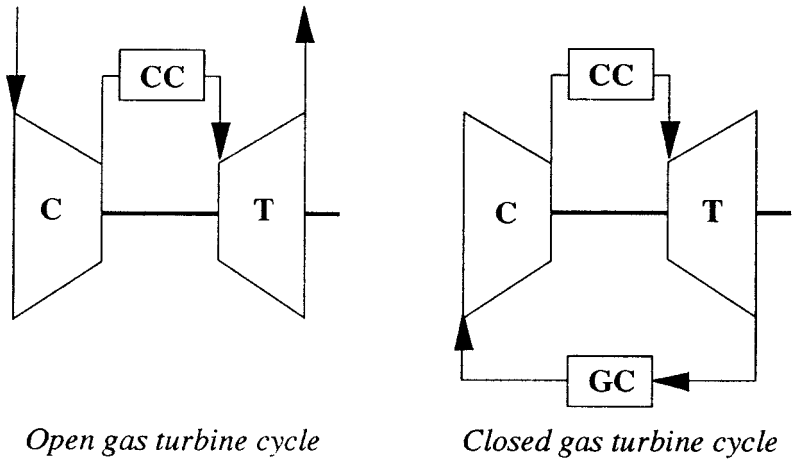


FIG. T-32 Open and closed cycles for a gas turbine. CC = combustion chamber, C = compressor, T = turbine, GC = gas cooler. (Source: Alstom.)

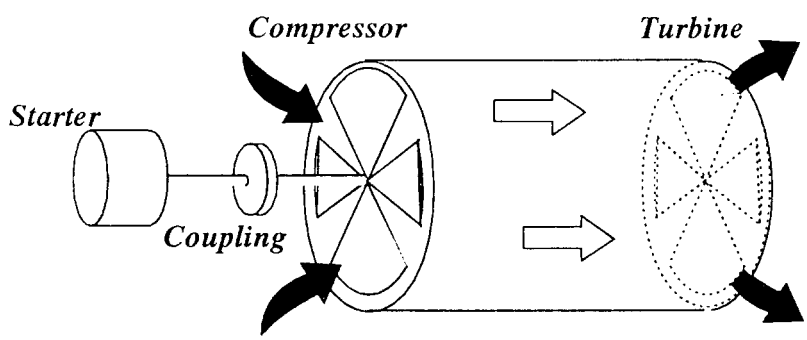


FIG. T-33 The compressor is "speeded up" by the starter. (Source: Alstom.)

energy flow can be noticed as compressor speed, increase of airflow velocity and pressure (by virtue of pressure increase as well as temperature increase), and turbine speed. If the process goes on without losses (which in practice is impossible but temporarily accepted to simplify the understanding), the turbine energy output is equal to the energy sacrificed to drive the compressor.

The airflow is heated

The heating means that the air temperature increases. Since the air pressure inside the tube is created by the compressor, the heating of the air does not result in further increased air pressure. Instead the air volume is increased. Increased air volume results in increased air velocity through the turbine. A larger amount of energy is transferred to the turbine, which then can give off a larger mechanical output. If the process goes on without losses, the turbine mechanical energy output is equal to the sum of the mechanical energy supplied to the compressor and the heat energy supplied to the airflow. See Fig. T-34.

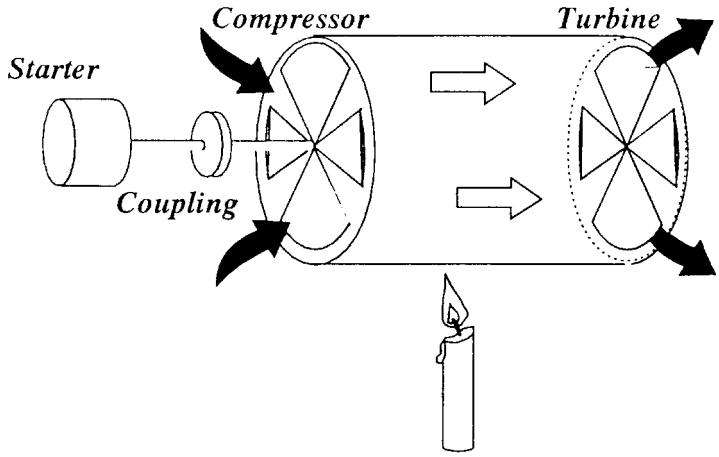


FIG. T-34 Airflow is heated from fuel combustion. (Source: Alstom.)

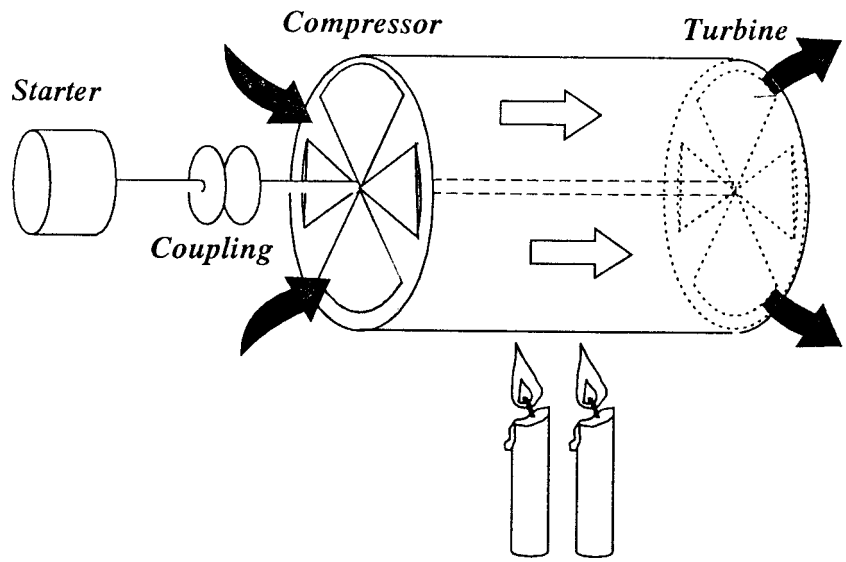


FIG. T-35 Starter is disconnected when gas turbine reaches self-sustaining speed. (Source: Alstom.)

Self-sustaining speed

Increased heat supply means that the turbine gives sufficient mechanical output to drive the compressor. If the compressor and turbine are mounted to a common shaft, the starter can be disconnected and self-sustaining condition is reached. The starter has been necessary to create the airflow through the tube. The airflow forces the process to continue by virtue of its momentum. Heating stationary air inside the tube would only have meant temperature increase and air expansion backward through the compressor as well as forward through the turbine. See Fig. T-35.

At self-sustaining condition the mechanical output extracted from the turbine is just enough to drive the compressor. The whole amount of energy supplied by heating is waste energy. In reality these losses consist of exhaust losses, losses due to turbulence, and radiation losses. For thermodynamic reasons the temperature of the exhaust gas must be higher than that of the sucked-in air and

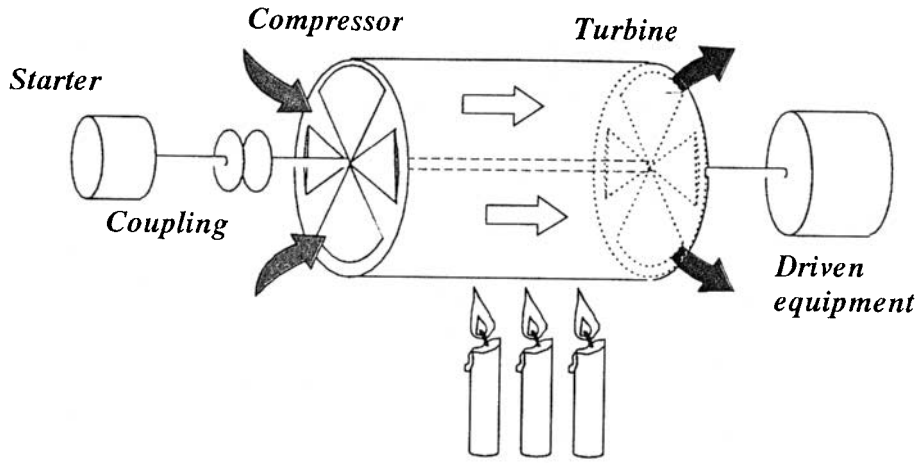


FIG. T-36 For useful work output, gas turbine is driven past self-sustaining speed. (Source: Alstom.)

that means losses. Further the exhaust gas must leave the turbine at a certain velocity.

Power generation

To get a useful mechanical output from the turbine the heat supply must be further increased to speed up the engine above self-sustaining speed. See Fig. T-36.

To make the gas turbine as efficient as possible, by converting heat energy into mechanical energy, its design must be much more complicated than that of the described simple model. However, the main features with cylindrical casings containing compressor, combustion chamber, and turbine, so that the air/gas flow is moving straight through the engine, are typical for many gas turbines. See Fig. T-37.

The compressor

The compressor is in practice not a simple fan, but a far more sophisticated construction to continuously compress an airflow to desired pressure. One of two basic types of compressors, one giving a radial flow and the other an axial flow, is normally used in a gas turbine. The axial flow compressor is easier to design for high-pressure ratios, is more efficient, and is thus common in high-performance units. Only the axial flow compressor is dealt with in this primer.

Axial flow compressor design

An axial flow compressor consists of one or more rotor assemblies that carry blades of airfoil section and are mounted between bearings in the casing. In the casing are mounted the stator vanes, which also are of airfoil section. The compressor is a multistage unit as the pressure increase by each stage is small (pressure ratio 1.15–1.25/compressor stage consists of a row of rotating blades followed by a row of stator vanes. When needed, an additional row of stator vanes, known as inlet guide vanes, is used to guide the air on to the first row of rotor blades. From the front to the rear of the compressor, i.e., from the low to the high pressure end, there is a gradual reduction of the airflow annular area. This is necessary to maintain

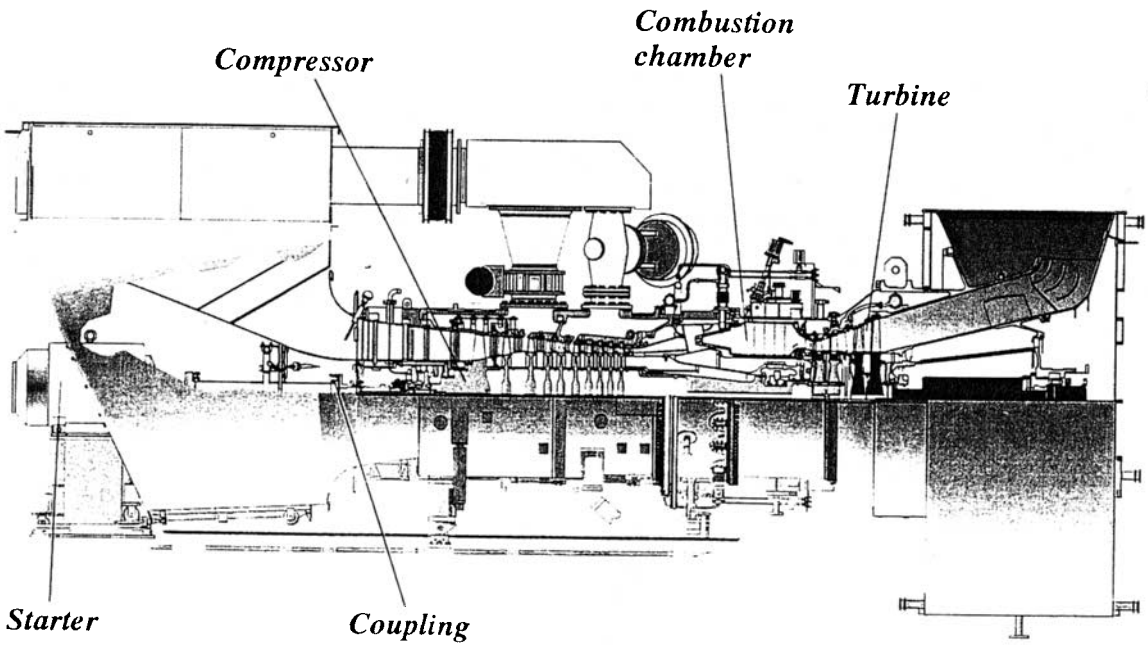


FIG. T-37 Section through a gas turbine. (Source: Alstom.)

the axial velocity of the air constant as the volume decreases during the compression. To prevent air leakage there are sealings between the stages and at the inlet and outlet ends of the compressor. See Fig. T-38.

Function principle

During operation the compressor is turned at high speed by the turbine. Air is continuously induced into the compressor, where it is accelerated by the rotating blades and swept rearward. In the subsequent stator vane passages, shaped as diffusers, the air velocity is decreased and thus the air pressure is increased. A similar process takes place in the rotor blade passages. The stator vanes also serve to correct the deflection given to the air by the rotor blades and to present the air at a correct angle to the next stage of rotor blades. The last stator vane row usually acts as "air straightener" so that the air enters the combustion chambers at a fairly uniform axial velocity.

Compressor stall and surging

The airfoil sections, the blade angles, and the reduction of the annular area are designed to give best performance at full load (full speed), i.e., for a certain relationship between airflow and blade velocity and for a certain compression ratio. If the airflow velocity is too low in relation to the blade velocity, which occurs if the compressor rotor accelerates too quickly or if the air intake filter is clogged, the airflow will break away from the blades. That phenomenon is known as stall when only a few stages are concerned and is known as surging when the complete airflow through the compressor is broken down. Stall or surging is a serious problem because the blading then is exposed to oscillating forces creating unwanted stresses. The compressor is designed to operate below its surge limit, but if the airfoil sections are spoiled by excessive fouling the surge limit is lowered so that stall or surging

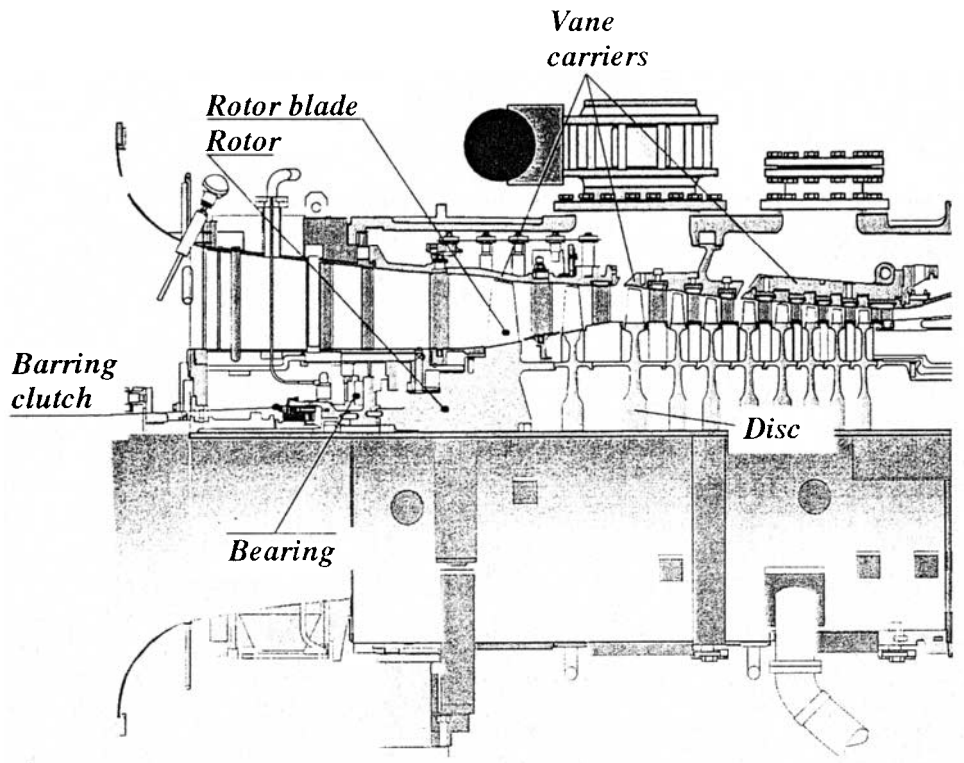


FIG. T-38 Gas turbine compressor. (Source: Alstom.)

can occur even at normal operating conditions. Thus, regular compressor cleaning is a necessity.

Airflow control

At low compressor speeds, i.e., during start or low load, the compressor gives a lower compression ratio and that calls for a smaller degree of annular duct convergence. That means that at lower speeds the front stages of the compressor tend to be stalled and the rear stages tend to be choked. This problem increases with the number of stages and the pressure ratio but can be managed by using bleed-off valves, variable guide vanes, or twin-spool compressors (each of the two compressor parts driven by its own turbine). All three means are used when needed. Simplified, the bleed-off valves cut off a part of the front stages by bleeding air from an intermediate stage, the variable guide vanes decrease the airflow to the rear stages by throttling the first stage(s), and the twin-spool compressor allows the relationship between the speed, and thus the capacity, of the two compressor parts to alter.

The combustion chamber

In the combustion chamber, the fuel, continuously injected through the fuel burners, is burnt with air, supplied by the compressor, and heat is released in such a manner that the gas is expanded and accelerated to give a smooth stream of uniformly heated gas at all conditions required by the turbine. This must be accomplished with the minimum loss in pressure and with the maximum heat release for the

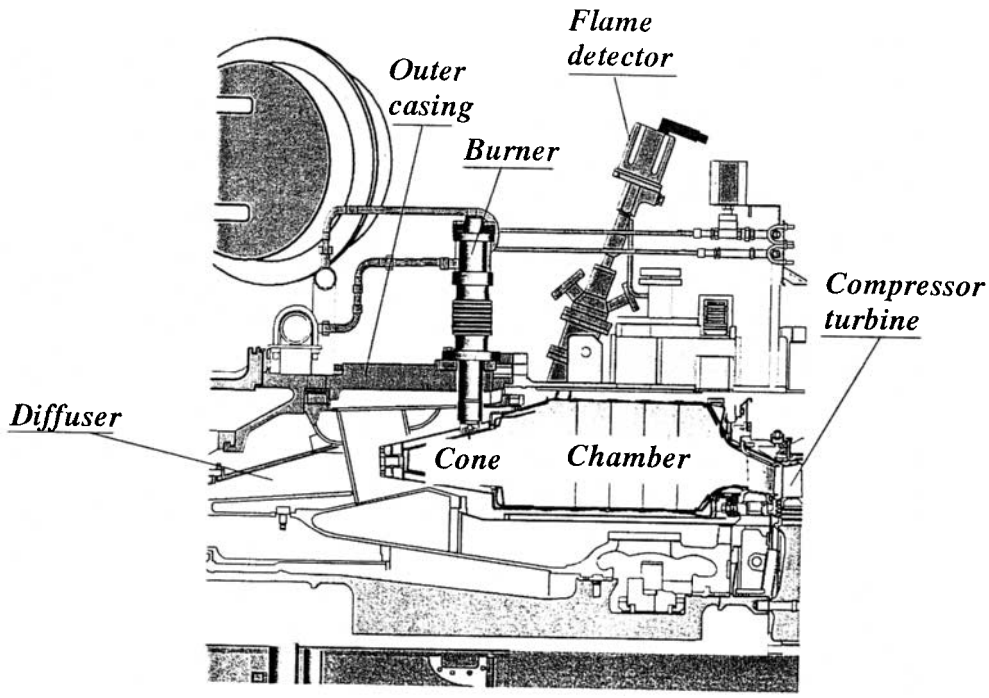


FIG. T-39 Combustion section, gas turbine. (Source: Alstom.)

limited space available. Efficient combustion is necessary to obtain high thermal efficiency and to minimize the exhaust gas emission.

Design

The combustion chambers can be designed in different ways. The following description refers to the canannular type. A number of flame tubes are fitted inside a common annular casing. The burners are fitted in the center of the forward end, known as the flame head, of the flame tubes. The compressor outlet is via a diffuser, in which the airstream is decelerated and the static pressure raised, connected to the forward end of the annular casing and each flame tube is via a gas collector connected to a section of the turbine inlet. During start the combustion is initiated by means of electrical igniter plugs fitted to one or more of the flame tubes. The flame is then spread to the other flame tubes through crossover tubes. See Fig. T-39.

Function principle

The air leaves the compressor outlet at a velocity in the region of 100 m/s, but the speed of burning fuel at normal mixture ratios is only a few meters per second. Thus, not to blow out the flame, the airflow must be decelerated. A region of low axial velocity has to be created inside the flame tube so that the flame will remain burning throughout the engine operating conditions. To obtain efficient combustion the flame temperature must be about 1400–2000°C. Since no material known today can stand such a temperature, excess air must be supplied to cool the flame tube walls and to dilute the hot gases to a temperature that the material of the turbine parts can stand. The combustion takes place in the combustion zone inside the flame tube. To that zone fuel is injected through a nozzle and air is induced through a

swirl surrounding the fuel nozzle, through the flame head slots, and through radial holes in the flame tube wall.

The air supplied creates a region of recirculating gas that takes the form of a torodial vortex, similar to a smoke ring, to stabilize and anchor the flame in the center of the combustion zone. The recirculating hot gases also greatly assist in atomizing up the fuel and mixing it with the incoming air. At full load only about 1/4 of the total airflow from the compressor is supplied to the combustion zone. That part is sufficient to obtain complete combustion. The remaining airflow, the excess air, is used to cool the flame tube walls and to dilute the hot gases. The cooling air is supplied in such a manner that a comparatively cool airstream is created nearest the flame tube wall. The dilution air is supplied through large holes downstream of the flame tube.

The turbine

The turbine provides the power for driving the compressor(s) and the power to give a useful mechanical output. That is done by extracting energy from the hot gases released from the combustion chambers and expanding them to a lower pressure and temperature. High stresses are involved in this process. Since the turbine operates at high speed it is exposed to large centrifugal forces and the operating medium. The gas enters the turbine at a very high temperature.

Two basic types of turbines can be used, the radial flow turbine and the axial flow turbine. In the radial flow turbine the gas enters the turbine in the radial direction and in the axial flow turbine the gas flow passes the turbine in the axial direction. Except from very small units the axial flow turbine is totally dominating and the following description is completely concentrated on that type.

Axial flow

The turbine normally consists of several stages, each stage combined with a row of stationary guide vanes or nozzles followed by a row of moving blades or buckets. The guide vanes are mounted to the turbine casing and the buckets are fitted to turbine discs, mostly by means of fir-tree roots. See Fig. T-40.

The discs are mounted to one or more shafts depending on the configuration. To prevent gas leakage there are sealings between the stages and there are also sealings to prevent leakage of hot gases toward the shafts and bearings.

Those sealings are often supplied with sealing air, bled off from suitable compressor stages, and this air is led off along the turbine discs to cool them and prevent heat transfer to shaft and bearings. See Fig. T-41.

Function principle

In the convergent passages between the guide vanes of the airfoil section, the hot gas is expanded. Pressure energy is converted into kinetic energy and the gas is accelerated. At the same time the gas is given a spin or swirl in the direction of rotation of the turbine buckets. By the buckets the gas is forced to deflect and, since the passages are convergent, the gas is further expanded. On impact with the buckets and during the subsequent reaction through the passages, energy is absorbed, causing the turbine to rotate and provide the power for driving the turbine shaft. By the guide vanes of the next stages the gas then is further expanded and directed to the following row of buckets. See Fig. T-42.

The number of stages depends on the number of shafts and on the pressure ratio. Several stages were required to compress the air, but since the gas expansion is

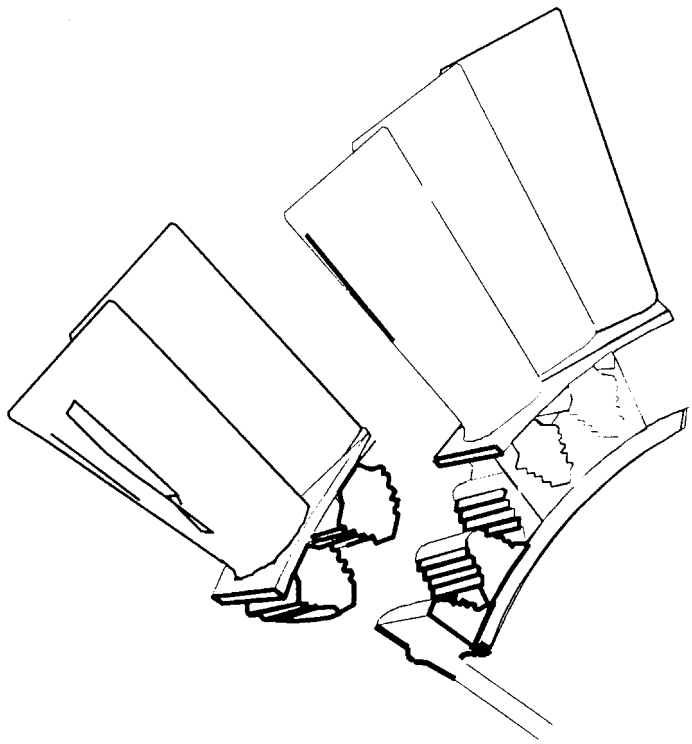


FIG. T-40 Fir-tree roots. (Source: Alstom.)

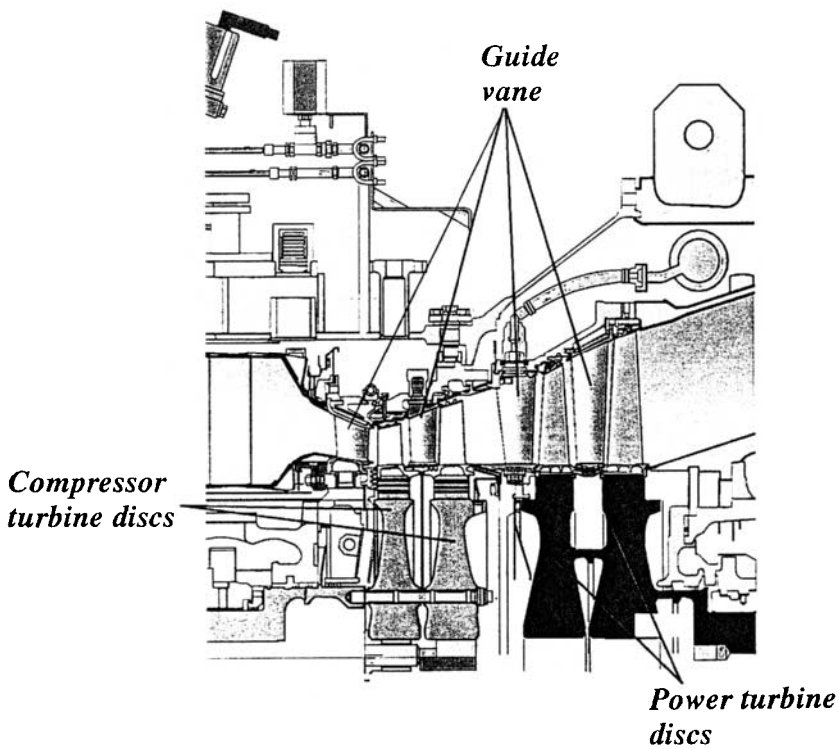


FIG. T-41 Turbine section, gas turbine. (Source: Alstom.)

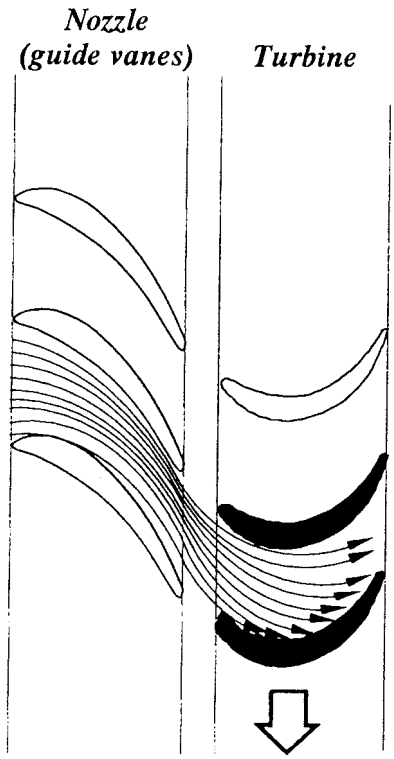


FIG. T-42 Airflow through a turbine stage. (Source: Alstom.)

a spontaneous process, in contrast to the forced compression, fewer stages are required to expand the gas back to atmospheric pressure.

If the energy is absorbed efficiently the swirl will be removed from the gas stream leaving the last stage, so that the flow at exit from the turbine will be substantially straightened out to give an axial flow into the exhaust system. To increase the available pressure ratio across the turbine the annular exhaust duct connected to the turbine is shaped as a diffuser. See Fig. T-43.

Gas turbine configurations

The gas turbine main parts can be arranged in different ways and be combined with other components. Some examples are given in Figs. T-44 through T-46.

The single shaft gas turbine is the simplest, but it is only suitable for fixed-speed applications. Due to the constant compressor speed, the efficiency drops sharply at lower loading rates.

The free power turbine configuration makes it possible to vary the output speed within a wide range. In fixed-speed applications the performance at low loading rates is improved compared to a single shaft gas turbine.

The free power turbine combined with a twin-spool gas generator is a more flexible engine. The twin-spool gas generator allows high-pressure ratios and still gives good low loading rate performance.

Air/gas conditions through a gas turbine

See Fig. T-47.

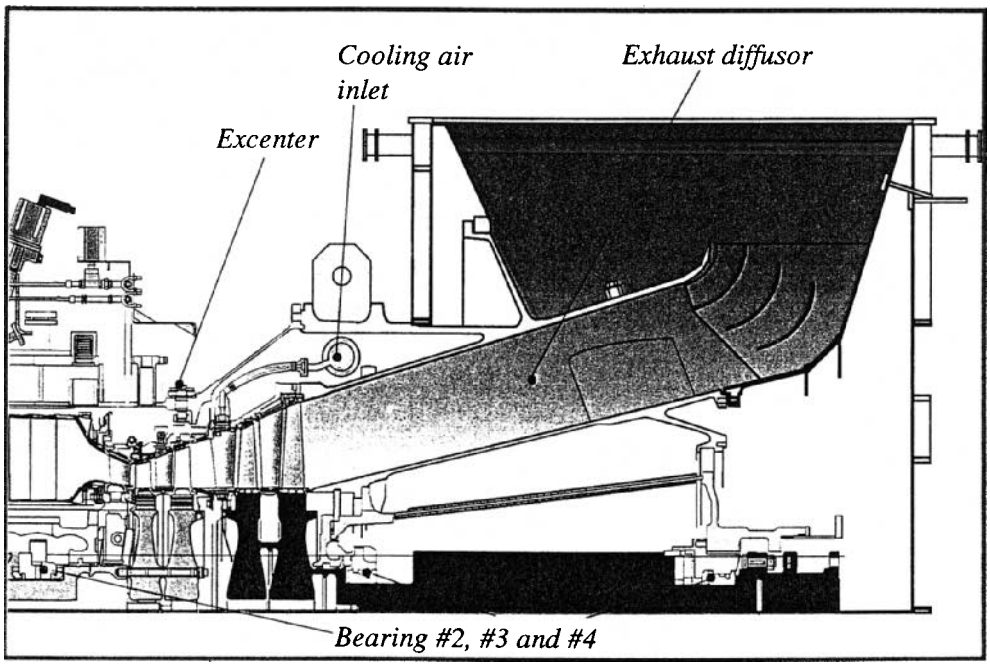


FIG. T-43 Gas turbine exhaust diffuser. (Source: Alstom.)

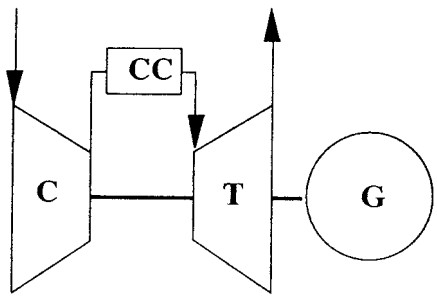


FIG. T-44 Single-shaft gas turbine configuration. C = compressor, CC = combustion chamber, T = turbine, CT = compressor turbine, PT = power turbine, G = generator. (Source: Alstom.)

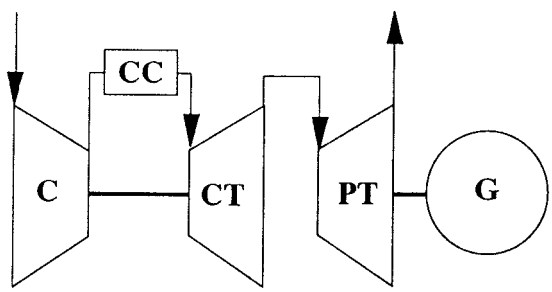


FIG. T-45 Free power turbine driving driven equipment. C = compressor, CC = combustion chamber, T = turbine, CT = compressor turbine, PT = power turbine, G = generator. (Source: Alstom.)

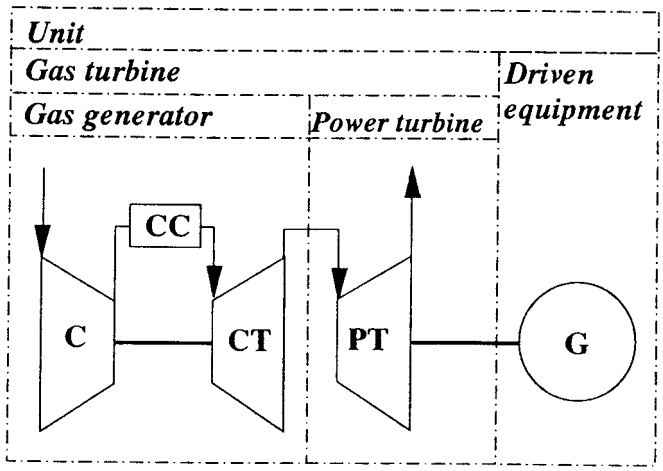


FIG. T-46 Power train terminology. C = compressor, CC = combustion chamber, T = turbine, CT = compressor turbine, PT = power turbine, G = generator. (Source: Alstom.)

Gas turbine performance

The factors affecting the gas turbine thermal efficiency and specific output (specific output = output per kg/s of gas flowing through the engine) can be divided into two groups—thermodynamic factors and the influence of ambient conditions.

Thermodynamic factors

These factors, which are component efficiencies, compressor pressure ratio, and turbine inlet temperature, are mainly determined by the design of the engine.

The component efficiency depends on the type of component used and its design. Improved component efficiencies give higher unit thermal efficiency and higher output. Typical figures are: axial flow compressor efficiency ηc = 0.87–0.90, combustion chamber efficiency ηcc = 0.96–0.98, and axial flow turbine efficiency ηt = 0.85–0.88.

The compressor pressure ratio and the turbine inlet temperature have a great influence on the unit thermal efficiency and the specific output. Further, those two factors interact in such a way that a certain pressure ratio is optimum for a certain turbine inlet temperature. See Figs. T-48 and T-49.

The allowable turbine inlet temperature depends on the material used in the "hot section" of the engine and the desired lifetime. Sometimes more than one temperature limit is stated given different performance and different lifetime. Increased gas temperature gives shorter lifetime, a 30°C increase can shorten the lifetime to one-third.

The hot section materials and the desired hot section lifetime determine the allowable gas temperature, which in turn, for an optimized unit, calls for a certain pressure ratio. The combination of gas temperature and pressure ratio gives a specific output and thermal efficiency, which also are influenced by the component efficiencies.

Ambient air pressure P_0

The nominal ambient air pressure, to which the gas turbine nominal output is related, is 1.013 bar, and the actual air pressure normally varies within 1.013 ± 0.05 bar. As the pressure affects the air density, and thus the mass flow through

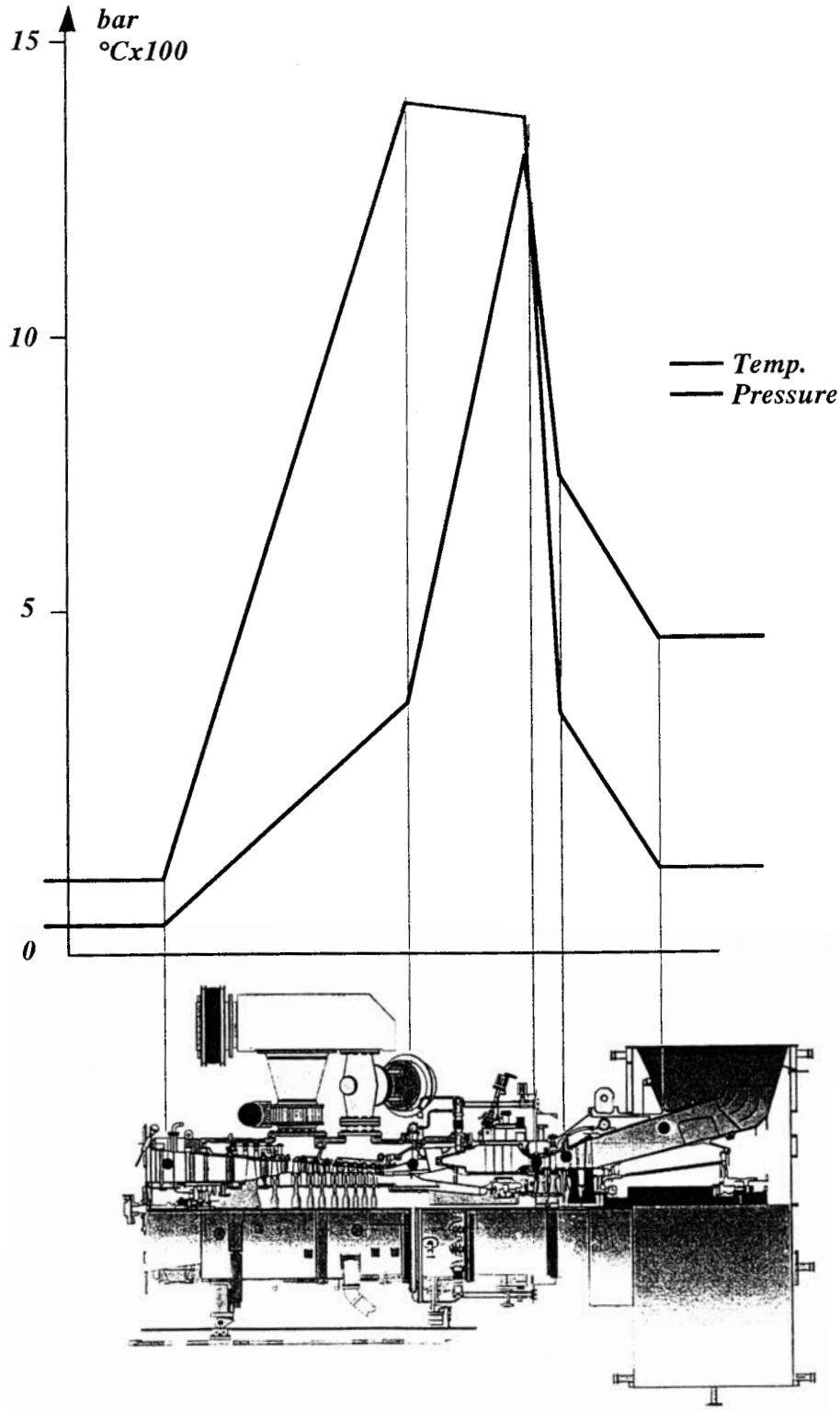


FIG. T-47 Air/gas conditions through a gas turbine. (Source: Alstom.)

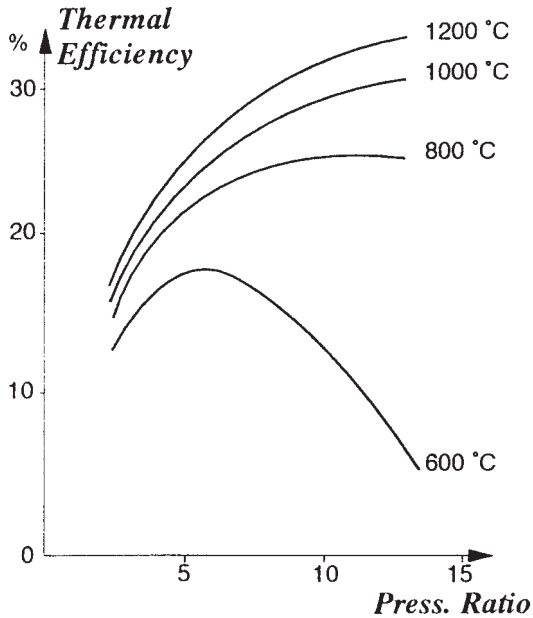


FIG. T-48 Gas turbine performance curves. (Source: Alstom.)

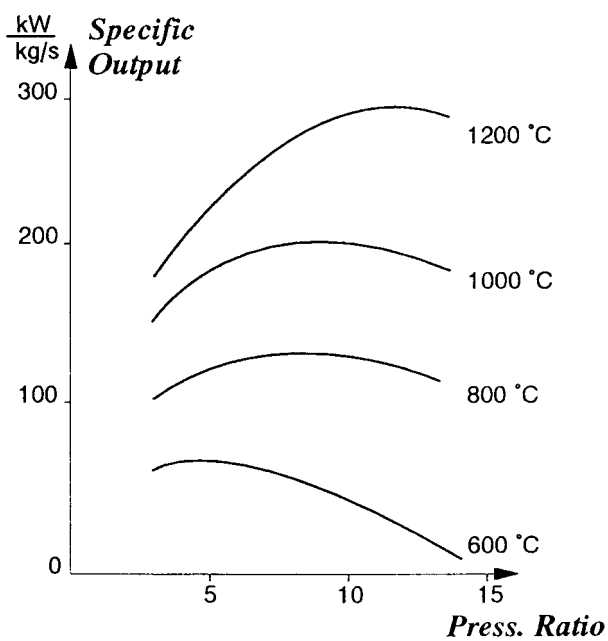


FIG. T-49 Gas turbine performance curves. (Source: Alstom.)

the engine, it also affects the actual output from the engine according to the following formula.

$$P_{nom} = P_{act} \cdot \frac{p_{nom}}{p_{act}}$$

P_{nom} = nominal output
P_{act} = actual output
p_{nom} = nominal air pressure
p_{act} = actual air pressure

At fixed rating (constant gas temperature), increased ambient air pressure gives:

- Increased output
- Unchanged unit efficiency
- Unchanged gas generator speed

At fixed output, increased air pressure gives:

- Decreased gas temperature
- Decreased unit efficiency
- Decreased gas generator speed

Ambient air temperature T_0

The gas turbine performance is normally related to +15°C, but the available output is influenced greatly by the ambient air temperature.

At fixed rating (constant gas temperature), increased ambient air temperature gives:

- Decreased output
- Decreased unit efficiency
- Decreased gas generator LP rotor speed
- Increased gas generator HP rotor speed

At fixed output, increased ambient air temperature gives:

- Increased gas temperature
- Decreased unit efficiency
- Increased gas generator speed

Ambient air relative humidity

The gas turbine nominal performance is related to 60 percent relative humidity in the ambient air. The gas turbine performance is influenced by the humidity, but the influence is of importance only at high ambient air temperatures.

At fixed rating (constant gas temperature), increased relative humidity gives:

- Increased output
- Unchanged unit efficiency
- Increased gas generator speed

At fixed output, increased relative humidity gives:

- Decreased gas temperature
- Unchanged unit efficiency
- Increased gas generator speed

Operating condition

An example of a gas turbine performance diagram is given in Fig. T-50. The diagram shows the generator electrical output versus ambient air temperature with the

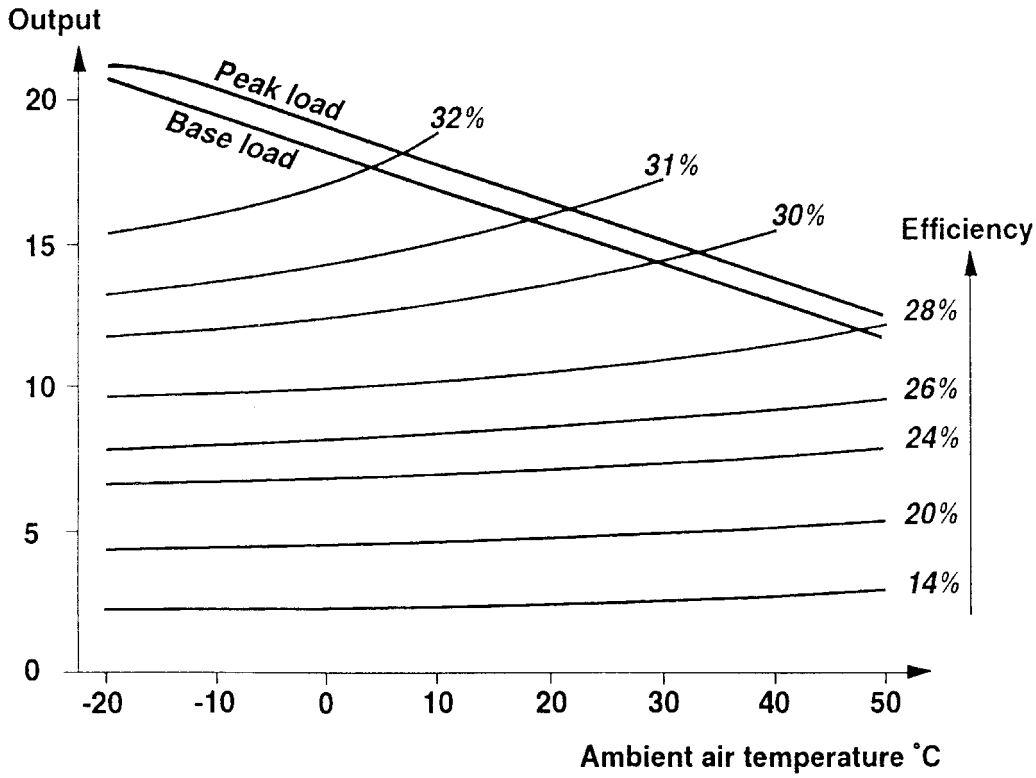


FIG. T-50 The effect of overheat and temperature on gas turbine output. (Source: Alstom.)

thermal efficiency as parameter. At low ambient air temperature the output is limited by the gas generator speed, and at higher temperatures it is limited by the turbine inlet temperature. The shown output and efficiency are of course affected by the ambient air pressure and humidity as described previously.

As can be read in the performance diagram the unit efficiency is influenced by the loading rate. Increased load at constant ambient air temperature gives higher efficiency by virtue of the increased compressor delivery pressure and the increased turbine inlet temperature. Two load limits, base load and peak load, are given. Operation above base load gives shorter service intervals due to the increased stresses and according to the ambient air pressure formulas. The performance curve is normally a nominal curve and the true curve can deviate more or less depending on individual engine qualities. During commissioning, the engine, if deviating from nominal performance, can be adjusted either to nominal turbine inlet temperature or to nominal output.

To check the condition of a unit, e.g., if the compressor has to be cleaned, an output check can be carried out. Such a check means that the actual output is measured, corrected due to deviating ambient air conditions, and then compared to a reference value. Allowable output loss is stated in a *compressor cleaning instruction*. Other reasons for output loss are clogged air intake filters, worn down or damaged turbine buckets, air/gas leakage, etc.

Service intervals are related to operating time and number of starts. Since the stresses to which the gas turbine is exposed vary with the load rating and if fast or normal start is chosen, the number of operating hours and starts are given in

the form of equivalent operating hours and equivalent starts according to the following.

$H_0 = C_f x C_x x H + 5S_0$
$S_0 = C_y x S$
H_0 = Equivalent operating hours
C_f = Fuel factor
- 1 for gas fuel acc. to BA-241-6E
- 1.2 for type 1 fuel acc. to AA-243-9E
- 1.5 for type 2 fuel acc. to AA-243-9E
- 2.0 for crude oil and sour gas

C_x = Stress factor
Exponential factor 0.9–4.0 depending on load rate. The factor is automatically calculated and updated by the unit control equipment.
H = Operating hours
S_0 = Equivalent starts
C_y = Start factor
- 1 for a normal start
- 5 for a fast start

S = No. of starts

Reference and Additional Reading

1. Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Gas Turbine: Cogeneration; Combined Cycle Applications (*see* Cogeneration)

Gas Turbine Compressor/Compressor Washing*

The earliest gas turbine cleaning was by crank soak washing and/or by injecting solid compounds such as nutshells or rice husks at full speed with the unit on line. This method of online cleaning by soft erosion has mainly been replaced by wet cleaning since the introduction of coated compressor blades for pitting corrosion protection. Further, unburnt solid cleaning compounds and ashes may also cause blockage of sophisticated turbine blade cooling systems if ingressed into the gas turbine air cooling stream. At the beginning of the introduction of compressor wet cleaning in the 1980s, time intervals between online washing and the combination with offline washing had first to be established. Further, there was also a belief among many users that online washing could replace offline washing. It has been said that an airflow reduced by 5 percent due to fouled compressor blades will reduce output by 13 percent and increase heat rate by 5.5 percent. With contemporary large-scale use of gas turbines in combined cycle base load application as well as with their increase in nominal output, gas turbine compressor washing has gained more attention.

Combination and frequency of online and offline washing

A first long-term test on the combined effect of online and offline washing at various washing intervals was performed over 4000 operating hours (op.h.) in the mid-

* Source: Adapted from extracts from Stalder, "Gas Turbine Compressor Washing State of the Art—Field Experiences," ASME paper 98-GT-420.

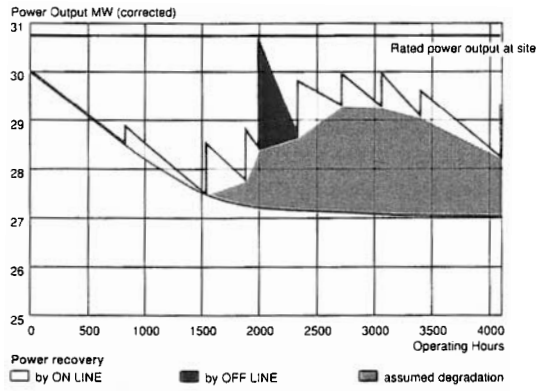


FIG. T-51 Typical effect of online and offline compressor wet cleaning. (Source: Stalder.)

1980s at the Energieproduktie-bedrijf Utrecht (UNA) PEGUS 100 MW combined cycle plant in the Netherlands (two gas turbines of 30.7 MW site output and one 38.6 MW steam turbine). The plant is situated beside the Merwedekanaal on the southwestern outskirts of Utrecht, some 60 km from the sea. A very busy motorway crosses over the canal near the plant and local industries include chemicals and food processing. Together all these various activities give rise to dust, salts, and fine aerosols in the air. The gas turbines ran on natural gas (no fouling of the hot section), and all power measurements were made with the gas turbine running in temperature control mode at base load hot gas inlet temperature. All results, see Fig. T-51, have been corrected to new and clean guaranteed conditions.

The following was established during the test:

- Without cleaning, the power output degradation tends to stabilize itself with increasing operating hours. It was confirmed in the unit tested that the degradation of output was stabilizing at 90 percent base load (ref. new and clean).
- Power recovery after offline washing (soak and rinse procedure) is significantly higher than after an online washing.
- Online washings were performed at time intervals in the range between 700, 350, and 120 op.h. It showed that plant performance is significantly higher at shorter online washing intervals, thus preventing incremental power degradation.
- The combination of both washing methods is the most effective and economical.
- Based on the evaluation of the above performed measurements and extrapolated to 8000 op.h., it was estimated that improved performance equivalent to approx. US$450,000 per year can be achieved with the combination of both online and offline compressor washing methods on one 30-MW gas turbine; the operating and maintenance cost of the systems are not considered in the above figure.

Improvements with shorter online washing intervals

In the spring of 1990, UNA and Turbotect decided jointly to conduct a second long-term field test on a 66-MW gas turbine operating in the Lage Weide 5 combined cycle plant located on the same site. The tests were conducted over 18 months under the combined online and offline wet cleaning regime. During the entire test period the gas turbine unit operated for a total of 8089 h. An outage for a major overhaul (at 26,408 op.h) took place after 3915 op.h since beginning the test. Thus, the tests

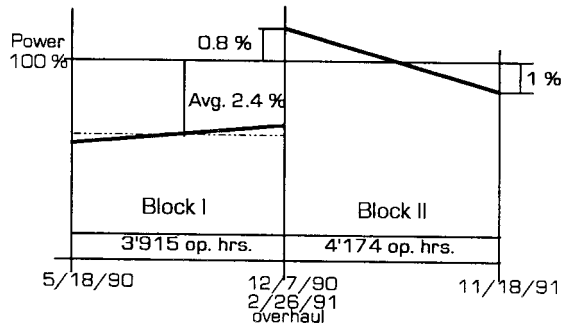


FIG. T-52 Summary of the pattern of the corrected power output for the complete test period. (Source: Stalder.)

aimed to give some comparative indications of the effectiveness of more frequent online washing as applied to a new machine, and to one that operated for several years, as well as on the plant's performance. The gas turbine ran also on natural gas. The air inlet filtration system consisted of a weather louvre, a first-stage coarse filter, and a second-stage fine filter.

Results of the improved compressor cleaning regime. Online compressor washes were performed on average every four days at base load with the gas turbine on temperature control mode. Gas turbine performance was measured before and after each wash.

Observations have shown a sustained high output level close to the nominal guaranteed rating, despite difficult atmospheric conditions.

In the first evaluation block (see Fig. T-52), the gas turbine plant was operated at a load factor of 97.6 percent, or 2.4 percent below the original guaranteed site power output at new and clean conditions. During this period 38 compressor online washes were performed, averaging one every four days. In addition, three offline washes were performed by taking the opportunity when the gas turbine plant was shut down for a few days, this, respectively, after intervals of 760, 2435, and 605 op.h. The average power output increase after an offline wash was approximately 1800 kW. The trend analysis of the performance tests made in this period is nearly horizontal, showing that aging due to mechanical wear and tear of the gas turbine had already stabilized.

In the second evaluation block (see also Fig. T-52), the gas turbine started in practically new and clean condition as the result of some work during the major overhaul. The corrected results of the compressor wet cleaning regime in the second evaluation block show that the gas turbine plant operated for 4174 h at a load factor of 100.16 percent or 0.16 percent above the original guaranteed site power output at new and clean conditions. At the end of this period the number of operating hours of the gas turbine was 30,725. During this second period, 45 compressor online washes were performed, also on the average of one every four days. In addition, two offline washes were performed, one after 1143 and the second 1381 op.h later. The average power output increase after each offline wash in the second period was approximately 1 MW.

Discussion of the results for the improved washing regime

- Out of the 83 online washes made during the total testing period covering 8089 op.h, 87 percent or 72 online washes have demonstrated a positive power recovery with the unit in operation at full load.

- 712 kW was the average power output recovery measured after an online wash. This relative small amount represents approximately 1 percent of the nominal gas turbine power output.
- No secured results on efficiency improvements could be demonstrated because of incomplete data over the testing period with regard to gas analysis and densities to determine the lower heating value, the latter being necessary to have accurate corrected efficiencies and turbine inlet temperature by heat balance calculations.
- Power recovery, due to offline cleaning, is not as significant if the unit's performance is close to nominal guaranteed values.
- Online and offline cleaning are complementary.
- Shutdowns and startups can positively affect compressor fouling by spalling off deposits. The deposits may soak humidity during standstill and the swelled-up material will partly spall off as the shaft is accelerated during startup of the gas turbine.
- The test program confirmed that frequent online cleaning extends the time interval between offline cleaning operations. Thus it is a real benefit to the operator, because the scheduled downtime allowed for maintenance can be reduced if the frequency of offline cleaning with its associated cooling downtime can be reduced. The availability and performance, as well as the overall profitability of the plant, will be improved.
- The results demonstrated that a combined online and offline washing regime can effectively be applied to a new or to an old engine.
- The unit performance measured in the second evaluation period with the regime of washing shows that the power output trend was most probably following the aging of the unit.
- Also worth noting is the impact of the major overhaul on the unit performance. The work involved in readjusting shaft alignment and clearances in the hot section path, etc., are likely to improve output and efficiency of the turbogroup and can offset the cost for the overhaul.

Considerations on the economical profitability. The average power degradation over 4000 op.h at the above tested plant and when compressor cleaning is not performed is up to 10 percent. By using similar criterias as presented by Diakunchak (1991), the plant production profitability during the regime of online and offline compressor wet cleaning was improved by US$1,175,000 over 8000 op.h, representing a very substantial additional profit. An amount of approximately US$20,000 was spent for the chemical cleaner consumed during the program, representing a very marginal cost as compared to the improved profitability. Further, and without a regime of online and offline compressor wet cleaning, there will be an additional loss of profitability due to reduced steam production as a result of compressor fouling in a combined cycle application. The reduction in mass flow has a greater effect on the steam production than the increase in exhaust temperature due to compressor fouling.

Compressor fouling phenomena

The cause of fouling and fouling rates of axial gas turbine compressors is a combination of various factors that can be divided into the following categories:

- Gas turbine design parameters
- Site location and surrounding environment
- Plant design and layout
- Atmospheric parameters
- Plant maintenance

Gas turbine compressor design parameters. Smaller engines have a higher sensitivity to fouling than larger engines. It was concluded that the degree of the particle deposition on blades increases with growing angle of attack. Further, the sensitivity to fouling also increases with increasing stage head. Multishaft engines are more sensitive to fouling than single-shaft engines. Design parameters such as air inlet velocity at the inlet guide vanes (IGVs), compressor pressure ratio, and aerodynamic and geometrical characteristics will determine the inherent sensitivity to fouling for a specific compressor design.

Site location and surrounding environment. The geographic area, the climatic condition, and the geological plant location and its surrounding environment are major factors that are influencing compressor fouling. These areas can be classified in desert, tropical, rural, arctic, off-shore, maritime, urban, or industrial site locations. The expected airborne contaminants (dust, aerosols) and their nature (salts, heavy metals, etc.), their concentration, their particle sizes and weight distribution, as well as the vegetation cycles and the seasonal impact are important parameters influencing the rate and type of deposition.

Plant design and layout. Predominant wind directions can dramatically affect the compressor fouling type and rates. Orientation and elevation of air inlet suctions must be considered together with the location of air/water cooling towers in a combined cycle plant, the possibility of exhaust gas recirculation into the air inlet, orientation of exhaust pipes from lube oil tank vapor extractors, as well as with other local and specific sources of contaminants such as location of highways, industries, seashores, etc.

Other plant design parameters that affect the rate of compressor fouling are:

- The selection of the appropriate type of air inlet filtration system (self-cleaning, depth loading-, cell-, pocket-, mat-, pleat-, oil-bath filter, etc.), the selection of filter media, the number of filtration stages, weather louvres, inertial separators, mist separators, coalescer, snow hoods, etc. Design parameters such as air velocity through the filters and filter loading and their behavior under high humidity, pressure drops, etc., are also factors of high consideration.
- When inlet conditioning systems are used, appropriate mist eliminators should be installed downstream of evaporative coolers. Inlet chilling in humid areas will result in continuous saturated conditions downstream. Thus, the presence of dust contamination in the air can combine with the moisture and additionally contribute to compressor fouling.

Plant maintenance. Quality of air filtration system maintenance, frequency of compressor blade washing (deposition leads to higher surface blade roughness, which in turn leads to faster rate of degradation), prevention of potential bearing seal oil leaks into air inlet stream, periodic water quality control in closed-loop evaporative cooling systems, etc., are all measures that can positively influence compressor fouling and their rate of fouling.

Atmospheric parameters. Ambient temperature and relative humidity (dry and wet bulb temperatures), wind force and direction, precipitation, fog, smog, or misty condition, atmospheric suspended dust concentration related to air density, air layer mixing by air masses, etc., are parameters that impact on the rates of fouling.

Correlation between rate of power degradation and atmospheric condition

Based on various field tests and observations, a correlation between rate of power degradation and atmospheric conditions prevailing at the site can be established. As a result of the test conducted over 18 months under the combined online and offline washing regime and presented earlier in this section, we would like to first discuss herewith one of the most important observations made.

Disparity in power loss gradients. Of 40 measured continuous operating periods (without shutdowns and startups), a total of 14 operating periods each between 70 to 72 h can be directly compared; the power output level at which the gas turbine was operated was always approximately the same. Power output measurements were made at the beginning of each period after online washing (100 percent reference point) and at the end of each period, prior to online washing of the next period. Figure T-60 shows the power output losses over 70 op.h. One can see that there is a widespread disparity in resulting power losses over such a short period, the highest loss in performance was 3.1 percent output, and the end of one period shows even a gain of 0.5 percent power output. This surprising result was obtained on the same unit, with the same air inlet filtration system, the same washing nozzle system, the same washing procedure, and the same detergent.

Explanation for the correlation between rate of power degradation and atmospheric condition. It is generally assumed that power losses will depend on the amount of humidity in a specific environment. With the data collected during the above comparative test periods, the total quantity of water and vapor mass flow ingested by the compressor was determined. The respective compressor air mass flows have been calculated by means of the heat balance. It was observed that the average ingested total humidity (water and vapor) amounted to 7.7 tons/h, or in total 548 tons during 70 op.h. The lowest average value during a period was 4.1 tons/h and the highest was 11 tons/h. The plot in Fig. T-53 shows the measured power losses versus the total quantity of humidity (water and vapor) ingested by the compressor for each of the selected comparative operating periods of 70 h; see also Fig. T-60. The results of this test clearly indicates that there must be a correlation between the mass flow of absolute humidity and the loss in power output. Loss in power output increases with increasing mass flow of absolute humidity until it reaches, on the unit tested, a peak at approximately 400 to 450 tons (total over 70 h) before decreasing again.

Correlation between power loss gradients and humidity. Due to the combination of pressure drop and increased velocity in the air inlet, humidity content in the air will start to condense in saturated condition. For instance, assuming a 250 MW gas turbine unit with an air velocity of 0.5 Mach at IGVs and an air mass flow of 500 m^3/s, operated at 12°C gas turbine compressor inlet temperature and 90 percent relative humidity (RH) as compared to 60 percent RH, then the total condensing water mass forming droplets will be up to 6.3 tons/h at 90 percent RH as compared to 3.1 tons/h at 60 percent RH. The latent heat released by the condensing water will be higher at 90 percent RH; therefore, the static temperature drop in the air

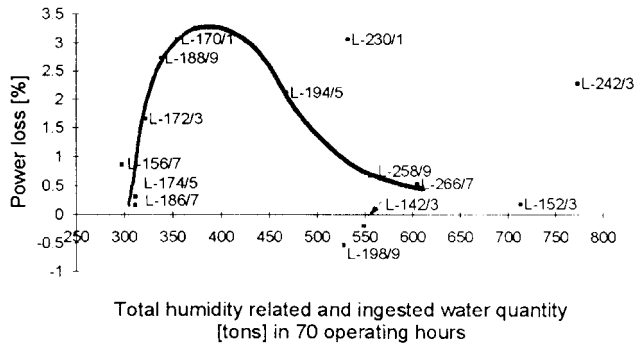


FIG. T-53 Power losses vs. total absolute humidity for 14 comparative operating periods. (Source: Stalder.)

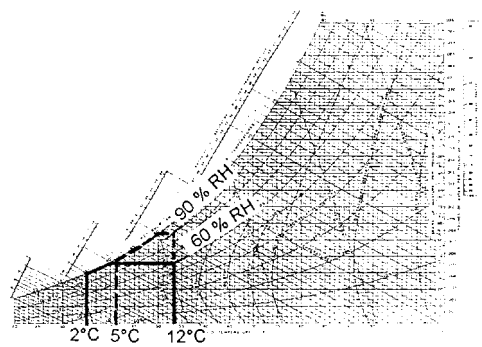


FIG. T-54 Psychrometric chart for air inlet saturating condition. (Source: Stalder.)

inlet at 90 percent RH is 7°C whereas it is 10°C at 60 percent RH. See also the illustrated psychrometric chart in Fig. T-54.

The above mass flow amount of condensed water droplets is impressive and gives rise to the following explanation or combination thereof for power degradation patterns:

- *Influence of humidity.* Surface wetness of compressor blades operating in saturated condition will modify the aerodynamic boundary layer and cause a decrease in performance.
- *Latent heat release.* The ingested air temperature will increase at the compressor bellmouth entry as condensation occurs and latent heat is released, thus reducing cycle efficiency.
- *Water wettable and water soluble deposits.* With lower amounts of condensed water droplets, the ingested soil will combine with the water droplets and deposit on the vane and blade profiles. The rate of deposition will increase with the resulting roughness. Above a certain amount of formed condensed water droplet mass flow, the blading will be naturally washed and power losses due to fouling will be generally recovered to some extent.
- *In case of hydrocarbon type of deposits.* The same effect as above will occur in the lower range of condensed water droplet mass flow. However, water being a

poor hydrocarbon dissolver, the natural washing effect in the higher range of condensed water droplet mass flow will be very limited, if at all. Power losses due to fouling will continue until reaching equilibrium.

- *Combination of water wettable/soluble and hydrocarbon type of deposits.* This type of deposit is very common, and depending upon the mass relationship of hydrocarbon vs. water wettable/soluble parts, and their respective embedment in the deposition layers, the natural washing effect in the high range of condensed water droplet mass flow can also be very limited.
- *Fouling rate in the low range of ambient temperature.* The flattening of the saturation curve in the low ambient temperature range (< approx. –10°C), in the psychrometric chart, shows that the amount of water droplets that can combine with soil is very much reduced and thus lower fouling rates, if at all, can be expected.
- *Duration and sequence of operation in saturated condition.* The changes in rate of power losses noticed over a given operating period will be very influenced by the duration and sequences of operation in saturated condition (vs. dryer condition). For instance, it is most probable that a high amount of water droplet mass flow ingested in the beginning of an operating period with a clean compressor will affect the changes in rate of power loss differently than if it would occur toward the end of such an operating period.

Discussion on washing technology, acquired know-how, and experiences

Fouling deposits. Most fouling deposits are mixtures of water wettable, water soluble, and water insoluble materials. Very often pH 4 and lower can be measured in compressor blade deposits. This represents a risk of pitting corrosion. Furthermore, these deposits become more difficult to remove if left untreated as the aging process bonds them more firmly to the airfoil surface, thus reducing cleaning efficiency.

Water-soluble compounds cause corrosion since they are hygroscopic and/or contain chlorides that promote pitting corrosion. They can be rinsed; however, they can also be embedded in water-insoluble compounds.

Water-insoluble compounds are mostly organic such as hydrocarbon residues or from silica (Si).

Online and offline compressor cleaners. The cleaners available today in the market are generally nonionic and designed to fulfill gas turbine engine manufacturers' specification for both online and offline cleaning, thus also simplifying stock keeping and handling on site. Offline cleaning is done at crank speed, the crank-soak washing method, with the engine cooled down (requires cooling down time), whereas online cleaning is done during operation of the engine. The used detergent and demineralized water in online cleaning applications must generally fulfill the manufacturers' fuel specification.

Function of a compressor chemical cleaner. The main constituent of a cleaner is its surfactant (surface active agent), the purpose of which is to reduce surface tension of the solution enabling it to wet, penetrate, and disperse the deposits. Such rapid intimate mixing cannot be achieved with water alone. Surfactants are therefore needed for water-insoluble deposits—both liquid and particulate types—to enable their removal from compressor blade surfaces and to prevent redeposition.

Foaming of the compressor cleaner. The amount of foam generated by a compressor cleaner is an indication of the degree of activity and therefore of the effectiveness

of the surfactants used in the cleaner. Foam, being light, will help to achieve a better distribution, and penetration of the cleaning solution into the deposits during offline washing; it will keep moisture inward and thus extend the contact time. The foam will be rapidly displaced during rinsing and will help to remove surfactants left on the blade surface. Water films alone will tend to drain off the blades more rapidly, thus reducing contact time during the offline soaking period. In addition, foam acts also as a dirt carrier.

Corrosion inhibitors/compressor rinse. Some compressor cleaners do contain a corrosion inhibitor claimed to inhibit corrosion by neutralizing the influence of salts during a certain period and thus sparing a time-consuming dry-out run after offline washing. This may be important for jet engines that are frequently not reoperated immediately after compressor crank washing. However, the above situation is different in a stationary gas turbine generating plant. pH or conductivity measurements in offline effluent water have shown that salts are often left behind to some extent after washing with the cleaner agent only, thus rinsing with water alone will be beneficial to remove the salt solution after washing. Consequently a blow-run followed by a dry-out run is recommended. The blow-run at crank speed will help to drain all trapped water in the gas turbine internal piping systems and the completed dry-run at no-load will give the assurance that the mechanical condition of the gas turbine has been reestablished and that the latter is ready for start by the plant dispatcher.

Corrosion inhibitors have a high affinity to surfaces and will tend to form a film on the blades; therefore, there is a high potential risk during online cleaning that this film can produce a decomposed material deposit in the temperature range of 200°C in the middle section of the compressor. One recommendation is to eliminate the salts left behind by doing a demineralized water rinse of the same duration as the cleaning period with detergent after each online cleaning.

Hot or cold water for offline washing? Hot wash water will soften the deposits better than cold water; further, it will also help to prevent thermal shocks and thus reduce the cooling time period. However, hot water has also several disadvantages. It has to be hot enough (in the range of 60 to 80°C), the wash skid requires a heating system, heating energy, an insulated tank, and piping. The heater has to be started approximately 12 h before washing (depending on heater capacity). Thus, washing has to be scheduled in advance. Many users find it easier to wash their engines off-line by opportunity. Using cold water reduces the manufacturing and installation cost of the washing skid as well as the auxiliary energy consumption (they can be significant for large engines having a high cleaning solution injection mass flow). Using cold water (approx. >15°C) will also increase significantly maintenance scheduling flexibility. A weaker softening of deposits by cold water is by far compensated by more frequent washing (combined online and offline washing regime) and good cleaner surfactant performance.

Water-based cleaners against solvent-based cleaners. Water-based cleaners have a broader range of application as they are able to deal with oily soils, water-soluble salts, and particulates (silica, clay, etc.). Solvent-based cleaners have a more specific application where the oily portion in deposits is predominant. For offline washing, to facilitate effluent water disposal, water-based cleaners must be easily bioeliminable and biodegradable and they should comply with the Organization of Economic Cooperation and Development (OECD-302B) requirements. When using solvent-based cleaners, effluent water generally needs to be handled and treated like used oil. Cleaner selection is normally dictated by the type of deposits, the

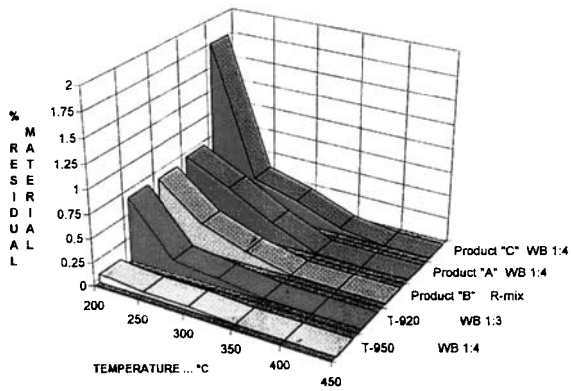


FIG. T-55 Material residual % vs. temperature for water-based product. Note that product "C" showed an oil film at 200°C. WB, water-based cleaner; R-mix, ready for use WB cleaner. (Source: Stalder.)

available effluent water disposal system, and the relevant local regulation for effluent water disposal.

For online washing, the selection and the quality of the type of cleaner can play a very important role. Water-based cleaners diluted with demineralized water in ratios as recommended by detergent manufacturers have much lower residual formation (normally <0.1 to 0.7 percent at 200°C) (see Fig. T-55), whereas solvent-based cleaners can have up to 3 percent or more of residual formation at 200°C. Turbotect also recognized in a very early stage that the selection of raw materials used in the manufacture of the cleaners for both solvent- and water-based cleaners is most important. An inappropriate selection of raw materials can lead to the formation of resinlike deposits or oily films on blade profiles during decomposition, generally in the middle section of the compressor. Such sticky deposits can collect dirt washed off from the front and foul the compressor downstream. Thus, frequent washing may lead to an undesired result whereby no power recovery after washing will be achieved. Properly selected raw materials in the manufacture of cleaners will not lead to the formation of such gummy buildup deposits. In order to reduce deposits downstream of the compressor, it is also preferable to use water-based cleaners for online washing (lower residual content).

Solvent-based cleaners do have a better cleaning efficiency against oily-type deposits. Figure T-56 shows cleaning efficiencies on oily deposits achieved by demineralized water and various water- and solvent-based cleaners. Note that cleaning efficiency will also be affected by the injection time.

Compressor online cleaning with water only. Some plant operators are doing online washing regime tests by comparing the power recovery results using demineralized water only against using a detergent cleaner. The value of such tests can be very important because it will help the plant users to understand the phenomena of compressor fouling by monitoring very closely the unit's performance and to experiment with various washing regimes. As already mentioned, compressor fouling behavior is plant specific. Water as an online cleaning media will certainly work, however, only if the deposits are totally water wettable and/or water soluble. We believe that a minority of engines will have only such deposits. Using water only as cleaning media will impose very short time intervals between washings to avoid any buildup at all. As otherwise it can be detrimental in case the deposits are water insoluble or a combination of water-soluble and -insoluble

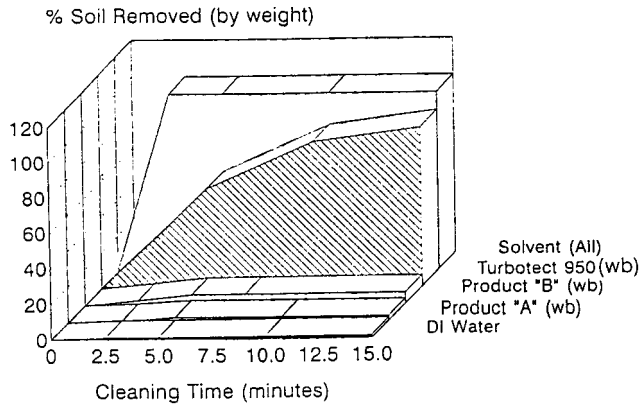


FIG. T-56 Comparison of cleaning efficiency. wb: water-based cleaner; DI water, demineralized water. (Source: Stalder.)

compounds, the insoluble part will not be washed off, thus allowing further buildup of deposits in the front stages. Water-based cleaners for online washing are recommended.

Detergent use and optimum washing regime. An optimum washing regime will keep power decay to a minimum by applying combined online and offline cleanings. An online washing period shall always be started with a clean engine after an overhaul or an offline cleaning. The time intervals between online washing shall be kept short, approximately every three days to every day. However, depending on the type of deposits (portion of insoluble compounds), detergent cleaners may be used for every second or third online wash or up to once a week only. Between times, online cleaning can be made with demineralized water only. In case the time intervals between detergent online washes is extended, there will be a higher risk of downstream contamination due to larger portions of insoluble compounds being lifted at once and carried through the compressor. Thus, frequent online cleanings with detergent in order to have only a minute portion of insoluble compounds being washed off at any given time are recommended.

Online compressor cleaning efficiency. Deposits on the profile of the first stage vanes are primarily responsible for a significant reduction in air mass flow through the compressor, thus reducing the power output. Online cleaning is most effective in removing such deposits on the first-stage guide vanes, thus restoring design air mass flow, e.g., see Figs. T-57 and T-58. Frequent online cleaning keeps the first-stage guide vanes clean. Droplets of cleaning solution may survive up to the sixth stage; however, most are vaporizing by then and the residue/ashes will be centrifuged along the compressor casing. Therefore, no cleaning solution will be available for removing deposits on downstream stages.

The pictures of the IGV (Fig. T-58) are interesting and show a very effective online washing. When observing the film deposit on the blade tip, one can easily conceive the severeness of power loss and deposits on this first stage if no online wash had been applied by the operator.

Offline compressor cleaning efficiency. The offline cleaning method is very efficient for removing all deposits on all the compressor stages. For this purpose offline washing shall preferably be performed at variable crank speed, e.g., by injecting

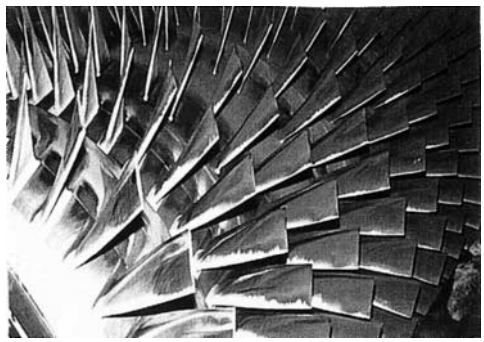


FIG. T-57 Picture of a compressor that was washed online every four days with detergent during approximately a one-month continuous operation period. (Source: Stalder.)

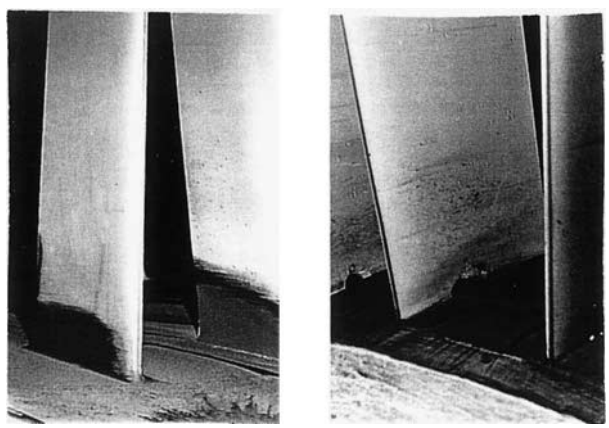


FIG. T-58 First-stage guide vanes of the same compressor (pictured in 1/3 vane length from the tip and from the root), washed online every four days with detergent during approximately a one-month continuous operation period. (Source: Stalder.)

the cleaning and rinsing solution during coast down of the shaft after an acceleration of up to 500 to 600 rpm. By doing this, the pattern of the centrifugal forces on the injected solution through the compressor will decrease and allow a better wetting and distribution on the blade and vane surfaces of all stages. By contrast, offline washing at high and constant cranking speed will end up in lower offline cleaning efficiency. Conductivity measurements in rinsing water as well as checks on the clarity/turbidity of rinsing water will help to assess the cleaning efficiency.

Offline cleaning intervals. Irrespective of the compressor performance degradation, a sound combined online and offline washing regime should incorporate at least four offline compressor cleanings per year in order to remove the salt-laden deposits on the downstream stages. This recommendation is also valid for a peak load unit running only a few hours per year.

Washing equipment and procedure

System engineering and equipment. The following two basic principles have been determined for the engineering of both online and offline injection systems:

- Limit the injection mass flow to what is absolutely required for a good cleaning efficiency.
- Totally reliable nozzle injection system, giving due consideration to all operational aspects.

Online cleaning injection system design criteria. The objective is to achieve the highest washing efficiency at the smallest injection mass flow for the following essential reasons:

- *Online injection mass flow*. Gas turbine engine manufacturers are very much concerned with deposition of washed off dirt from the front stages onto downstream compressor stages. Further, that washed off dirt may enter into sophisticated airfoil film cooling systems of turbine blades with the potential to clog them, resulting in temperature hot spots. In this respect, a sound isolation scheme to avoid runoff effluent water penetrating into the internal engine piping systems during offline washing is of prime importance. Low level of flame detector intensities during online cleaning on units with dry low NO_x combustors are also claimed for potential tripping of the units (e.g., fogging of flame detector lenses, etc.). Some users also observe higher CO emission levels during online washing. Small quantity online cleaning injection mass flow combined with the optimum washing regime will counteract against the previous claimed potential problems in modern gas turbines. Therefore, a small injected quantity of online cleaning solution mass flow is preferable. For instance, Turbotect's online compressor cleaning system for a 120 MW heavy duty engine comprises 30 injection nozzles, the total mass flow applied is only a fraction of comparative online washing systems. This low mass flow has demonstrated that it does not impair the cleaning efficiency and also has the further advantage of lowering significantly overall demineralized water and cleaner consumption. In addition, it will also reduce the required size, volume, and cost of the washing skids.
- *Wetting*. A very effective wetting of the IGVs is reached by a uniform and finely distributed atomized cleaning solution. Droplets are subject to gravity, they must be stable in size, small enough that they do not cause blade erosion, and light enough that they do not drop out of the airstream before they reach the compressor blade surface. A nonuniform wetting of the IGVs will result in spot cleanings and heavier droplets will most likely fall to the bottom, wasting some injected cleaning solution.
- *Equipment*. The online nozzle injection design is of primary importance to achieve a high washing efficiency. The nozzles are designed to inject a small quantity of fine atomized cleaning solution into the airstream where it will be thoroughly mixed and carried uniformly into the compressor bellmouth. A relatively high number of nozzles positioned in the air inlet casing on both upstream and downstream sides of the bellmouth ensure a better distribution of the injected fluid into the airstream and consequently a better wetting. See Fig. T-59.

The nozzle atomizer is integrated into a spherical body that can be rotated in two dimensions to set the spray angle. The adjustment of the spray angle allows proportioning of atomized cleaning solution mass flow that shall penetrate into the airstream from the mass flow that shall remain trapped in the boundary layer for root and tip blade cleaning. The nozzle body is installed flush mounted under the surface of the intake structure and penetration of the nozzle into the airstream is

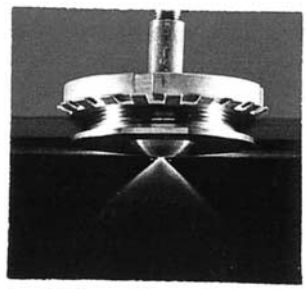


FIG. T-59 Online injection nozzle (patented) allowing orientation and fine tuning of the spray in any direction. (Source: Stalder.)

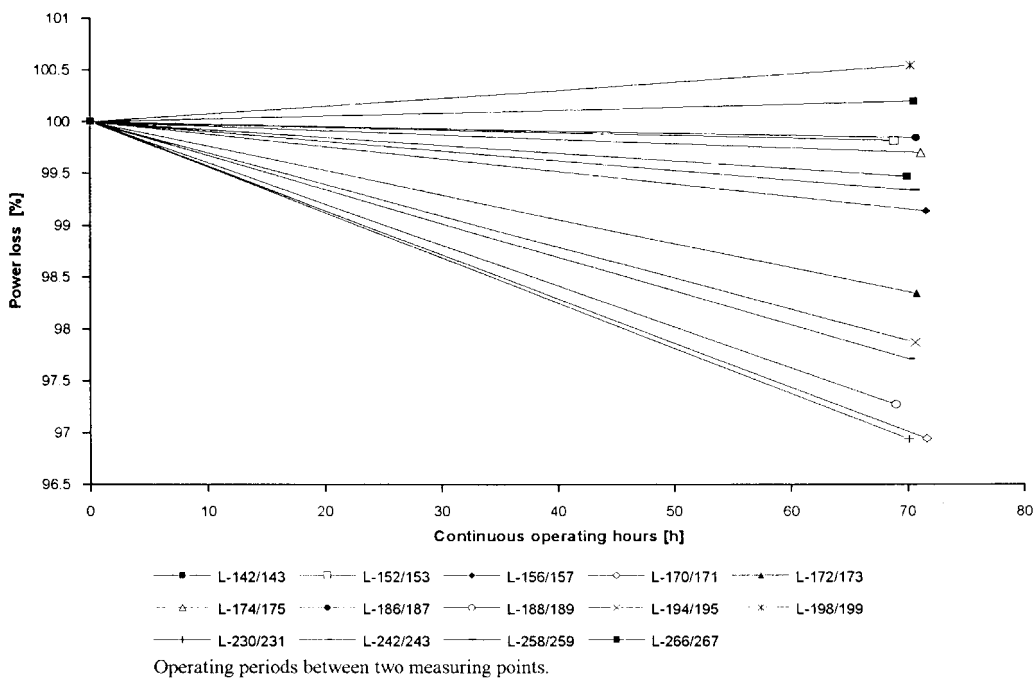


FIG. T-60 Power losses over 70 continuous operating hours. (Source: Stalder.)

minimal to avoid inducing vibration into the airstream. Further, this design also prevents misuse of the nozzles as climbing support during compressor intake inspection. Its design is ruling out loss of any parts into the airstream.

Offline cleaning injection system design criteria. The objective again is to achieve the highest washing efficiency at the smallest injection mass flow for the following essential reasons:

- *Offline injection mass flow*. First, it is obvious that users are interested in low quantities of effluent water to be disposed off. It is also claimed that offline water effluent transported up to the exhaust during offline washing may wet and soak into the expansion joint fabric (or also in recuperator compensator). Thereupon

some expansion joint fabric may lose some of its thermal insulation capacity, resulting in overheating of the fabric, which can damage the expansion joint.

A low offline injection mass flow will also reduce the potential risk and measures to be taken against trace metals and alkali compound contaminations in exhaust systems where selective catalytic reactors for NO_x reduction or CO catalysts are installed. A low offline injection mass flow will significantly reduce the required size, volume, and cost of washing skids and consequently the overall water and cleaner consumption.

- *Wetting / equipment*. A very effective wetting of the IGV's suction area is achieved by using full cone jet spray nozzles. The number of nozzles will be defined by the area to be wetted, usually the area between two struts. The necessary offline injection mass flow characteristic will therefore be determined by the area to be wetted and impacted by the jet spray and the distance between the location of installation in the inlet casing up to the area of impact; the injection pressure is generally between 5.5 to 6 bars. The spray jets are also subject to gravity, the nozzle is designed such that an angle (up to 5°) can be adjusted for compensation. The offline washing (soaking) and rinsing method can be considered as a mechanical erosion of deposit layer and soaking time will allow the cleaner to penetrate and soften the deposit layers. For instance, systems with high atomization pressure will have no impact pressure on the IGVs, this because the spray pressure will have lapsed approximately 20 cm from the nozzle outlet and the atomized droplets will need to be carried by the small airstream produced at crank speed. These high-pressure systems do not show the same effectiveness in removing salts and insoluble compounds on downstream stages.
- *Drainage*. The effluent water collection system to drain the dirty water outside the engine and the isolation scheme to hinder runoff water to penetrate sensitive systems such as sealing and cooling air systems, instrumentation air systems, etc., during washing is of prime importance.

The physical location of the air system taps on compressor casing is also important. Taps on the bottom are likely to drain runoff water; they should be located in the upper part of the casing. Drains in air inlet casing, compressor casing, combustion chamber, and exhaust should be located at the lowest point. The drain diameters should allow a good runoff; it can be observed that sometimes they get plugged because of dirt not properly evacuated. A good isolation and drainage concept will ensure that no dirty water enters into cooling-, sealing-air, and other systems during the offline washing procedure.

Paint and corrosion damage in air inlet manifolds. Some users are claiming that the corrosion protective paint applied originally inside the air inlet casing softens and some detaching paint spots are deposited by the airstream on IGVs due to frequent online washings, possibly because of the chemistry involved in the cleaner.

Chemistries involved in the formulation of compressor cleaners are very soft. These cleaners have all been tested for immersion corrosion, sandwich corrosion, and hot corrosion on metals, alloys, coatings, etc., used in the manufacture of compressors, and their affect on polymeric materials, rubber, silicone elastomers, epoxy adhesives, on painted surfaces, etc., according to existing US MIL-C-85704B specification "Cleaning Compound, Turbine Engine Gas Path."

Investigations conducted together with paint manufacturers have shown that softening and detaching paint systems in air inlets are caused for various reasons or combination thereof; even paint systems have shown damages in units where no online washing was performed. Casings are also very often wetted by the existing

condensed water generated in saturated condition. Some causes for potential paint degradation in air inlet casings are as follows. Quality of surface preparation was not sufficiently carried out (sharp-edged material to rough the surface by blasting is a prerequisite).

- Recommended paint thickness, drying time, and application temperatures were not followed.
- In case of paint repairs, old paint should be grinded away and not removed with liquid paint removers.
- Combination of selected paint systems (type 1 or 2 component systems, etc.) for priming, first coat, and finish coating are not compatible.
- Some paint systems show swelling characteristics.

Fouling rates can vary greatly and are very specific to each plant. The gas turbine unit design parameters, the site location, and its surrounding environment, the climatic conditions, and the plant concept-, design-, and its layout are given once the plant has been built. The site weather parameters are then having the largest impact on fouling rates and performance degradation. Therefore, improvements on the plant profitability by reducing the impact of compressor fouling can only be related to the plant maintenance, monitoring of plant performance, the performance of the installed washing system, the selection and quality of the detergent used, and the optimum chosen washing regime.

References and Additional Reading

Aker, G. F., and Saravanamuttoo, H. I. H., 1988, "Predicting Gas Turbine Performance Deterioration Due to Compressor Fouling Using Computer Simulation Techniques," ASME Paper No. 88-GT-206.

Bird, J., and Grabe, W., 1991, "Humidity Effects on Gas Turbine Performance," ASME Paper No. 91-GT-329.

Diakunchak, I. S., 1991, "Performance Deterioration in Industrial Gas Turbines," ASME Paper No. 91-GT-228.

Haub, G. L., and Hauhe, W. E. Jr., 1990, "Field Evaluation of On Line Compressor Cleaning in Heavy Duty Industrial Gas Turbines," ASME Paper No. 90-GT-107.

Hoeft, R. F., 1993, "Heavy Duty Gas Turbine Operating and Maintenance Considerations," GER-3620B, GE I&PS.

Jeffs, E., 1992, "Compressor Washing On Line for Large Gas Turbine," Turbomachinery International, Vol. 33, No. 6, September/October 1992.

Kolkman, H. J., 1992, "Performance of Gas Turbine Compressor Cleaners," ASME Paper No. 92-GT-360.

Kovacs, P., and Stoff, H., 1985, "Icing of Gas Turbine Compressors and Ways of Achieving Uninterrupted Operation," Brown Boveri Revue, Vol. 72, April 1985, pp. 172–177.

Lakshminarasimha, A. N., Boyce, M. P., and Meher-Homji, C. B., 1992, "Modelling and Analysis of Gas Turbine Performance Deterioration," ASME Paper No. 92-GT-395.

Mezheritsky, A. D., and Sudarev, A. V., 1990, "The Mechanism of Fouling and the Cleaning Technique in Application to Flow Parts of the Power Generation Plant Compressors," ASME Paper No. 90-GT-103.

Saravanamuttoo, H. I. H., and Lakshminarasimha, A. N., 1985, "A Preliminary Assessment of Compressor Fouling," ASME Paper No. 85-GT-153.

Sedigh, F., and Saravanamuttoo, H. I. H., 1990, "A Proposed Method for Assessing the Susceptibility of Axial Compressors to Fouling," ASME Paper No. 90-GT-348.

Stalder, J. P., and van Oosten, P., 1994, "Compressor Washing Maintains Plant Performance and Reduces Cost of Energy Production," ASME Paper No. 94-GT-436.

Stalder, J. P., 1992, "Professional System Approach to Compressor Cleaning, Case Studies," GCC-CIGRE Paper, Third Annual Conference, Dubai, May 1992.

Tarabrin, A. P., Schurovsky, V. A., Bodrov, A. I., Stalder, J. P., 1996, "An Analysis of Axial Compressor Fouling and a Cleaning Method of their Blading," ASME Paper No. 96-GT-363.

Thames, J. J., Stegmaier, J. W., and Ford, J. J. Jr., 1989, "On Line Compressor Washing Practices and Benefits," ASME Paper No. 89-GT-91.

Ul Haq, I., and Saravanamuttoo, H. I. H., 1993, "Axial Compressor Fouling Evaluation at High Speed Settings Using an Aerothermodynamic Model," ASME Paper No. 93-GT-407.

Ul Haq, I., and Saravanamuttoo, H. I. H., 1991, "Detection of Axial Compressor Fouling in High Ambient Temperature Conditions," ASME Paper No. 91-GT-67.

Zaba, T., and Lombardi, P., 1984, "Experience in the Operation of Air Filters in Gas Turbine Installations," ASME Paper No. 84-GT-39.

Gas Turbine Controls (*see* Control Systems)

Gas Turbine Explosion Hazards (*see* Explosion)

Gas Turbine Fuel (*see* Fuels, Alternative)

Gas Turbine Inlet Air Filtration (*see* Air Filtration)

Turbines, Steam

Steam Turbine Components* (see Fig. T-61)

Steam turbine casing

The steam turbine casing is split horizontally on the plane of the shaft centerline. The steam chest and high pressure end are constructed of cast alloy or carbon steel. The casing material is determined by the inlet pressure and temperature of the steam.

For high steam ratings, suspended or inner casing designs are available. This design is thermoflexible centerline supported in the outer casing. All steam turbine casing front ends are designed with centerline support to maintain proper alignment. Casing and supports (flexplates or sliding supports) are designed to prevent excessive distortion caused by temperature differential. Noncondensing steam turbines also have centerline support systems on the back end. Condensing steam turbines incorporate an exhaust casing, constructed of either fabricated steel, cast steel, or cast nodular iron.

Solid, forged rotor construction

This design is superior in operating reliability when compared to rotors that are built up or contain mechanically fastened blade wheels. Smaller, lighter, high-speed steam turbine rotors may be broached, which provides uniformity in manufacture. This process allows 360° blading and blade shrouding resulting in reduced stress concentrations. Ultimately, this protects blading from vibratory stresses while increasing reliability. Solid rotor construction (with the shaft and discs machined from a single steel alloy forging) is standard on steam turbines. The billet is forged, examined using nondestructive testing methods, degaussed, and rough machined.

Blades

Steam turbines are constructed with rugged impulse-type blading to minimize vibration stresses. Moving blades are machined from forgings or bar stock. Leading edges are thermally hardened or mechanically protected with inserts to reduce erosion on wet stages.

Nozzle rings

Nozzle rings are bolted to the steam chest at the inner circumference and are held in place at the outer circumference by a segmented wedge ring where practical. This permits the steam chest to expand at a different rate than the nozzle rings. As

---

* Source: Demag Delaval, USA.

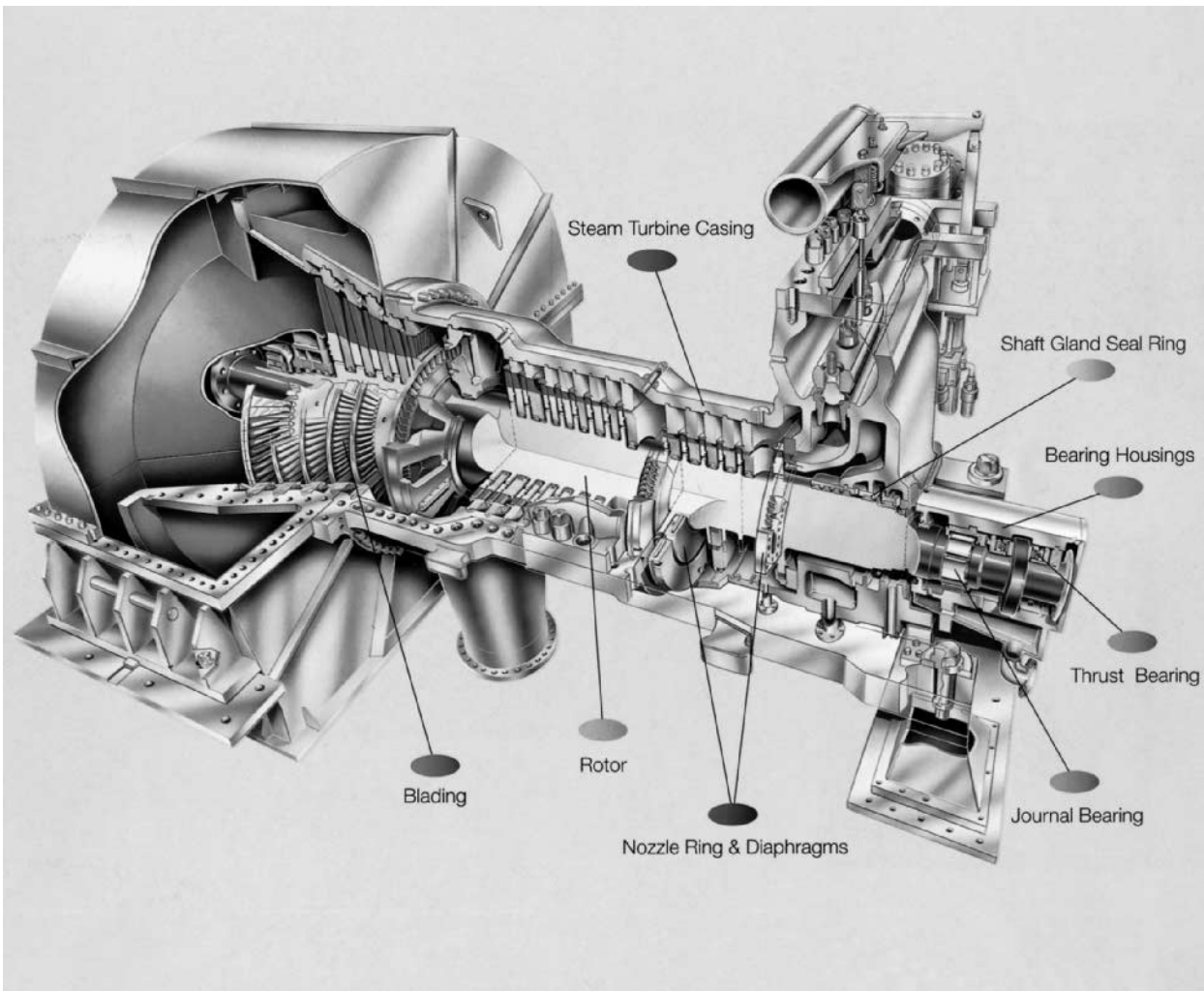


FIG. T-61 Steam turbine components. (Source: Demag Delaval.)

a result, there is a reduction of thermal distortion and reduced possibility of cracking.

Diaphragms

The diaphragm requires that nozzle vanes be welded to stainless steel guide rims (or spacer bands), with the finished assembly welded to the inner and outer diaphragm rings. The assembly is stress relieved and machined to final dimensions. The diaphragms are split and located in grooves within the casing by a centerline support system. Noncorrosive keys are fitted between the diaphragm halves to prevent steam leakage. This technique results in sound fabrication and provides accurate control of the steam flow.

Interstage steam sealing

Each diaphragm is fitted with a spring-loaded metal labyrinth seal located inside a groove machined in the diaphragm. This step prevents steam leakage from stage to stage. Blading tip seals are applied for optimum efficiency.

Shaft gland seal ring

Spring-loaded labyrinth-type gland seals are used to minimize air leakage into and steam leakage from the steam turbine. An automatic gland steam control system and condenser can be supplied.

Bearing housings

The bearing housings are of horizontally split design and are arranged for pressure lubrication of the bearings. Bearings can be inspected and serviced without removing the coupling hub or breaking the pressure-containing casing joint.

Journal and thrust bearings

Tilting-pad radial bearings provide optimum rotor stability at all operating loads by forming an "oil wedge" at each shoe. This design is very effective in damping vibrations and is far superior to sleeve-type radial bearings. Double-acting, self-leveling thrust bearings are used, providing maximum protection against process upsets.

Optional turning gear prior to startup and shutdown for large bearing span rotors can be supplied for automatic or manual operation.

Auxiliary Systems*

Microprocessor-based steam turbine governor control system

The electrohydraulic control system automatically and continuously monitors and sets the steam turbine speed to satisfy customers' specific requirements. Low maintenance operation typifies this type of system, which also features solid-state electronics, redundant speed pickups, and signal processing channels to reduce problems ordinarily associated with mechanical controls. The result is improved reliability, accuracy, and overall enhanced steam turbine performance.

Valve gear assembly

The valve gear assembly controls the steam flow to the turbine. It utilizes a simple but rugged bar lift design for increased reliability. The individual valve spindles are designed with spherically seated nuts to compensate for any minor misalignment while reducing the possibility of valve spin. The contact surfaces of both the control valves and seats are stellited for increased life. Another major feature is the large springs that are designed to enable fast closing times for the valves.

Overspeed protection system

Steam turbines are equipped with an emergency overspeed protection system that is separate from the main governing system. In the event of excessive rotor speed, this system shuts down the steam flow to the steam turbine by closing both the stop and control valves. This is an electronic failure detection system that is redundant and can be tested during operation.

Electronic control system

The steam turbine can be supplied with a programmable logic controller (PLC) system for digital control of steam turbine auxiliary systems and diagnostic monitoring of the steam turbine unit. This PLC system monitors operation and performance and safeguards against excess pressure and steam flow.

---

* Source: Demag Delaval, USA.

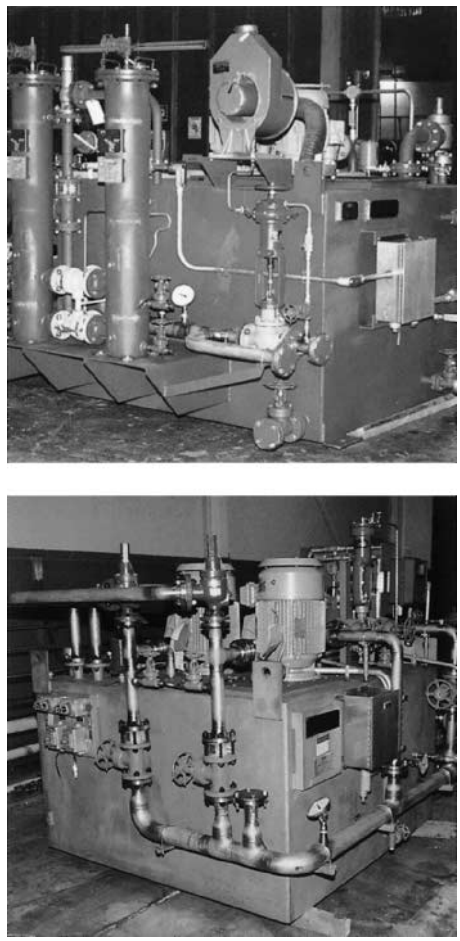


FIG. T-62 Steam turbine lube oil and control oil consoles maximize the use of space while providing ample access for maintenance. (Source: Demag Delaval.)

Oil supply systems

The oil supply system is comprised of an integrated, freestanding console that provides oil for control valve and trip system hydraulics and bearings. These systems are furnished complete with coolers, filters, oil reservoirs, and integral piping. Either positive displacement or centrifugal oil pumps (sized to provide an adequate supply of oil to all bearings) are used. A separate hydraulic system can be supplied for larger capacity steam turbines and steam turbines with controlled extractions. See Fig. T-62.

The turning gear system

This system incorporates a modern "overrunning automatic clutch" that engages automatically when the rotor reaches turning gear speed, and it automatically disengages when that speed is exceeded. See Fig. T-63.

The steam seal system

This system features labyrinth-type packing to minimize steam leakage outward and air leakage inward. It vents high-pressure steam leakage to seal the low-pressure packing while the steam turbine is in operation. See Fig. T-64.

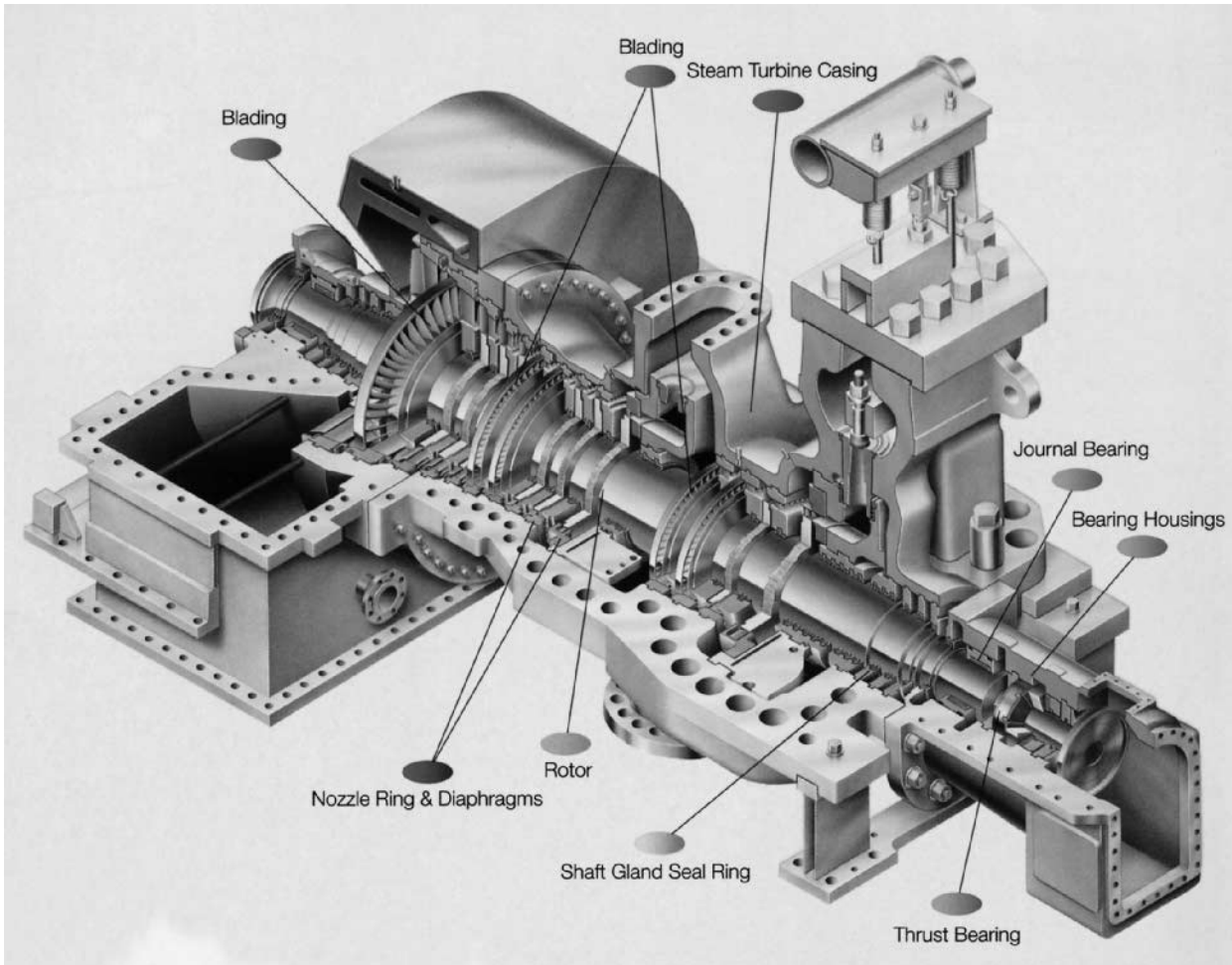


FIG. T-63 Section through a steam turbine. (Source: Demag Delaval.)

Vibration monitoring

Shaft vibration detecting equipment continuously monitors the actual dynamic conditions of the machine.

Temperature sensors

Indicated thrust and journal bearing temperatures are systematically monitored.

Steam Turbine Theory*

General

The steam turbine is the most widely used prime mover on the market. In large capacities, it rules without competition; for smaller sizes, the gas turbine and the internal combustion engine are its only competitors; but for the smallest sizes both

*Source: Demag Delaval, USA.

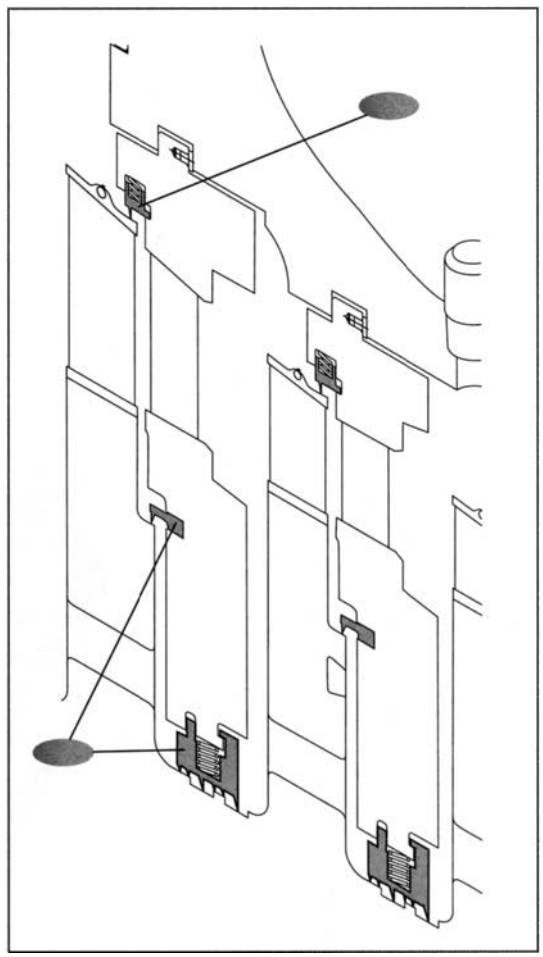


FIG. T-64 Interstage steam sealing. (Source: Demag Delaval.)

the reciprocating steam engine and the internal combustion engine compete with the steam turbine for the market.

Steam turbines have been designed and built for an output ranging from a few horsepower to more than 1,300,000 kW, with speeds ranging from less than 1000 rpm to more than 30,000 rpm, for inlet pressures from subatmospheric to above the critical pressure of steam, with inlet temperatures from those corresponding to saturated steam up to 1050°F, and for exhaust vacuums up to $29^1/_2$ inHg.

The turbine requires much less space than an internal combustion engine or a reciprocating steam engine and much lighter foundations since reciprocating forces on the foundations are eliminated. Another major advantage is the turbine's ability to extract power from the steam and then exhaust all the steam or part of it into a heating system or to a manufacturing process, entirely free from oil.

Simplicity, reliability, low maintenance cost, and ability to supply both power and heat are the main justifications for the industrial turbine. A small factory or a building complex cannot produce electric power as cheaply as a large central-station power plant, but if steam is needed for industrial purposes or for heating, the production of power can be combined with the utilization of extracted or exhaust steam and the power becomes a cheap by-product.

The small noncondensing turbine also occupies a large and important field in power plants and marine installations because it is particularly well adapted to drive variable-speed auxiliaries and because its exhaust steam can be used to supply heat to the feedwater. A further advantage of the auxiliary turbine is its availability and convenience as a standby unit in case of interruptions to the power supply of motor-driven auxiliaries.

Steam cycles

The Rankine cycle. Potential energy of steam is transformed into mechanical energy in a turbine. The number of Btu required to perform work at the rate of one horsepower for one hour is 2544; for one kilowatt for one hour it is 3413 Btu.

The enthalpy, or heat content, is expressed as Btu per pound of steam. This is, in effect, the potential energy contained in the steam measured above the conventionally accepted zero point (that of condensed steam at 32°F). Practically, it is not possible to release all the energy so that the end point of heat extraction in a condensing turbine is given by the temperature attainable in the condenser. The considerable amount of energy still contained in the steam at this point cannot be recovered and must be rejected to the cooling water.

The portion of the potential energy that can be used to produce power is called the *available energy* and is represented by the isentropic enthalpy difference between the initial steam condition h_1 and the final condition corresponding to the exhaust pressure h_2. If the condensate enthalpy is h_w, the *ideal Rankine cycle efficiency*, or the *thermal efficiency*, is

$$\eta_R = \frac{h_1 - h_2}{h_1 - h_w}$$

The available energy can be converted into mechanical (kinetic) energy only with certain losses because of steam friction and throttling, which increase the entropy of the steam. The end pressure is therefore attained at a higher steam enthalpy h_2' than with isentropic expansion. The *internal turbine efficiency* then is

$$\eta_i = \frac{h_1 - h_2'}{h_1 - h_2}$$

This efficiency may be reduced to the *external turbine efficiency* by including mechanical and leakage losses not incident to the steam cycle.

Since one horsepower-hour is equivalent to 2544 Btu per hour, the theoretical steam rate of the Rankine cycle in pounds per horsepower-hour is obtained by dividing 2544 by the available energy in Btu. The corresponding value on a pound-per-kilowatt hour basis may be found by dividing 3413 by the available energy. To obtain the actual steam rate at the coupling of the turbine, the theoretical steam rate is divided by the external turbine efficiency, which includes the mechanical losses.

To facilitate steam-cycle calculations, standard tables of the thermodynamic properties of steam can be used. The data contained in these tables are plotted on a *Mollier diagram* (Fig. T-65), which is employed extensively to solve thermodynamic problems relating to steam turbines.

Example. Determine the performance of a condensing turbine operating on a Rankine cycle based on the data in Tables T-4 and T-5.

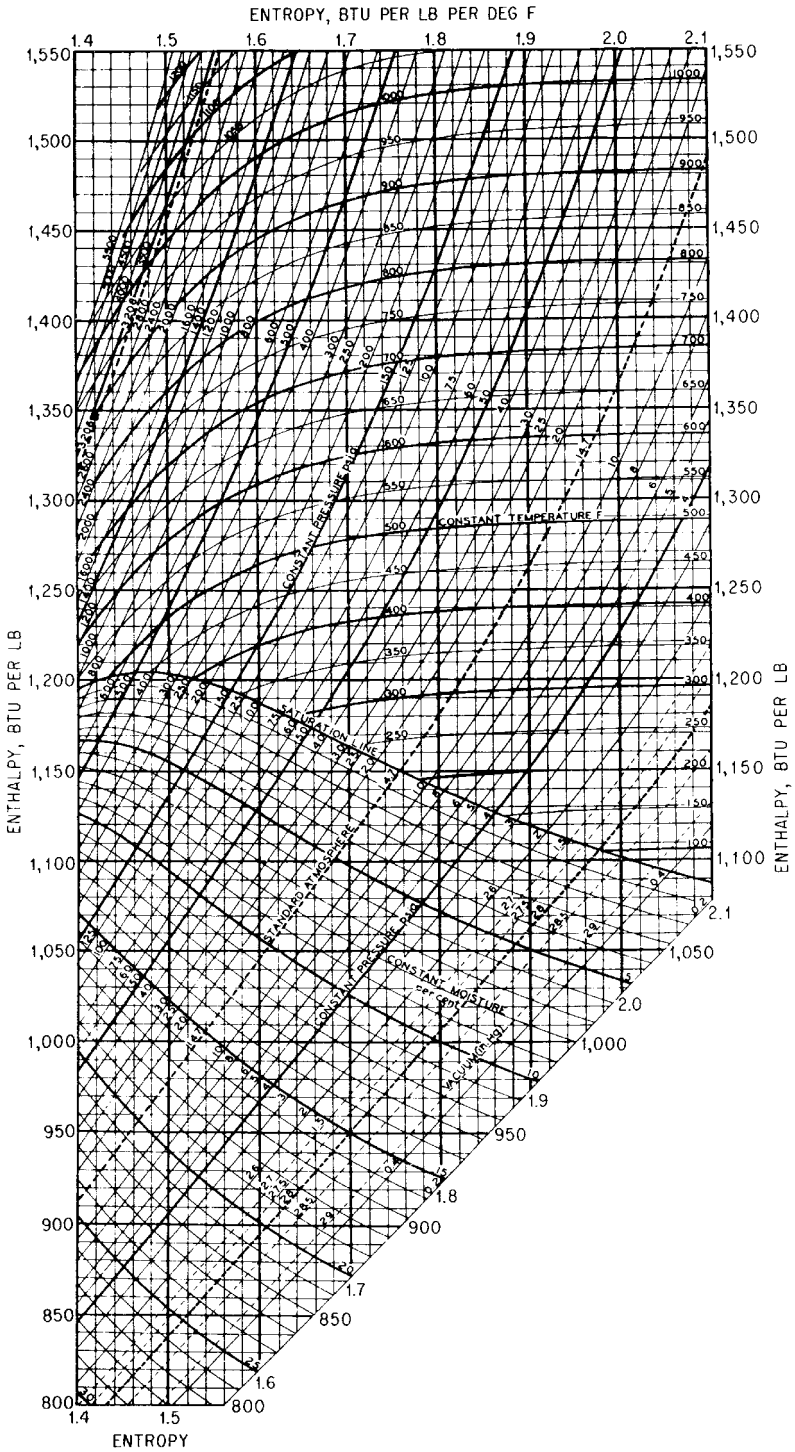


FIG. T-65 Mollier diagram (by permission from 1967 ASME Steam Tables). (Source: Demag Delaval.)

TABLE T-4

Initial steam pressure	200 psia
Initial steam temperature	600°F
Exhaust steam pressure	2 inHg
Moisture in exhaust steam	5 percent
Exhaust steam temperature	101°F
Measured steam rate lb/hp·h	10.5

TABLE T-5

Enthalpy at inlet h_1	1322 Btu/lb
Entropy at inlet	1.6767 Btu/°F
Enthalpy at 2 inHg and entropy of 1.6767(h_2)	936 Btu/lb
Enthalpy at 2 inHg and 5 percent moisture (h_2')	1054 Btu/lb
Enthalpy of saturated at 2 inHg (h_w)	69 Btu/lb

Calculations:

$$\text{Isentropic enthalpy drop} = h_1 - h_2 = 1322 - 936 = 386 \text{ Btu/lb}$$

$$\text{Actual enthalpy drop} = h_1 - h_2' = 1322 - 1054 = 268 \text{ Btu/lb}$$

$$\text{Ideal Rankine-cycle efficiency} = \eta_R = \frac{h_1 - h_2}{h_1 - h_w} = \frac{1322 - 936}{1322 - 69} = \frac{386}{1253} = 30.8 \text{ percent}$$

$$\text{Internal turbine efficiency} = \eta_i = \frac{h_1 - h_2'}{h_1 - h_2} = \frac{1322 - 1054}{1322 - 936} = \frac{268}{386} = 69.5 \text{ percent}$$

$$\text{Rankine-cycle steam rate} = \frac{2544}{h_1 - h_2} = \frac{2544}{386} = 6.6 \text{ lb/hp} \cdot \text{h}$$

$$\text{External turbine efficiency} = \frac{\text{Rankine-cycle steam rate}}{\text{measured steam rate}} = \frac{6.6}{10.5} = 63 \text{ percent}$$

Improvements in the Rankine cycle may be obtained by raising the initial pressure and temperature. However, to avoid excessive moisture in the low-pressure stages, the increase in pressure must be accompanied by a corresponding increase in temperature. With present alloy steels the upper limit of the cycle is about 1050°F. The lower limit of the cycle depends on the maximum vacuum obtainable with the available cooling water and rarely exceeds $29^1/_4$ to $29^1/_2$ inHg. The economical limit of the cycle for a particular size of plant may be determined by a study of relative costs and savings.

The reheat cycle. The reheat cycle, which is sometimes used for large units, is similar to the Rankine cycle with the exception that the steam is reheated in one or more steps during its expansion.

The reheating may be accomplished by passing the partly expanded steam through a steam superheater, a special reheat boiler, or a heat exchanger using high-pressure live steam. The internal thermal efficiency of the cycle is calculated by totaling the available energy converted in each part of the expansion, as shown on the Mollier diagram, and dividing by the total heat supplied in the boiler, in the superheater, and in the reheat boiler or heat exchanger.

In a plant operating with a steam pressure of 1000 lb/in^2, a steam temperature of 750°F, and an exhaust pressure of 1 inHg absolute with one stage of reheating to 750°F at 175 lb/in^2 in a reheat boiler, the increase in thermal efficiency is about $7^1/_2$ percent. With two reheating stages the improvement over the straight Rankine cycle becomes approximately $10^1/_2$ percent.

The main advantage of the reheat cycle is that excessive moisture in the low-pressure stages is avoided without employing a high initial steam temperature.

The regenerative cycle. In the regenerative, or feed-heating, cycle, steam is withdrawn from the turbine at various points to supply heat to the feedwater. A considerable gain in economy may be obtained by using this cycle because the extracted steam has already given up part of its heat in doing work in the turbine and because the latent heat of the steam condensed in the feedwater heaters is conserved and returned to the boiler, thus reducing the heat loss to the condenser.

The cycle efficiency may be calculated using a method similar to that already mentioned, but in connection with this cycle it is customary to design a flow diagram and to prepare a complete heat balance of the plant. In small and medium-sized plants, one or two extraction heaters may be used in addition to the exhaust heater that serves the steam-driven auxiliaries, and in large plants up to seven heaters may be employed.

Additional plant economies result from reduced size of the condenser. From the viewpoint of steam generation, however, the load on the boiler is slightly increased to compensate for the steam extracted to the feed heaters. Furthermore, the higher temperature of the feedwater, while reducing the size of the economizer, also decreases the boiler efficiency by raising the lower level of the combustion gas cycle. This conflict between turbine and boiler cycle efficiencies may be removed by installing an air heater, which restores this lower level and permits the full benefit of the more economic method of regenerative feed heating.

Regenerative feedwater heating. The basic principles of this cycle have been discussed previously. There is an optimum temperature to which the condensate can be heated. When this limit is exceeded, the amount of work delivered by the extracted steam is reduced and the benefit to the cycle gradually diminishes. If we assume, as an example, steam conditions of 400 lb/in^2 and 750°F at the throttle and a 29-inHg vacuum, the most favorable feedwater temperature is about 240°F for one stage of feedwater heating, 290°F for two stages, 320°F for three stages, and 330°F for four stages, as shown in Fig. T-66.

As the number of heating stages is increased, the savings become proportionately less, as illustrated by the curves. For the steam conditions noted above, the cycle is improved by a maximum of 6 percent with one stage, $7^3/_4$ percent with two stages, 9 percent with three stages, and $9^3/_4$ percent with four stages. For this reason, it is not economically sound to install more than one or two heaters for a small-capacity turbine. Furthermore, the overall plant economy may limit the maximum feedwater temperature. With the condensate heated to a higher temperature because of the increased number of feed-heating stages, the temperature difference available to the economizer, usually provided in the boiler, becomes less; therefore, less heat will be extracted from the flue gases by the economizer. The resulting increase in stack loss and corresponding decrease in boiler efficiency may thus more than outweigh the improvement in the turbine cycle. The use of air preheaters instead of economizers to recover the stack loss makes it possible to obtain the full benefit from the regenerative feed-heating cycle.

Regenerative feedwater heating affects the distribution of steam flow through the turbine. The steam required to heat the feedwater is extracted from the turbine at

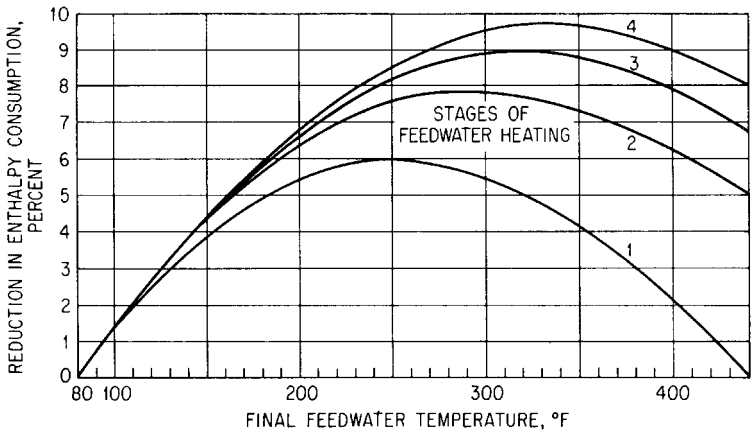


FIG. T-66 Reduction in enthalpy consumption due to regenerative feedwater heating (steam conditions: 400 lb/in^2, 750°F, 29 inHg). (Source: Demag Delaval.)

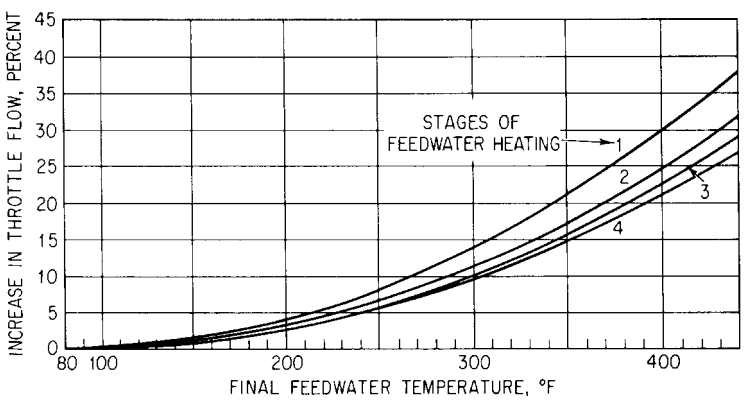


FIG. T-67 Increase in steam flow to turbine due to regenerative feedwater heating (steam conditions: 400 lb/in^2, 750°F, 29 inHg). (Source: Demag Delaval.)

various points, determined by the temperature in the corresponding feed-heating stage. The extracted steam does not complete its expansion to the vacuum at the turbine exhaust; thus somewhat less power is delivered than with straight condensing operation. To obtain equal output, the steam flow to the turbine must therefore be slightly increased, as shown in Fig. T-67, which refers to the same steam conditions as in Fig. T-66. It may be noted from Fig. T-67 that, for instance, with one stage of feedwater heating to the optimum temperature of 240°F, it is necessary to add about $7^1/_2$ percent to the throttle flow and that with two stages the increase is about $10^1/_2$ percent, etc.

On the other hand, a certain percentage of the total steam flow is extracted; thus the flow to the condenser is reduced as shown in Fig. T-68. For one and two feed-heating stages in the above example the decrease in steam flow to the condenser is about 8 and $10^1/_2$ percent, respectively, as compared with straight condensing operation. The tube surface and size of the condenser can therefore be reduced by similar amounts.

Furthermore, the redistribution of the flow benefits the turbine; the first stages, which usually operate with partial admission, can easily handle more steam

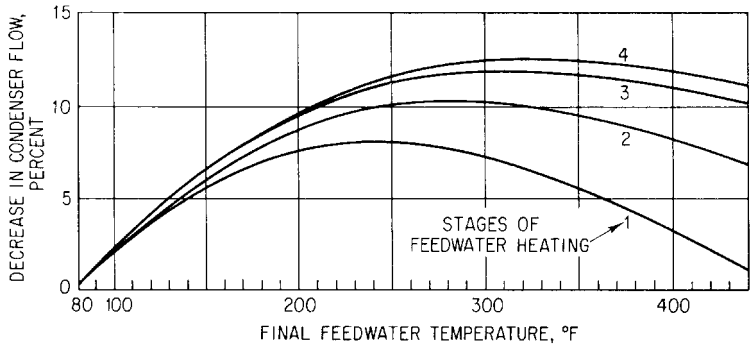


FIG. T-68 Decrease in steam flow to condenser due to regenerative feedwater heating (steam conditions: 400 lb/in², 750°F, 29 inHg). (Source: Demag Delaval.)

efficiently, and the last stage in particular will gain in efficiency, mainly because of a decrease in leaving loss resulting from less flow to the condenser.

Fuel savings of 5 to 10 percent, increasing with steam pressure and the number of heating stages and decreasing with superheat, may be obtained by the use of regenerative feedwater heating. The additional equipment is simple and inexpensive; therefore, this cycle is generally employed in preference to straight condensing operation.

Classification of turbines

To broaden the understanding of turbines and to assist in the preliminary selection of a type suitable for a proposed application, Table T-6 has been prepared. In this table the general field of application is shown, with corresponding steam and operating conditions that may be provided for in the design of the turbine.

As an example, an industrial plant may use a moderate amount of power that can be obtained at low cost from the steam required for some chemical process; in this case a condensing high-pressure turbine with single or double extraction would be selected, with steam pressure, temperature, and extraction corresponding to the desired conditions. As an alternative, a noncondensing back-pressure turbine might be considered, particularly when power and steam requirements are nearly balanced. The advantage of this type of plant is a less expensive turbine and the elimination of condensing equipment.

In recent years, the superposed, or topping, turbine has found considerable favor in large power stations and industrial plants to provide additional power or process steam and, incidentally, to improve station economy. This turbine is usually of the high-speed multistage type. Because of the small specific volume of the steam at high pressure, it becomes possible to concentrate a large amount of power in a turbine and boiler plant of relatively small physical dimensions; thus in many cases plant capacity may be greatly increased without extensions to existing buildings.

Small turbines for auxiliary drives are usually of the single-stage noncondensing type exhausting at atmospheric or slightly higher pressure into a deaerating chamber.

Turbine steam-path design

The steam turbine is a comparatively simple type of prime mover. It has only one major moving part, the rotor that carries the buckets or blades. These, with the stationary nozzles or blades, form the steam path through the turbine. The rotor is

TABLE T-6 Classification of Steam Turbines with Reference to Application and Operating Conditions

Basic Type	Operating Condition	Steam Condition	Application
Condensing	High-pressure turbine (with or without extraction for feedwater heating)	100–2400 psig; saturated, 1050°F; 1–5 inHg absolute	Drivers for electric generators, blowers, compressors, pumps, marine propulsion, etc.
	Low-pressure turbine (with high-pressure insert)	Main: 100–200 psig; 500–750°F; 1–5 inHg absolute Insert: 1450–3500 psig; 900–1050°F; 1–5 inHg absolute	Electric utility boiler-feed-pump drives
	Low-pressure bottoming turbine	Atmospheric, 100 psig; saturated, 750°F; 1–5 inHg absolute	Drivers for electric generators, blowers, compressors, pumps, etc.
	Reheat turbine	1450–3500 psig; 900–1050°F; 1–5 inHg absolute	Electric utility plants
	Automatic-extraction turbine	100–2400 psig; saturated, 1050°F; 1–5 inHg absolute	Drivers for electric generators, blowers, compressors, pumps, etc.
	Mixed-pressure (induction) turbine	100–2400 psig; saturated, 1050°F; 1–5 inHg absolute	Drivers for electric generators, blowers, compressors, pumps, etc.
	Cross-compound turbine (with or without extraction for feedwater heating, with or without reheat)	400–1450 psig; 750–1050°F; 1–5 inHg absolute	Marine propulsion
Noncondensing	Straight-through turbine	600–3500 psig; 600–1050°F; atmospheric, 1000 psig	Drivers for electric generators, blowers, compressors, pumps, etc.
	Automatic-extraction turbine	600–3500 psig; 600–1050°F; atmospheric, 600 psig	Drivers for electric generators, blowers, compressors

supported on journal bearings and axially positioned by a thrust bearing. A housing with steam inlet and outlet connections surrounds the rotating parts and serves as a frame for the unit.

However, a great number of factors enter into the design of a modern turbine, and its present perfection is the result of many years of research and development. While the design procedure may be studied in books treating this particular subject, a short review of the main principles may serve to compare the various types. This will aid in the selection and evaluation of turbines suitable for specific requirements.

In considering the method of energy conversion, two main types of blading, impulse and reaction, are employed. An impulse stage consists of one or more stationary nozzles in which the steam expands, transforming heat energy into velocity or kinetic energy, and one or more rows of rotating buckets that transform the kinetic energy of the steam into power delivered by the shaft. In a true impulse stage the full expansion of the steam takes place in the nozzle. Hence, no pressure drop occurs while the steam passes through the buckets.

A reaction stage consists of two elements. There is a stationary row of blades in which part of the expansion of the steam takes place and a moving row in which the pressure drop of the stage is completed.

Many turbines employ both impulse and reaction stages to obtain the inherent advantages of each type.

Figure T-69 illustrates some of the most common types of nozzle and blade combinations used in present turbines. Four of the diagrams, *a*, *b*, *c*, and *d*, apply

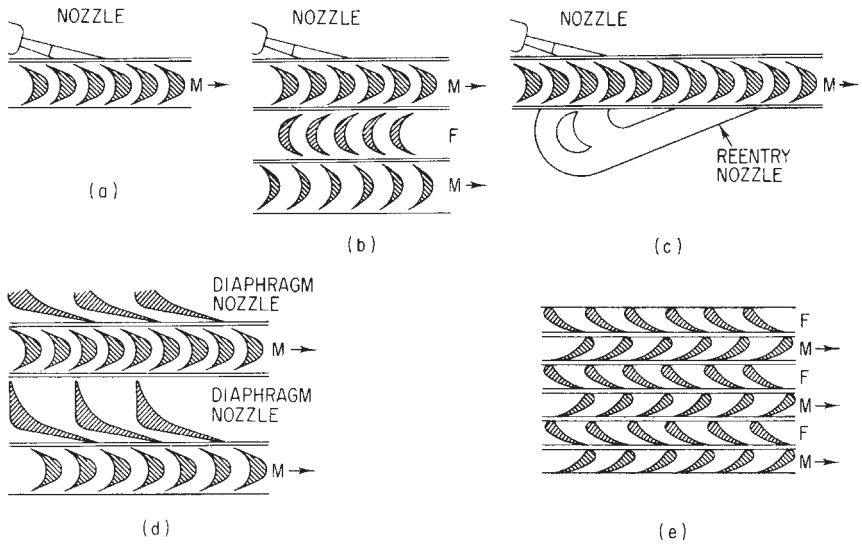


FIG. T-69 Main types of turbine blading (*F* = fixed row; *M* = moving row). (*a*) Impulse turbine: single velocity stage. (*b*) Impulse turbine: two velocity stages. (*c*) Reentry impulse turbine: two velocity stages. (*d*) Impulse turbine: multistage. (*e*) Reaction turbine: multistage. (Source: Demag Delaval.)

to the impulse principle, as noted in the legend, and the last one, *e*, shows a type of reaction blading. A constructional difference may also be pointed out: impulse buckets are usually carried on separate discs with nozzles provided in stationary partitions called diaphragms, while the moving reaction blades are generally supported on a rotor drum with the stationary blades mounted in a casing.

The impulse stage has a definite advantage over the reaction stage in handling steam with small specific volume as in the high-pressure end of a turbine or in cases in which the enthalpy drop per stage is great; thus small single-stage turbines are always of the impulse type. The stage may be designed for partial admission with the nozzles covering only a part of the full circumference; therefore, the diameter of the wheel may be chosen independently of the bucket height. Used as a first stage in a multistage turbine, the impulse stage with partial admission permits adjustment of the nozzle area by arranging the nozzles in separate groups under governor control, thus improving partial-load performance.

The dominating principle in turbine design involves expression of the efficiency of the energy conversion in nozzles and buckets or in reaction blades, usually referred to as stage efficiency, as a functon of the ratio u/C. The blade speed u, feet per second, is calculated from the pitch diameter of the nozzle and thus determines the size of the wheel at a given number of revolutions per minute and C, also in feet per second, is the theoretical velocity of the steam corresponding to the isentropic enthalpy drop in the stage, expressed by the formula

$$C = 223.8\sqrt{\text{Btu}}$$

Figure T-70 illustrates average stage efficiencies that may be attained in various types of turbines operating at design conditions. The losses that are represented in the stage-efficiency curves are due to friction, eddies, and flow interruptions in the steam path, plus the kinetic energy of the steam as it leaves a row of blades.

Part of the latter loss can be recovered in the following stage. Additional losses not accounted for in the stage-efficiency curves are due to windage and friction of the rotating parts and to steam leakage from stage to stage. With the exception of

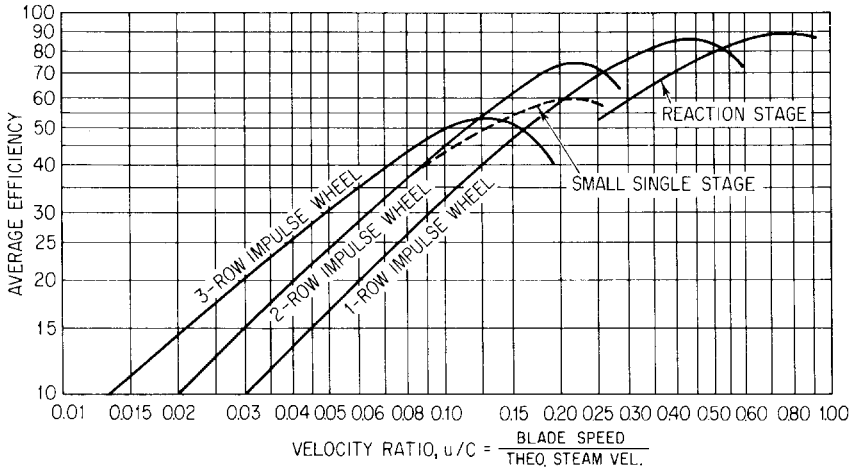


FIG. T-70 Average efficiency of turbine stages. (Source: Demag Delaval.)

the kinetic energy that may be recovered, all losses are converted to heat with a corresponding increase in the entropy of the steam.

From the group of curves of Fig. T-70 it follows that the maximum combined efficiency for various types of stages is attained at different velocity ratios. This ratio is highest for reaction stages and lowest for three-row impulse wheels. This implies that for equal pitch-line speeds the theoretical steam velocity or the stage enthalpy drop must be lowest for reaction stages and highest for three-row wheels to maintain the maximum possible efficiency. At this maximum efficiency, the three-row wheel can work with many times the steam velocity and a correspondingly larger enthalpy drop compared with a reaction stage.

The maximum efficiency of reaction stages may exceed 90 percent at a velocity ratio of 0.75, as shown in Fig. T-70. However, such values can be attained only with a great number of stages. Hence, reaction stages are normally not designed for a higher velocity ratio than 0.65. A section of reaction blading is shown in Fig. T-69*e*.

Single-row impulse stages have a maximum efficiency of about 86 percent at a velocity ratio of 0.45. Figure T-69*a* shows a combination of impulse buckets with an expanding nozzle, and Fig. T-69*d* shows multistage impulse blading with nonexpanding nozzles.

Let us assume, as an example, a blade speed of 500 ft/s, corresponding to a turbine wheel with 32-in pitch diameter operating at a speed of 3600 rpm; the optimum steam velocity would be 500/0.45 = 1100 ft/s. The kinetic energy of the steam may be expressed in Btu by the relation Btu = $(C/223.8)^2$ = $1100^2/50{,}000$ = 24; thus the enthalpy drop utilized per stage at the point of maximum efficiency is about 24 Btu for the above condition.

In the case of a turbine operating at high steam pressure and temperature, exhausting at low vacuum, the available energy may be approximately 500 Btu; therefore, about 20 single-row impulse stages would be required for maximum efficiency. Obviously the pitch diameter of the wheels cannot be chosen arbitrarily, but this example illustrates the method of dividing the energy in a number of steps called pressure stages. The turbine would be classified as a multistage impulse turbine.

Figure T-70 further shows one curve labeled “two-row” with an extension in a broken line referring to small single-stage turbines and one curve marked “three-row impulse wheel.” These refer to so-called velocity-compounded stages as

illustrated by Fig. T-69*b* and *c*. The purpose of the two- and three-row and also the reentry stage is to utilize a much greater enthalpy drop per stage than that possible in a single-row impulse stage. When the enthalpy drop per stage is increased, the velocity ratio is reduced and the kinetic energy is only partly converted into work in the first row of revolving buckets; thus the steam leaves with high residual velocity. By means of stationary guide buckets the steam is then redirected into a second, and sometimes a third, row of moving buckets, where the energy conversion is completed.

In the so-called helical-flow stage, with semicircular buckets milled into the rim of the wheel, and also in the reentry stage shown in Fig. T-69*c*, only one row of revolving buckets is used. This type of velocity compounding is sometimes employed in noncondensing single-stage auxiliary turbines.

The curve marked "two-row impulse wheel" indicates that a maximum stage efficiency of about 75 percent may be attained at a velocity ratio of approximately 0.225. At this condition, the two-row velocity-compounded stage will utilize about 4 times as much energy as a single-row impulse stage. When we compare the efficiencies on the basis of operating conditions as defined by the velocity ratio, it appears from the curves that the two-row wheel has a higher efficiency than a single-row wheel when the velocity ratio is less than 0.27.

Occasionally, in small auxiliary turbines operating at a low-speed ratio, a three-row stage may be used. The curve marked "three-row" indicates a maximum efficiency of about 53 percent at a speed ratio of about 0.125. Apparently, at this point the efficiency of a two-row wheel is almost as good; thus the three-row stage would be justified only at still lower-speed ratios, that is, for low-speed applications.

The design of a turbine, especially of the multistage type, involves a great many factors that must be evaluated and considered. A detailed study of the steam path must be made, and various frictional and leakage losses that tend to decrease the efficiency, as well as compensating factors such as reheat and carryover, must be computed and accounted for in the final analysis of the performance of the turbine. Stresses must be calculated to permit correct proportioning of the component parts of the turbine, and materials suitable for the various requirements must be selected.

Single-stage turbines

Single-stage turbines, sometimes called *mechanical-drive* or *general-purpose turbines*, are usually designed to operate noncondensing or against a moderate back pressure. The principal use of these turbines is to drive power plant and marine auxiliaries such as centrifugal pumps, fans, blowers, and small generator sets. They may also be applied as prime movers in industrial plants, and in many cases small turbines are installed as standby units to provide protection in case of interruption of the electric power supply.

They are built in sizes up to 1500 hp and may be obtained in standardized frames up to 1000 hp with wheel diameters from 12 to 36 in. Rotational speeds vary from 600 to 7200 rpm or higher; the lower speeds apply to the larger wheel sizes used with direct-connected turbines, and the higher speeds are favored in geared units. The bucket speed usually falls between 250 and 450 ft/s in direct-connected turbines operating at 3600 rpm and may exceed 600 ft/s in geared turbines.

The efficiency of a turbine generally improves with increasing bucket speed as noted by referring to efficiency versus velocity ratio curves in Fig. T-70; thus it would seem that both high revolutions and large diameters might be desirable. However, for a constant number of revolutions per minute the rotation loss of the disc and the buckets varies roughly as the fifth power of the wheel diameter and for a constant bucket speed almost as the square of the diameter. Thus, in direct-

connected turbines with the speed fixed by the driven unit, the rotation losses may become the dominating factor in selecting the wheel size for maximum efficiency. On the other hand, when reduction gears are adopted, the velocity ratio may be increased by means of higher revolutions, sometimes even with smaller wheel diameter; thus considerably higher efficiencies may be expected, as shown by the dashed curve in Fig. T-67. Since the rotation losses vary approximately in direct relation to the density of the steam surrounding the wheel, it follows that small wheel diameters should be used particularly for operation at high back pressure.

Turbine manufacturers have complete test data on standard sizes of small turbines on which steam-rate guarantees are based. Knowing the characteristics of different turbines, they are in a position to offer suggestions regarding the most suitable type and size to choose for specific requirements.

The single-stage turbine is simple and rugged and can be depended on to furnish many years of service with a minimum of maintenance expense. The few parts that may require renewal after long periods of operation, for instance, bearings, carbon rings, and possibly valve parts, are inexpensive and easy to install. It is also comparatively simple to exchange the steam nozzles to suit different steam conditions, as sometimes encountered in connection with modernization of old plants, or to adapt the turbine to new conditions due to changes in process-steam requirements.

Steam-rate calculations. Approximate steam rates of small single-stage turbines (less than 500 hp) may be computed by the following general method:

1. The available energy, $h_1 - h_2 = H_a$, at the specified steam condition is obtained from the Mollier diagram.
2. Deductions are made for pressure drop through the governor valve (12.5 Btu), loss due to supersaturation C_s (about 0.95), and 2 percent margin (0.98). The remaining enthalpy drop is called net available energy H_n.
3. The theoretical steam velocity C, ft/s, is calculated, based on net available energy H_n. The formula for steam velocity is $C = 223.8 \times \sqrt{H_n}$.
4. The bucket speed u, ft/s, is calculated from the pitch diameter, in (of the nozzles), and the rpm.
5. The velocity ratio u/C is calculated and the "basic" turbine efficiency E is obtained from an actual test curve similar to those given in Fig. T-70.
6. The "basic" steam rate for the turbine is calculated from the formula

$$\text{Basic steam rate} = \frac{2544}{H_n E} = \text{lb/hp} \cdot \text{h}$$

7. The loss horsepower for the specific turbine size is estimated from Fig. T-71, corrected for back pressure as noted on the diagram.
8. The actual steam rate of the turbine at the specified conditions is

$$\text{Basic steam rate} \times \frac{\text{rated hp} + \text{loss hp}}{\text{rated hp}} = \text{lb/hp} \cdot \text{h}$$

Example: As a matter of comparison with the short method of estimating turbine performance, the same example of a 500-hp turbine with a steam condition of 300 lb/in^2, 100°F superheat, and 10 lb/in^2 back pressure at a speed of 3600 rpm may be selected. It is further assumed that a frame size with a 24-in-pitch-diameter two-row wheel is used.

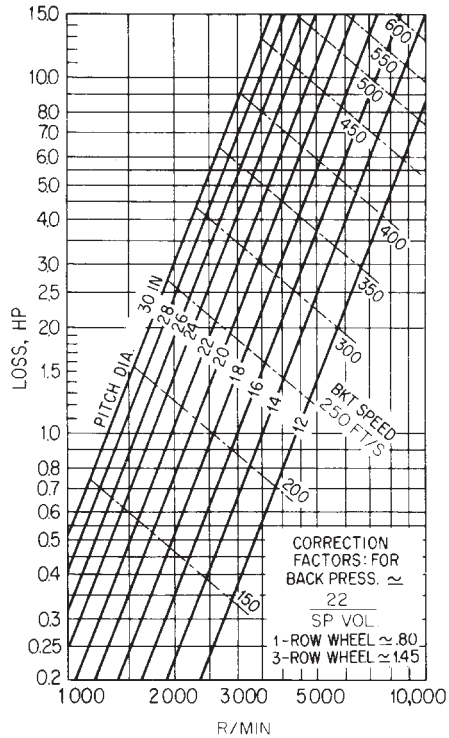


FIG. T-71 Rotational loss, average for single-stage turbines (two-row wheel; atmosphere exhaust). (Source: Demag Delaval.)

The available energy is 205 Btu; subtracting a 12.5-Btu drop through the governor valve leaves 192.5 net Btu, which corresponds to a theoretical steam velocity $C = 223.8 \times \sqrt{192.5} = 3104$ ft/s.

The bucket speed $u = 3600 \times 24 \times \pi/60 \times 12 = 377$ ft/s. Thus the velocity ratio $u/C = 377/3104 = 0.12$. From Fig. T-70 the approximate efficiency 0.47 is obtained on the curve marked "two-row impulse wheel" at $u/C = 0.12$.

The supersaturation loss factor C_s (due to the expansion of the steam into the supersaturation state) is a function increasing with the initial superheat and decreasing with the available enthalpy, in this case about 0.96; a margin of 2 percent may also be included, thus the

$$\text{Basic steam rate} = \frac{2544}{192.5 \times 0.47 \times 0.96 \times 0.98} = 30.0 \text{ lb/hp} \cdot \text{h}$$

The rotational loss of a 24-in-pitch-diameter wheel at 3600 rpm, determined from Fig. T-71, is about 6.3 hp. This diagram is based on atmospheric exhaust pressure; therefore, a correction factor must be applied as noted. At 10-lb back pressure the specific volume of the steam is about 16.3 ft^3/lb. Thus

$$\text{Loss hp} = 6.3 \times \frac{22}{16.3} = 8.5$$

$$\text{Steam rate of turbine} = 30.0 \times \frac{500 + 8.5}{500} = 30.5 \text{ lb/hp} \cdot \text{h}$$

TABLE T-7

Small units	150 to 400 lb/in^2; 500 to 750°F
Medium units	400 to 600 lb/in^2; 750 to 825°F
Large units	600 to 900 lb/in^2; 750 to 900°F
Large units	900 to 3500 lb/in^2; 825 to 1050°/F

The use of the short method and Fig. T-67 results in this case in a steam rate of 31.4 lb/hp · h, which is about 3 percent higher than that obtained by calculations applying Figs. T-70 and T-71; both methods are consistent and may serve the purpose for which they are suggested.

Multistage condensing turbines

The most important application of the steam turbine is that of serving as prime mover to drive generators, blast-furnace blowers, centrifugal compressors, pumps, etc., and for ship propulsion. Since the economic production of power is the main objective, these turbines are generally of the multistage type, designed for condensing operation, i.e., the exhaust steam from the turbine passes into a condenser, in which a high vacuum is maintained.

The dominating factor affecting the economy, which may be expressed in terms of station heat rate or fuel consumption, is the selection of the steam cycle and its range of operating conditions, as previously discussed in connection with turbine cycles. For smaller units the straight condensing Rankine cycle may be used; for medium and large turbines the feed-heating, regenerative cycle is preferred; and in large base-load stations a combination of a reheating, regenerative cycle may offer important advantages.

If we assume average economic considerations, such as capacity of the plant and size of the individual units, load characteristics, and amount of investment, the initial steam conditions may be found to vary approximately as shown in Table T-7.

Similar conditions may prevail with reference to the vacuum; smaller units may operate at 26 to 28 inHg in connection with spray ponds or cooling towers, while larger turbines usually carry 28 to 29 inHg and require a large supply of cooling water.

These general specifications are equivalent to an available enthalpy drop varying from about 350 Btu to a maximum of about 600 Btu. Therefore, the modern condensing turbine must be built to handle a large enthalpy drop; hence a comparatively large number of stages is required to obtain a high velocity ratio consistent with high efficiency, as indicated in Fig. T-70. Incidentally, the average efficiency curves of condensing multistage turbines in the lower part of Fig. T-66 cover a range from 363 Btu at 200 lb/in^2 to 480 Btu at 1500 lb/in^2.

As shown in Fig. T-72, the overall efficiency of multistage turbines is sometimes expressed as a function of the so-called quality factor, which serves as a convenient criterion of the whole turbine in the same manner as the velocity ratio applies to each stage separately. The quality factor is the sum of the squares of the pitch-line velocity of each revolving row divided by the total isentropic enthalpy drop. The pitch-line velocity is expressed in feet per second and the enthalpy drop in Btu.

The curve is empirical, determined from tests of fairly large turbines, and indicates average performance at the turbine coupling. It may be used to evaluate preliminary designs with alternative values of speed, wheel diameters, and number of stages or to compare actual turbines when pertinent information is available. To

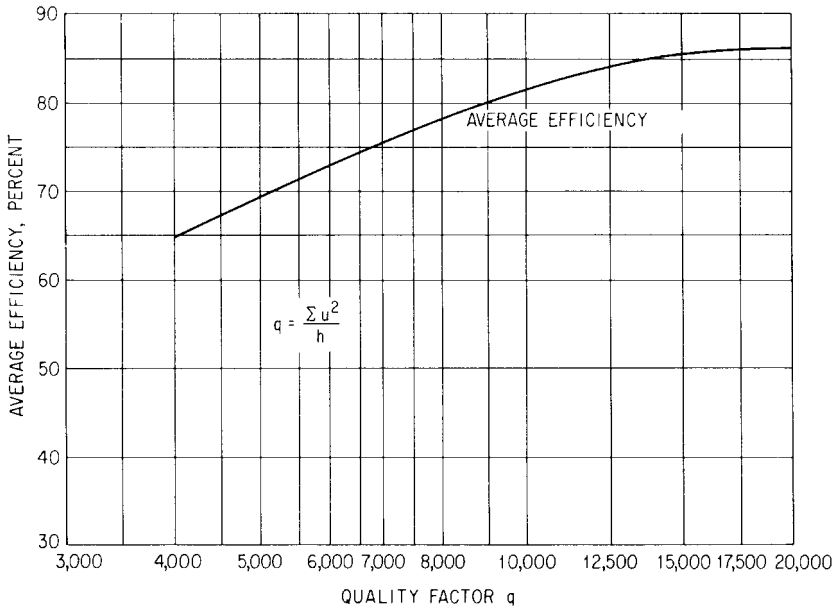


FIG. T-72 Average efficiency of multistage turbines on the basis of the quality factor. (Source: Demag Delaval.)

obtain consistent results the size and type of the turbine must be considered; generally, the internal efficiency improves appreciably with increased volume flow, and the mechanical efficiency also improves slightly with increased capacity, thus a size factor should be applied to the efficiency curve to correlate units of different capacity, or individual efficiency curves based on tests may be used for each standard size.

Example: Determine provisional dimensions of a 3000-hp 3600-rpm condensing turbine operating at 400 lb/in^2, 750°F, and 28 inHg. A turbine efficiency of 73 percent is desired; thus, for a size factor of, say, 95 percent, the required efficiency is 77 percent, corresponding to a quality factor of about 7500. The available enthalpy is 460 Btu; consequently the sum of velocity squares is $7500 \times 460 = 3{,}450{,}000$. Various combinations of bucket speed and number of moving rows may be selected; for instance, a bucket speed of 500 ft/s corresponding to a pitch diameter of about 32 in would require 14 rows of buckets; 475 ft/s equals $30^1/_4$-in diameter with 15 rows, etc.

The pitch diameter usually increases gradually toward the exhaust end; therefore, the so-called root-mean-square diameter is used in these calculations. In this example the diameters would be adjusted in relation to the flow path through the turbine and the number of stages, perhaps 14, resulting in the most satisfactory bucket dimensions and in general compactness of design. This discussion illustrates the general principle of the interdependence of diameters and number of stages for a required turbine efficiency.

In analyzing the design of a condensing turbine as shown in Fig. T-73, the first stages must be suitable for steam with comparatively high pressure, high temperature, and small specific volume. The last stage, on the other hand, presents the problem of providing sufficient area to accommodate a large-volume flow of low-pressure steam. Taking a large enthalpy drop in the first stage by means of a two-row velocity stage as shown in this particular case results in a moderate first-stage pressure with low windage and gland leakage losses. Furthermore, the remaining

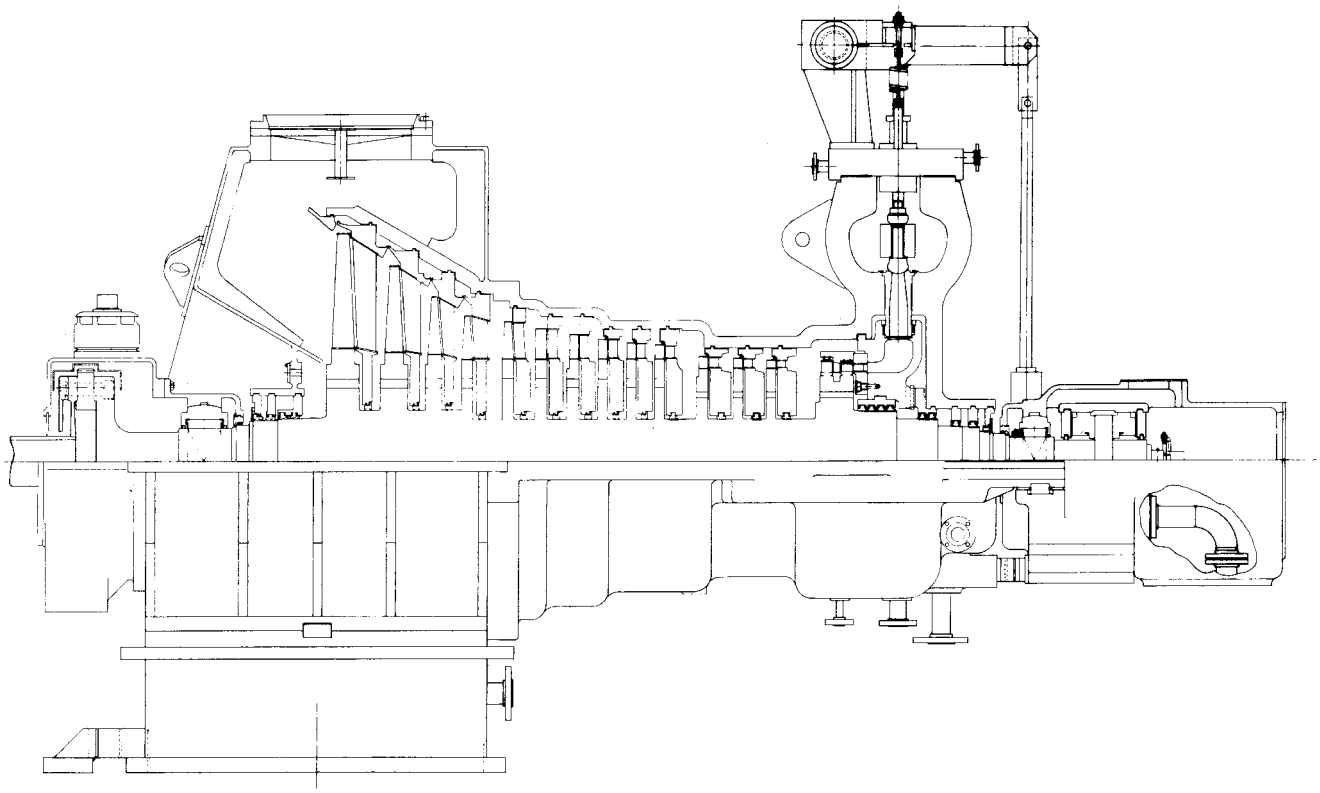


FIG. T-73 Multistage condensing turbine (56,000 kW, 3600 rpm, 1250 psig, 950°F, 2.5 inHg absolute). (Source: Demag Delaval.)

enthalpy drop, allotted to the following stages, also decreases; i.e., the velocity ratio improves, and thus a good overall turbine efficiency results from this combination.

Extraction points for feed heating may be located in one or more stages as required, and provision may also be made to return leakage steam from the high-pressure gland to an appropriate stage, thus partly recovering this loss by work done in succeeding stages.

The journal bearings are of the tilting-pad type with babbitt-lined steel pads. They are made in two halves and arranged for forced-feed lubrication. Thus turbine-shaft seals are of the stepped-labyrinth type, with the labyrinths flexibly mounted.

The turbine casing is divided horizontally with the diaphragms also made in two halves, the upper ones being dismountable with the top casing. The turbine support is arranged to maintain alignment at all times. The turbine is anchored at the exhaust end, and the casing is permitted to expand freely with changes in temperature.

Group nozzle control, operated from a speed governor by a hydraulic servo motor, results in economic partial-load performance combined with desirable speed-governing characteristics.

This condensing turbine represents a logical application of design principles to obtain maximum efficiency by the proper selection of wheel diameters and number of stages and by proportioning the steam path to accommodate the volume flow of steam through the turbine.

Superposed and back-pressure turbines

Superposed and back-pressure turbines operate at exhaust pressures considerably higher than atmospheric and thus belong to the general classification of

noncondensing turbines. Relatively high efficiency is required; therefore, these turbines are of the multistage type. The small single-stage auxiliary turbines previously described are also of the noncondensing type, but of a much simpler design, suitable for less exacting steam conditions.

The main application of superposed turbines, often referred to as *topping turbines*, is to furnish additional power and to improve the economy of existing plants. Since boilers usually fail or become obsolete long before the turbines they serve, it has proved economically sound in many plants to replace old boilers with modern high-pressure, high-temperature boilers supplying steam to a new superposed turbine with its generator. The superposed turbine may be an extracting unit supplying such steam to process and its exhaust steam to the existing condensing turbines operating at the same inlet conditions as before. A considerable increase in plant capacity and improvement in station economy is thus obtained with a comparatively small additional investment.

Superposed turbines have been built in sizes of 500 kW and above. The initial steam conditions may vary from 600 to 2000 lb/in^2 with steam temperatures from 600 to 1050°F; the exhaust pressure may range from 200 to 600 lb/in^2 and must correspond to the initial pressure of the existing plant. Topping units are usually arranged to serve a group of turbines but may also be proportioned for individual units.

Investigations in connection with proposed topping units may cover various aspects, for instance, determination of additional capacity obtainable with assumed initial steam conditions or, conversely, selection of initial steam conditions for a desired increase in power. Incidentally, the improvement in station heat rate is also calculated for use in evaluating the return on the proposed investment. However, this evaluation involves heat-balance calculations for the complete plant including the feed-heating cycle adjusted to the new conditions.

To indicate the possibilities of the superposed turbine the following example is suggested. An existing plant of 5000-kW rated capacity is operating at 200 lb/in^2, 500°F, and $1^1/_2$ inHg absolute condenser pressure. If we assume a full-load steam rate of 13.0 lb/kWh based on two 2500-kW units, the total steam flow is about 65,000 lb/h. Determine the additional power to be expected from a topping unit operating at 850 lb/in^2, 750°F initial steam condition at the turbine throttle, and exhausting into the present steam main.

The available energy of the high-pressure steam is 147 Btu, corresponding to a theoretical steam rate of 23.2 lb/kWh. If we assume a generator efficiency of about 94 percent and a "noncondensing" turbine efficiency of 63 percent, approximated from the curve sheet in Fig. T-66, the steam rate becomes about 39 lb/kWh. Incidentally, the enthalpy at the turbine exhaust, calculated from the efficiency, is about 1272 Btu; according to the Mollier diagram, this corresponds to about 508°F at 215 psia; thus the initial steam temperature of 750°F selected for the topping unit matches approximately the 500°F assumed at the existing steam header.

Based on a total steam flow of 65,000 lb/h and a steam rate of 39 lb/kWh, the increase in power is about 1665 kW at the full-load condition. Thus the increase in capacity is 33.3 percent; likewise, the combined turbine steam rate is 9.75 lb/kWh, an improvement of 25 percent. To calculate the corresponding fuel saving, additional data for the boiler and plant auxiliaries would be required.

The approximate size of the unit may be arrived at by the quality-factor method referred to in Fig. T-72. By applying an appropriate-size factor, the topping turbine may in this case be designed for an efficiency of, say, 67 percent, corresponding to a quality factor of about 4500. With an available enthalpy drop of 147 Btu the sum of the velocity squares is 660,000. Because of the comparatively small volume flow and the high density of the steam, small wheel diameters are used; thus the bucket speed is rather low. If we assume, for instance, 350 ft/s, corresponding to about

$22^1/_2$-in pitch diameter at 3600 rpm, the number of stages required would be about 5; and at 300 ft/s with 19-in pitch diameter the number of stages would be 7, etc. (Provisional inlet and outlet connections can be determined from Fig. T-89, thus indicating the general overall dimensions of the turbine.)

Back-pressure turbines, frequently of fairly large capacity, are often installed in industrial plants where a large amount of process steam may be required. In this case, the electric power required to operate the plant may be obtained from the process steam as a by-product at very low cost. Since good economy is important, these turbines are generally of the multistage type. The usual range of initial pressure is from 200 to 900 lb/in^2 with corresponding steam temperatures from 500 to 900°F. The back pressure, which depends on the requirements of the process steam, may fall between the limits of 5 and 150 lb/in^2.

The approach to the problem is to estimate the amount of power that can be obtained from the process steam with various initial steam conditions. In this manner a balance between available steam and power demand is determined, and as a preliminary step the appropriate initial steam condition is selected. A check on the enthalpy at the turbine exhaust then indicates possible adjustment of the initial steam temperature to obtain approximately dry steam at the point where the process steam is used. Occasionally, heavy demands for steam in excess of the power load may be provided for by supplying the additional steam through a reducing valve directly from the boilers. Supplementary power for peak loads may be obtained from an outside source or from a condensing unit.

Extraction and induction turbines

Many industrial plants requiring various quantities of process steam combined with a certain electric power load make use of extraction turbines. It is possible to adapt the extraction turbine to a great variety of plant conditions, and many different types are built, among them noncondensing and condensing extraction turbines with one or more extraction points and automatic and nonautomatic extraction; additionally, in certain urban areas, extraction turbines are used by the utility company to supply steam to buildings in the neighborhood of the plant.

A related type of turbine, the so-called mixed-flow or induction turbine, with provision for the use of high-pressure and low-pressure steam in proportion to the available supply, may also be mentioned in this connection. Generally, the low-pressure steam is expected to carry normal load, and high-pressure steam is admitted only in case of a deficiency of low-pressure steam. Even in case of complete failure of the low-pressure supply the turbine may be designed to carry the load with good economy on high-pressure steam alone.

The most frequently used extraction turbine is the single automatic-extraction condensing turbine as shown in Fig. T-74. For design purposes it may be considered as a noncondensing and a condensing turbine, operating in series and built into a single casing. Because of the emphasis placed on compactness and comparatively simple construction, the number of stages is usually limited. The performance may therefore not be quite equal to the combined performance of a corresponding back-pressure turbine and a straight condensing turbine built in two separate units. On the other hand, the price of the extraction turbine is also less than the total price of two independent units.

Guarantees of steam rate for condensing and noncondensing automatic-extraction turbines are always made on a straight condensing or a straight noncondensing performance, respectively, obtained with no extraction but with the extraction valve wide open, that is, not functioning to maintain the extraction pressure. This nonextraction performance guaranteed for an automatic-extraction

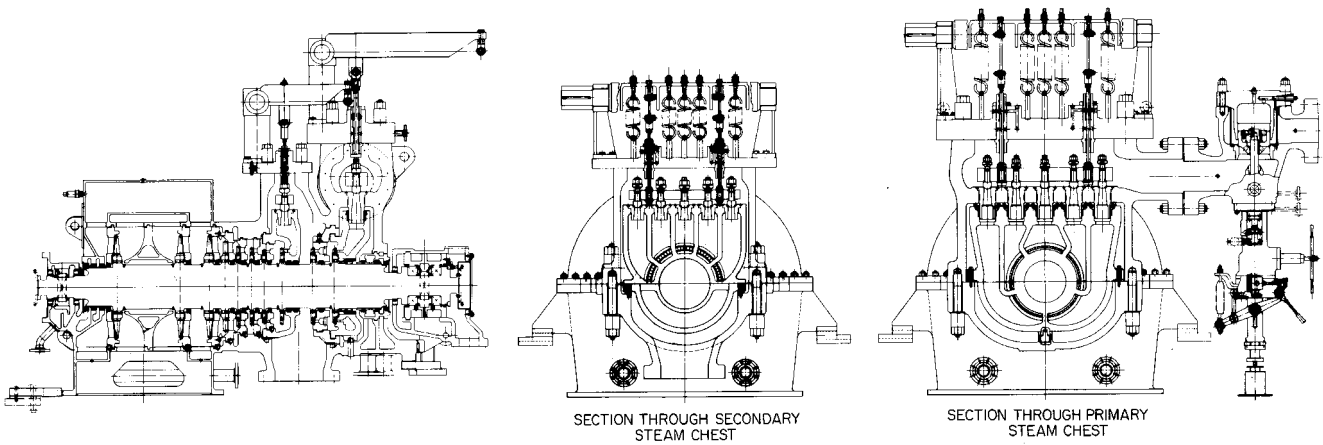


FIG. T-74 Single automatic-extraction turbine (20,000 bhp, 10,600 rpm, 1500 psig, 800°F, 2 inHg absolute, automatic extraction at 400 psig). (Source: Demag Delaval.)

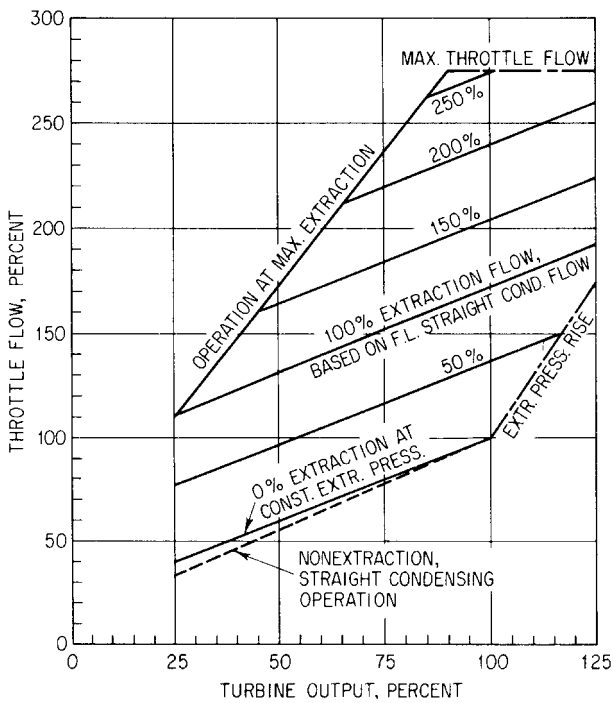


FIG. T-75 Throttle flow versus output of condensing automatic-extraction turbine. (Source: Demag Delaval.)

turbine will not differ much from that for a straight condensing or a noncondensing unit of the same capacity and designed for the same steam conditions.

The complete performance of an extraction turbine can be represented by a diagram such as Fig. T-75 in which the output is expressed in percentage of rated capacity and the throttle flow in percentage of that at full load without extraction. The line labeled "0% extraction at const. extr. press." represents the performance of the turbine when no steam is extracted but with the extraction valve acting to hold extraction pressure at the bleed connection.

The guaranteed steam flow for nonextraction, with the pressure at the bleed point varying with the load, that is, with the extraction valve wide open, is also plotted as a broken line on Fig. T-75. This line intersects the zero-extraction line at full load, while at partial loads the throttle flow for nonextraction is less than for zero extraction. The reason for this is that the low-pressure end of the turbine has been designed for the steam flow that at full load, nonextraction, with the extraction valve wide open, will give the extraction pressure required. If the steam flow through the low-pressure end of the turbine is decreased, as at partial loads, the absolute pressure at the extraction point would decrease in proportion to the steam flow if it were not for the action of the extraction valve, which throttles the steam to maintain the required extraction pressure. This throttling loss occurs when operating with zero extraction, but not when operating at nonextraction.

When steam is extracted from a turbine carrying a given load, the throttle flow must increase, but the increase is not equal to the amount extracted. For a given turbine and set of steam conditions, the increase in throttle steam over that required for zero extraction will bear nearly a constant ratio to the amount extracted. This ratio is called the *extraction factor*. As the extraction pressure is raised from exhaust pressure to inlet pressure by extracting at points of progressively higher pressure, the extraction factor increases from 0 to 1.

The line labeled "operation at max. extraction" represents the performance when all steam entering the throttle, except the cooling steam, is extracted. The line "max. throttle flow" represents the maximum flow that the high-pressure section can pass when the turbine is operated with its normal steam conditions. The corresponding limit for the low-pressure section is the one titled "extr. press. rise." The turbine can operate in the region to the right of this limit but will not then maintain normal extraction pressure. For any given load the flow to exhaust is maximum at zero extraction, so that the maximum flow through the exhaust section for which the turbine must be proportioned is determined by the maximum load to be carried with minimum extraction.

Similar diagrams may be constructed to apply to other combinations such as double automatic and mixed-flow turbines. As an example, lines of "constant induction flow" would be located below and parallel to a line of "zero induction flow" in the case of mixed-pressure or induction turbines.

Low-pressure turbines (with high-pressure insert)

Electric utility boiler-feed pumps require large blocks of power that can be most economically supplied by a steam-turbine driver. Such a unit is illustrated in Fig. T-76. See also Fig. T-77.

Normal operation is with low-pressure steam extracted from the main turbine driving the generator. The steam chest for this steam is in the upper half of the casing. Operation at low power output, i.e., somewhat less than 50 percent, causes extraction steam pressure from the main turbine to decrease until there is an insufficient supply to drive the pump. At this point, full boiler-pressure steam is admitted through the high-pressure insert located in the lower half of the casing. As the plant load is decreased further, a point is reached when the extraction steam pressure is too low and the nonreturn valves close to prevent a backflow through the low-pressure steam chest into the main turbine.

Calculation methods for sizing a feedwater pump and its turbine driver are readily available for interested persons but are somewhat beyond the scope of this handbook.

Turbine governors

The governor is the "brains" behind the "brawn" of the turbine. The governor may sense or measure a single quantity such as turbine speed, inlet, extraction,

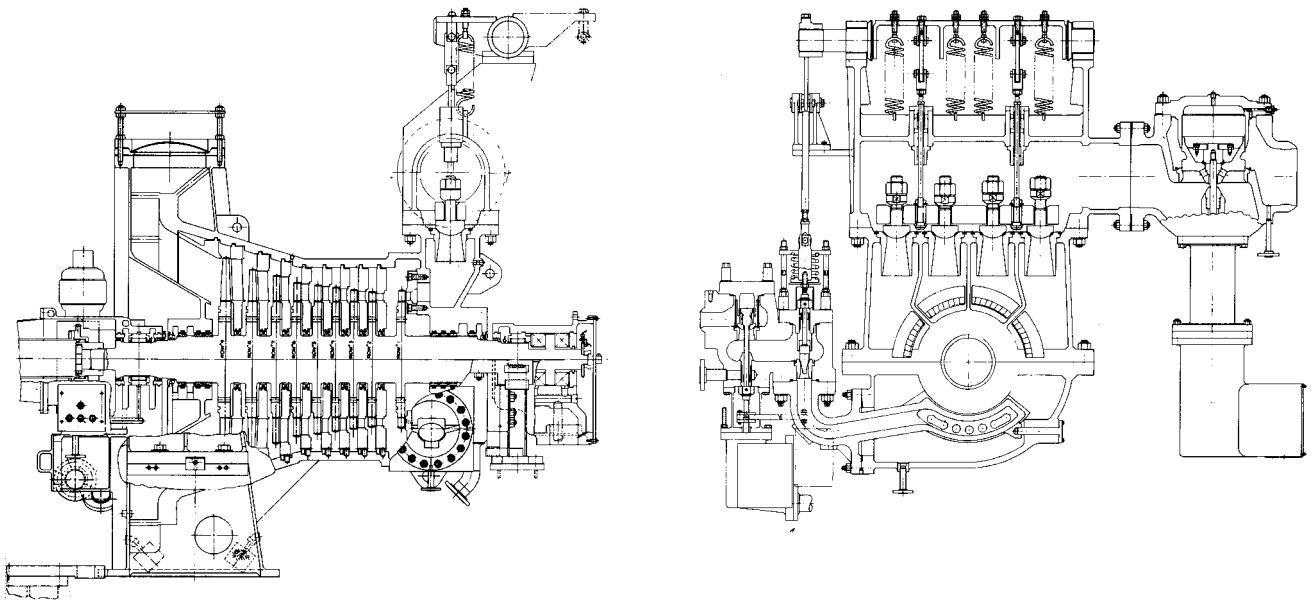


FIG. T-76 Low-pressure turbine with high-pressure insert (10,000 bhp, 5200 rpm, 105 psig, 623°F, 3 inHg absolute). (Source: Demag Delaval.)

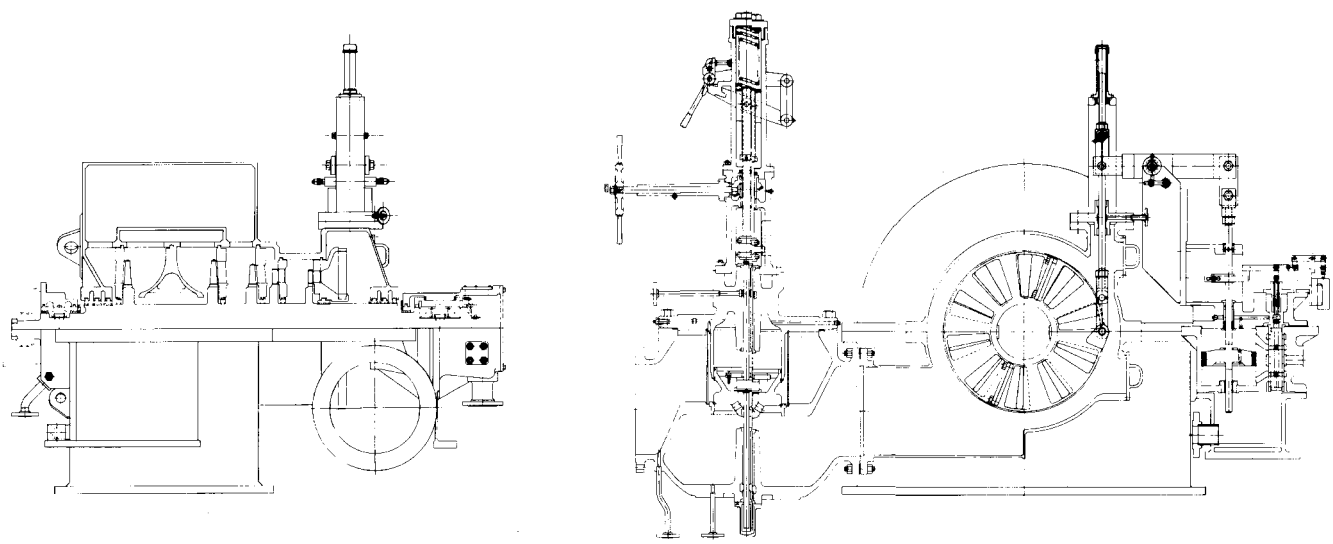


FIG. T-77 Low-pressure bottoming turbine (9000 bhp, 8700 rpm, 45 psig, 375°F, 3.5 inHg absolute). (Source: Demag Delaval.)

induction, or exhaust pressure, or any combination of these quantities and then control the turbine to regulate the quantities measured. Shaft-speed governors are the most common. A simple speed governor will first be considered.

Mechanical governors. In the direct-acting mechanical governor shown in Fig. T-78 speed is measured by spring-loaded rotating weights. As the weights are rotated, they generate a force proportional to the product of their mass, the radius of their rotation, and the square of their speed of rotation. Under steady-state conditions the weight force is balanced by the opposing force of the weight spring, and the governor stem remains stationary.

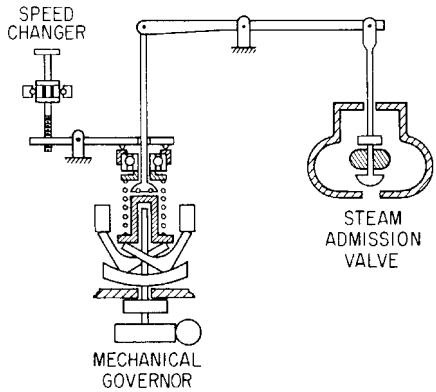


FIG. T-78 Direct-acting mechanical speed governor. (Source: Demag Delaval.)

If some load is removed from the turbine, the turbine would speed up and the governor weights would move outward. As the governor weights move outward, their force is further increased, but the force of the weight spring increases even faster and soon limits the travel of the weights. The movement of the weights is transferred through the governor stem and connecting linkage to the turbine control valve to reduce the flow of steam to the turbine, limiting the turbine-speed increase.

If some load is added to the turbine, the turbine will slow down and the governor weights will move inward. As the governor weights move inward, their force is further decreased, but the force of the weight spring decreases even faster and limits the travel of the weights. The movement of the weights is transferred through the governor stem and connecting linkage to the turbine control valve to increase the flow of steam to the turbine, limiting the turbine-speed decrease.

For any constant setting of the weight spring a certain change in speed is required to provide a full travel of the governor weights. This change in speed between the full-load and no-load speed of the governor is called either the *governor droop* or, when expressed as a percentage of the full-load speed, the *governor regulation*. When units are operated in parallel, any changes in the total load will be shared by the units in inverse proportion to their individual governor regulation. Thus, for equal load sharing all units should have equal governor regulation, or in the case of dissimilar units, their respective governor regulation can be set to ensure proper load sharing.

Frictional forces in the governor in the connected linkage and in the control valve must be overcome before the weights can move. This means that the governor will not react to small speed changes. This small range of speed in which no governor action occurs is called the *governor dead band*.

Mechanical-hydraulic governors. For most applications the direct-acting mechanical governor does not develop enough force to operate the turbine control valve, so that a force amplifier, or servomotor, is needed. In the governor shown in Fig. T-79 movement of the governor stem causes the servomotor relay valve to move, directing operating oil to one side of the servomotor power piston and opening the other side of the power piston to drain. The power-piston movement is fed back through the servomotor linkage to the relay valve, using the speed-governor stem as a fulcrum, returning the pilot valve to neutral.

Another type of servomotor is used in the governor shown in Fig. T-80. In this governor movement of the governor stem causes changes in the speed-governor oil-

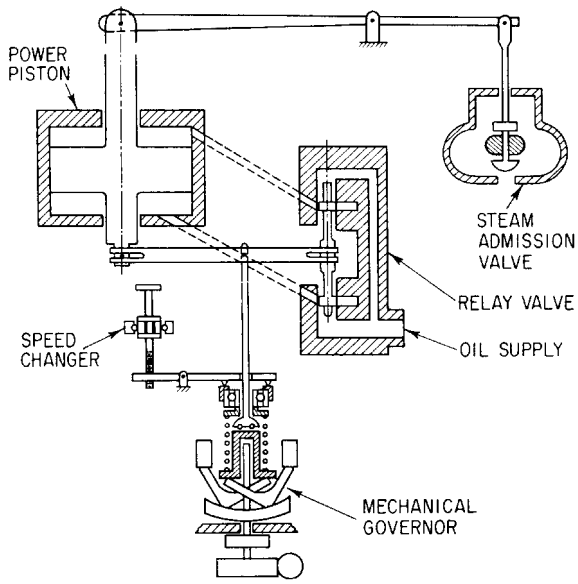


FIG. T-79 Speed governor with direct-acting servomotor. (Source: Demag Delaval.)

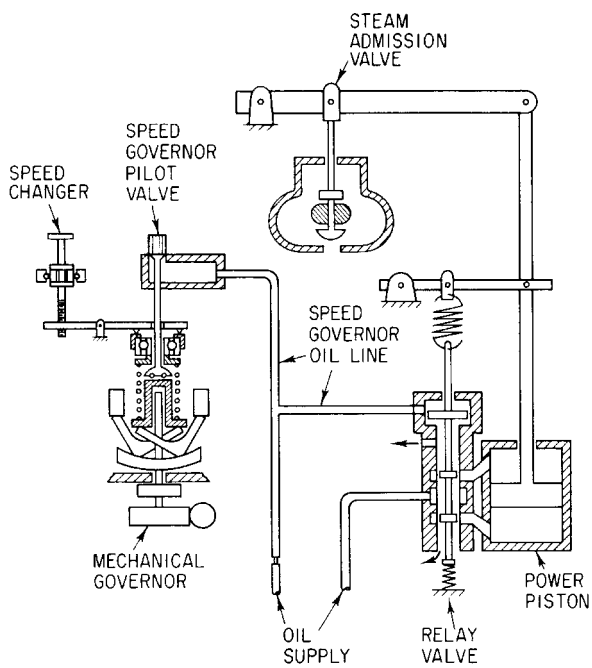


FIG. T-80 Speed governor with hydraulic servomotor. (Source: Demag Delaval.)

line pressure. This is possible because the pilot valve has a larger capacity than the orifice that supplies the speed-governor oil line. The speed-governor oil pressure acts on the relay piston against the relay-valve spring to position the relay valve and cause the power piston to move. As in the preceding servomotor the power-piston movement is fed back to the relay valve, returning it to neutral. The pilot valve on the speed-governor stem is subject to loading from the pressure in the

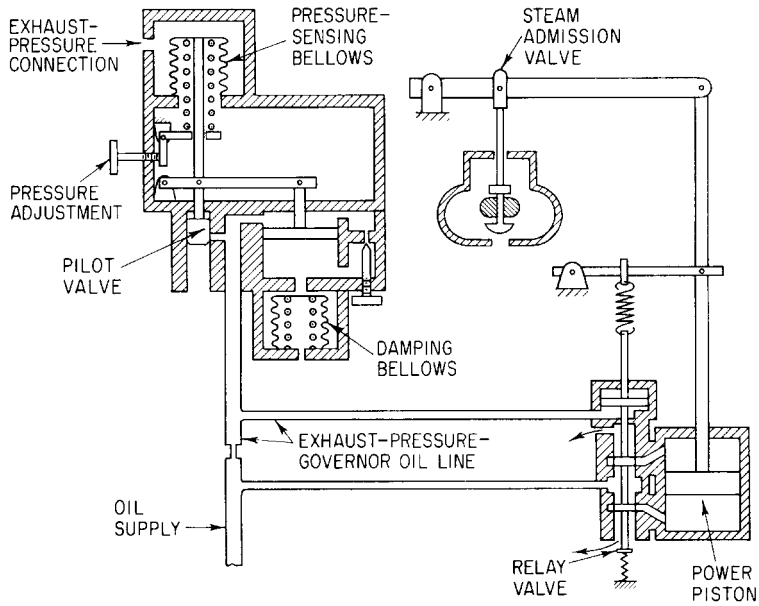


FIG. T-81 Exhaust-pressure governor. (Source: Demag Delaval.)

speed-governor oil line. This pressure loading makes the pilot valve harder to move and increases the governor regulation.

Another common governor is a pressure governor. Inlet- or exhaust-pressure governors are commonly used on turbines driving generators when the unit speed is held constant by operating in parallel with other generators. In the example of an exhaust-pressure governor shown in Fig. T-81 the exhaust pressure working on a spring-loaded bellows operates a pilot valve that causes changes in the exhaust-pressure-governor oil-line pressure. The exhaust-pressure-governor oil-line pressure, in turn, controls the inlet-nozzle-valve servomotor. A decrease in the exhaust pressure relaxes the sensing bellows, moves the pilot valve in the opening direction, and causes a drop in the exhaust-pressure-governor oil-line pressure. The decrease in the exhaust-pressure-governor oil-line pressure lowers the servomotor pilot valve. This causes the servomotor power piston to move upward, opening the nozzle valves. As the nozzle valves are opened, more steam passes through the turbine to maintain the desired exhaust pressure within the limit of the governor regulation. The action of the exhaust-pressure governor is damped by a piston with a bypassing needle valve and by a spring-loaded bellows.

Electrohydraulic governors. These governors employ an electric speed-sensing element whose signal is transmitted to an electrohydraulic actuator that drives the same primary relay valve as that shown in the other governors. Such electric governors have the advantage of a wider speed range and more precise control than have mechanical-hydraulic governors. Electrohydraulic governors may use hydraulic oil systems developed for mechanical governors, in which case the oil pressure would be in the neighborhood of 100 psig. Such a system is shown in Fig. T-82. It is used for a boiler-feed-pump turbine. It receives a feedwater demand signal (e.g., 4 to 20 mA DC) and uses it to set unit operating speed during normal operation (40 to 100 percent speed) or to set manually the valve position from 0 to 100 percent speed or the minimum governor set point during startup. The system consists of:

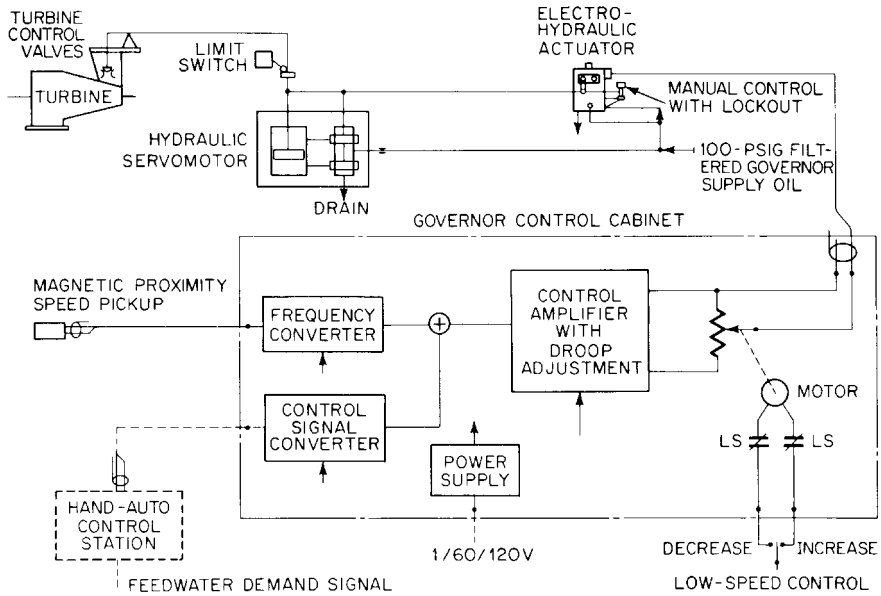


FIG. T-82 Electrohydraulic governor. (Source: Demag Delaval.)

1. A governor control cabinet that contains the electronic circuits necessary for establishing startup valve position and normal speed control
2. A hand-automatic control station to facilitate manual control of the unit
3. A magnetic proximity speed pickup and a gear mounted on the turbine shaft that produce a frequency signal proportional to the unit speed
4. An electrohydraulic actuator mounted on the hydraulic servomotor that positions the servomotor relay valve in response to the control signals from the electronic circuits of the governor
5. A hydraulic servomotor for positioning the valve gear operating the steam control valves in response to the electrohydraulic actuator

A electrohydraulic governor using control oil pressure in the range of 1000 to 1500 psig is shown in Fig. T-83. It is used in the control system of a process-industry turbine-driven unit. This system may be used with modular control valves. Steam is admitted to the turbine by any of several valves; the opening of each one is determined by the governor servoamplifier. The system consists of:

1. A governor control cabinet that contains the electronic circuits to establish a speed set point by a signal from the controller that measures process variables
2. A manual control station that may be used to establish a speed set point
3. Magnetic speed pickups and a gear mounted on the turbine shaft that produce a frequency signal proportional to the unit speed
4. An electrohydraulic relay valve mounted on the hydraulic servoactuator
5. A feedback transducer mounted on the hydraulic servomotor that returns an electric signal of valve position to servoamplifier circuits in the governor control cabinet

The complete servoactuator–steam-control-valve module is adjusted at assembly so that the control valve will be closed in the absence of a control signal. The electric

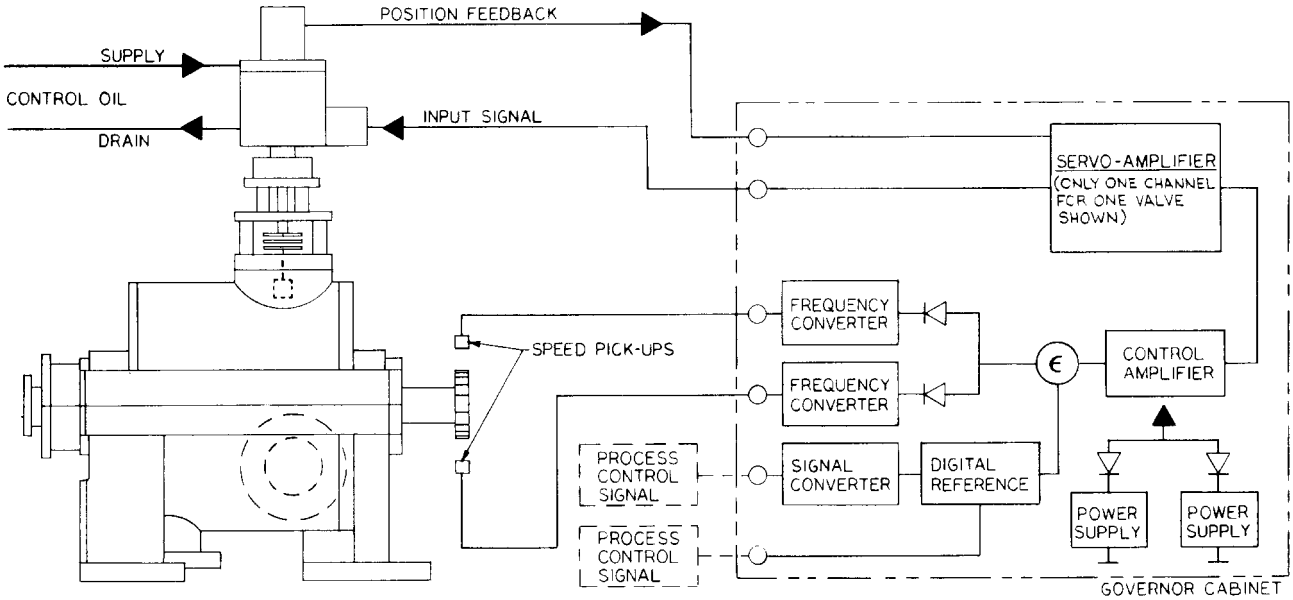


FIG. T-83 Electrohydraulic control system process reset speed governor. (Source: Demag Delaval.)

control system is interlocked with the emergency shutdown system so that the steam control valves act as a backup to the trip-and-throttle valve during emergency shutdown, e.g., absence of steam, electricity, or oil.

Extraction governors. Another form of pressure governor is the extraction governor. This governor could operate, through a servomotor, a set of nozzle valves (secondary valves), which as they are opened pass more steam through the later stages of the turbine and less into the extraction line. Normally when an extraction governor is fitted, it must be coordinated with a speed governor to ensure complete control of the turbine. This can be done by using a master regulator to connect the speed-governing and extraction-governing systems, as shown in Fig. T-84. Pistons in the master regulator receive control-pressure signals from the speed governor and from the extraction-pressure governor. The control-piston movements are transmitted through the regulator linkage to the pilot valves, which control the pressure in the servomotor-control oil lines. These controlled pressures cause the servomotors to make the necessary corrections in the nozzle-valve settings.

Overspeed trips. Mechanical overspeed trip devices have been used for safety against disasters caused by runaway turbines. The unbalanced and spring-loaded plunger is probably the most common (see Fig. T-85). Mounted in a hole through the rotor and across the axis, it is held in position by a spring until the turbine speed is sufficient for the plunger unbalance to generate centrifugal force greater than the spring force. It then pops out a short distance and strikes a lever system that mechanically or hydraulically actuates the turbine trip valve.

A variation of the plunger type is shown in Fig. T-86; actually two plungers are used, each attached to a surrounding ring. The rings are attached to each other so that they can move only in opposite directions. A spring-loaded plunger that would be tripped by a shock (e.g., an earthquake) in its tripping direction will be opposed by the other spring-loaded plunger.

Figure T-87 illustrates a hydraulic trip valve used with a mechanical shaft-mounted trip. When trip speed is reached, the plunger (or ring) mounted in the trip

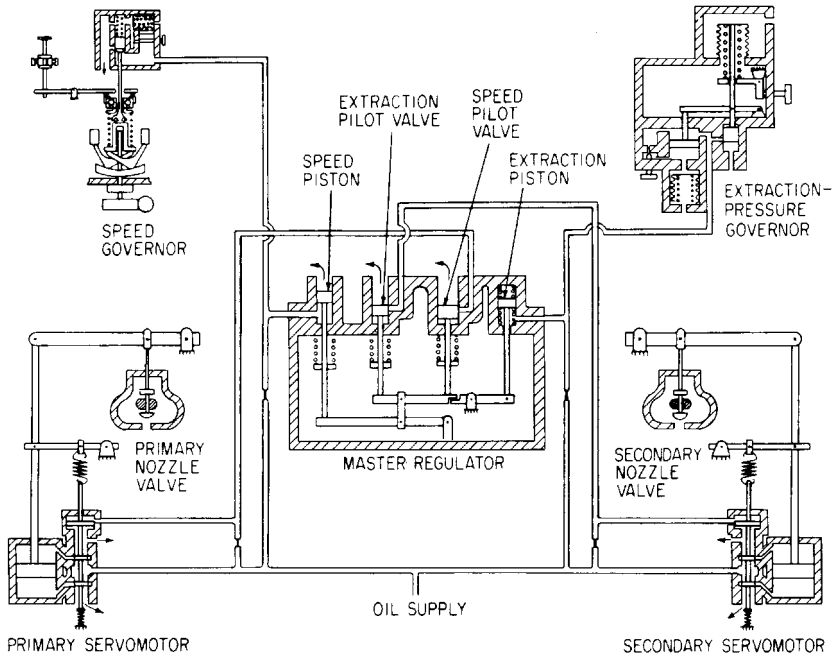


FIG. T-84 Extraction governor with master regulator. (Source: Demag Delaval.)

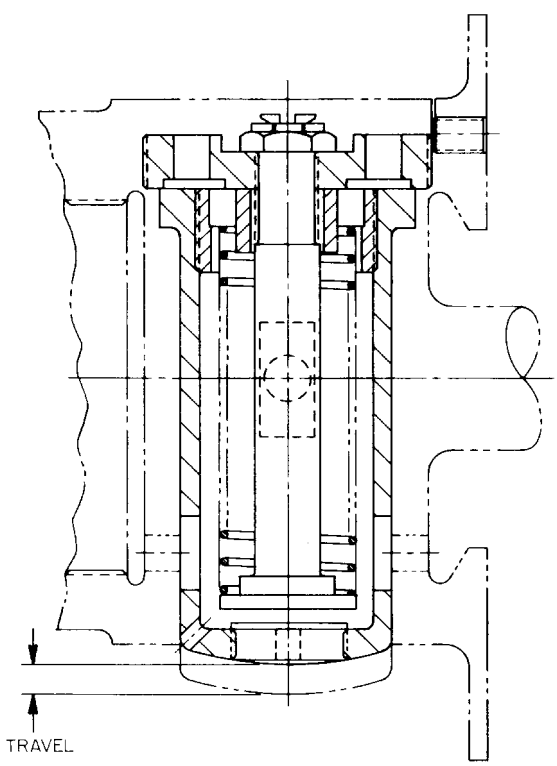


FIG. T-85 Overspeed trip plunger. (Source: Demag Delaval.)

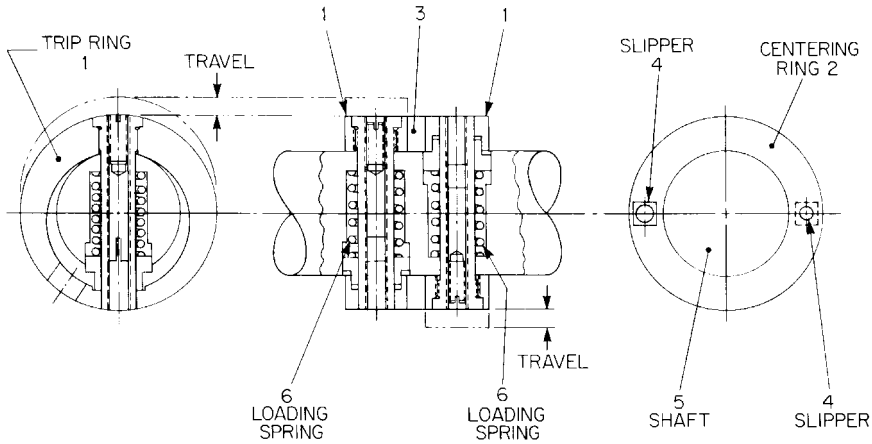


FIG. T-86 Two-ring (shockproof) overspeed trip. (Source: Demag Delaval.)

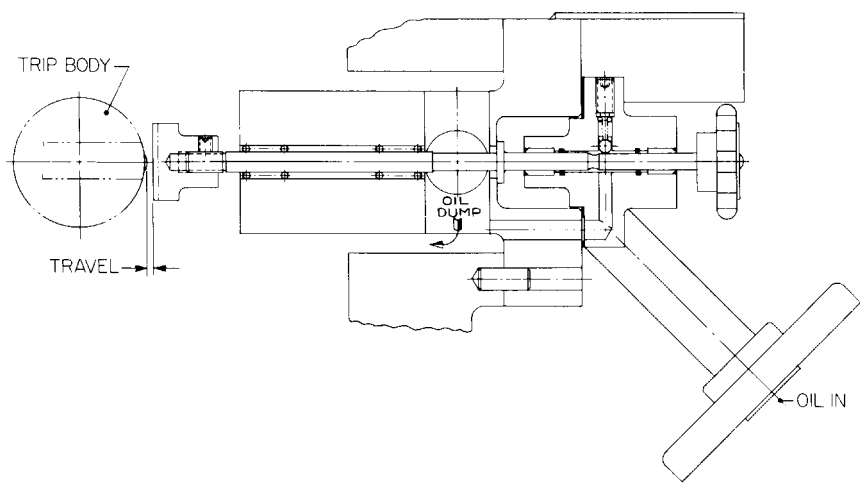


FIG. T-87 Hydraulic trip valve. (Source: Demag Delaval.)

body moves out and strikes the knob mounted on the valve stem. Axial motion to the right unseats the valve to dump oil from the trip circuit. Contemporary designs of overspeed trips favor electrical non-shaft-contacting trips. These use electrical circuitry that actuates the hydraulic trip valve connected to the steam trip valve (see Fig. T-83). See Figs. T-88 and T-89.

Reference and Additional Reading

1. Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Steam Separators for Steam Drum Applications*

Solids in boiler water

This information source guarantees less than 1 ppm of total dissolved solids in the outlet steam with 2000 ppm boiler water concentration.

* Source: Peerless, USA.

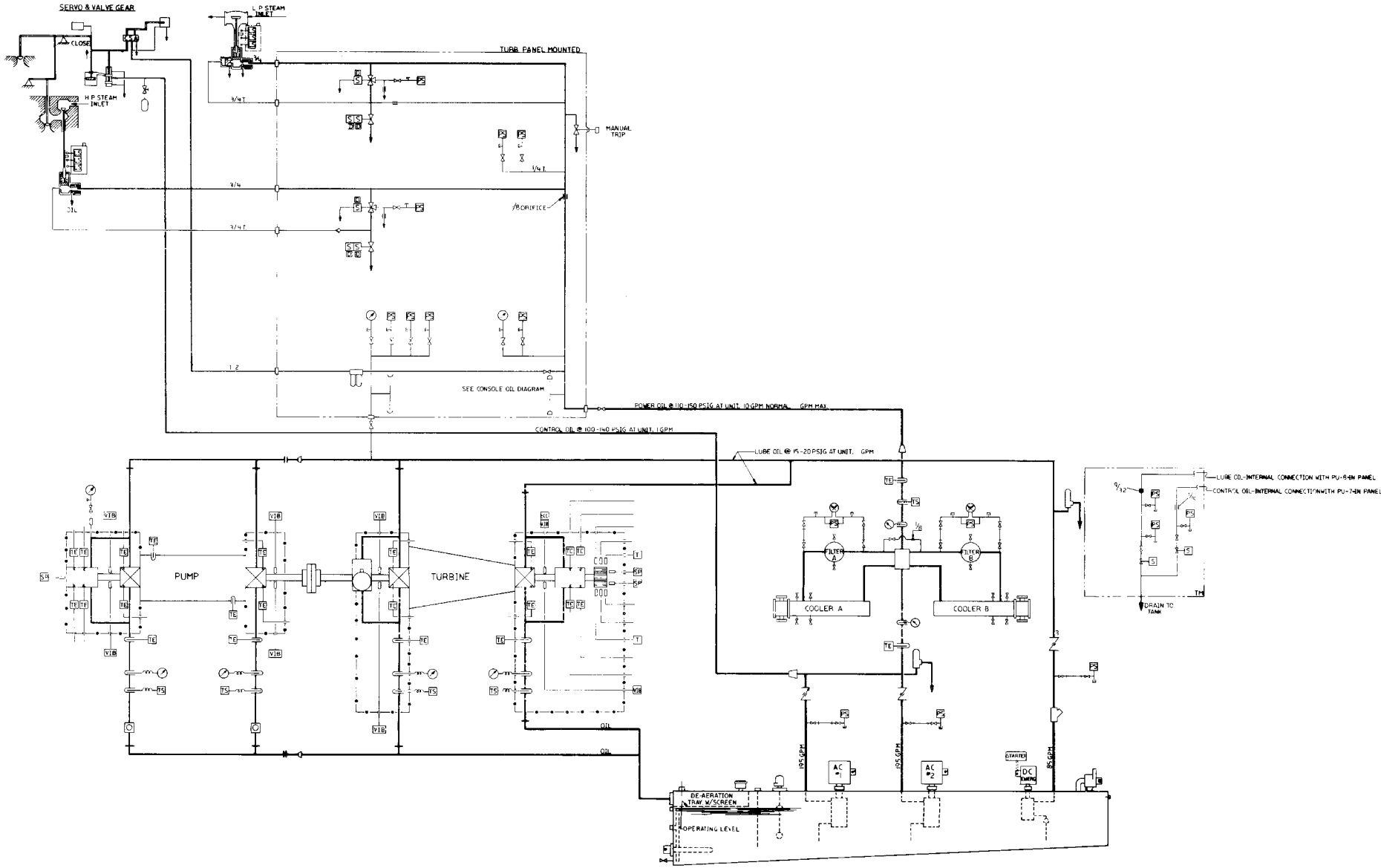


FIG. T-88 Lubrication and control oil system: turbine-driven boiler-feed water pump. (Source: Demag Delaval.)

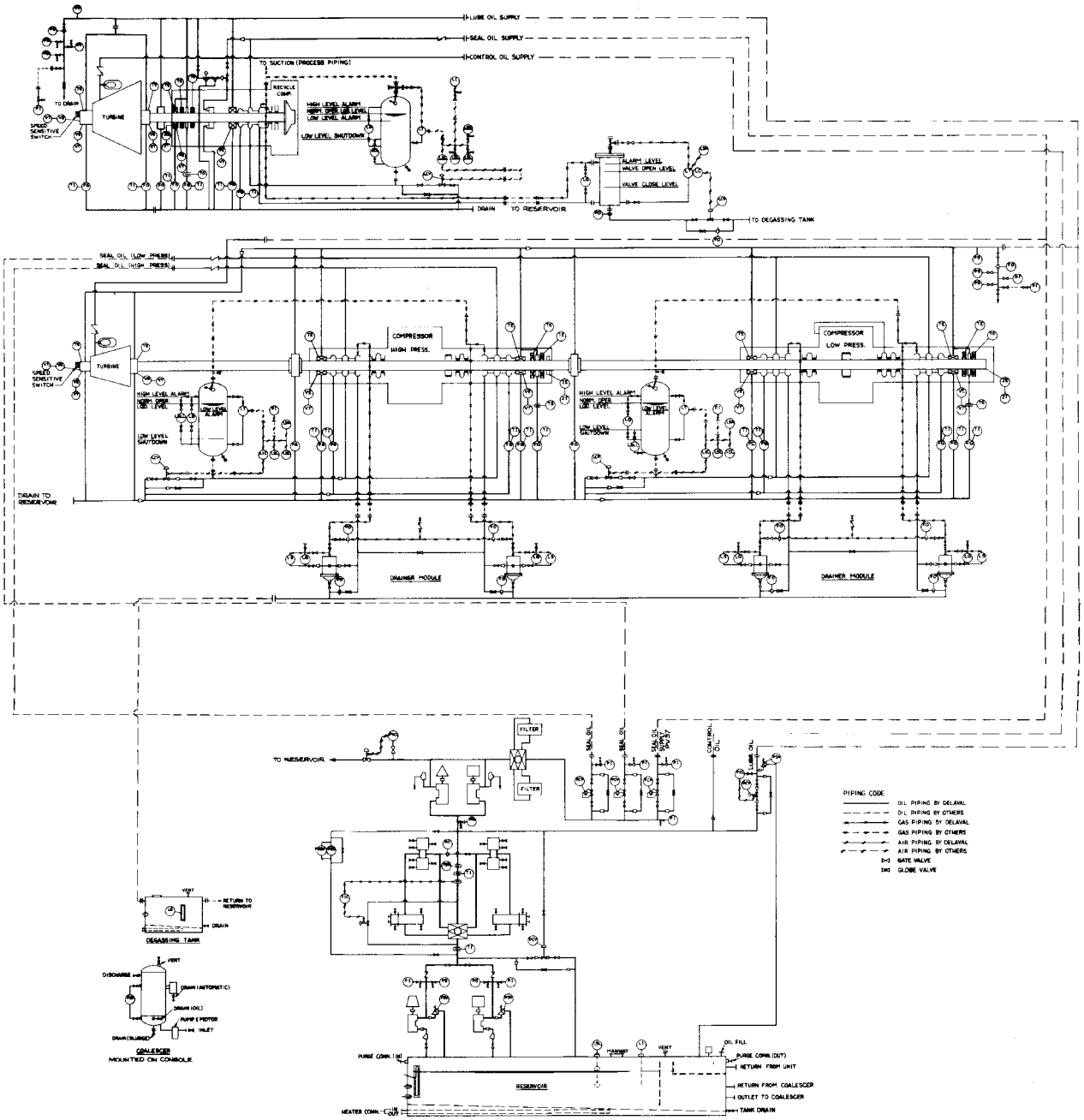


FIG. T-89 Lubrication and control oil system: petrochemical plant. (Source: Demag Delaval.)

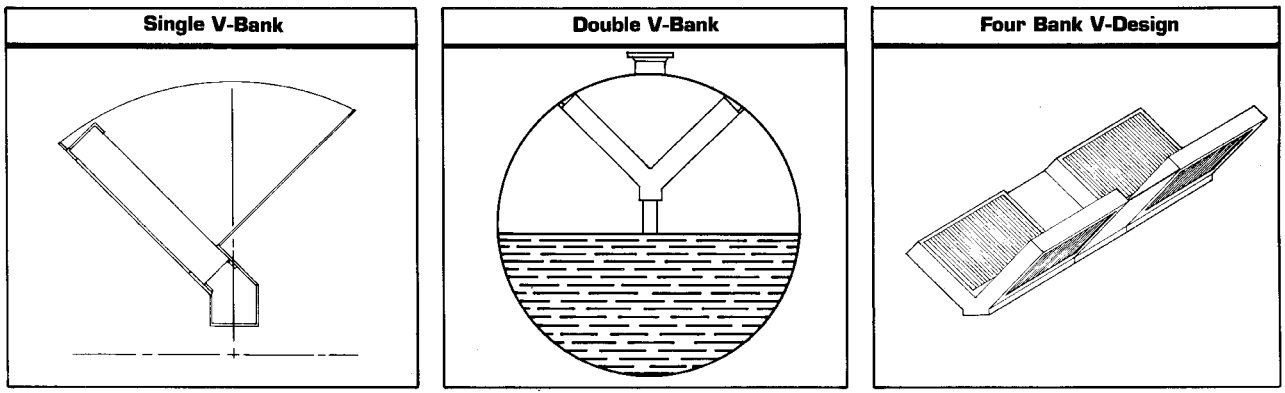


FIG. T-90 Typical steam separator designs. (Source: Peerless.)

Steam dryness

Manufacturer guarantees an outlet steam dryness of 99.9+%. This incredible separation performance is achieved with the compact, low-profile "P5" (this information source's model designation) separation element. See Fig. T-90.

The P5 profile

The low-profile P5 increases internal steam drum space while effectively checking troublesome deposits from silica-laden steam. Total system cost is reduced because the P5's massive liquid handling ability often eliminates the need for primary separation.

The unique separation performance characteristics of the P5 allow it to achieve a much lower turndown ratio than cyclonic alternatives while providing stabilized separation performance in varying boiler environments. Against mesh pad the P5 eliminates the carryover and deterioration commonly experienced with pads thereby reducing expensive line clogs and turbine rotor damage.

Some custom designs incorporate removable vane element features that facilitate maintenance after the system is installed.

Steam washing designs are also available when control of silica-laden steam vapor is needed to prevent damaging turbine deposits.

Field installation is a new dimension to the turnkey system approach.

Note: Retrofit systems designed for installation in prefabricated vessels are possible if manway access is provided. The manufacturer can preassemble and tack-weld the unit, match-marking the adjacent parts for easy disassembly in the field. Final assembly is completed inside the vessel by realigning the match-marked assembly map and welding the unit in place.

Principles of operation

The vane unit (see Figs. T-91 and T-92) is the heart of the separator. As the gas enters the vane unit, it is divided into many vertical ribbons (A). Each ribbon of gas is subjected to multiple changes of direction (B) as it follows its path through the vanes. This causes a semiturbulence and a rolling of the gas against the vanes (C). The entrained droplets are forced to contact the vane walls where they impinge and adhere to the vane surface (D). This liquid then moves into the vane pockets (E) and out of the gas stream where it is drained by gravity into the liquid reservoir. The collected liquid can then be disposed of as desired.

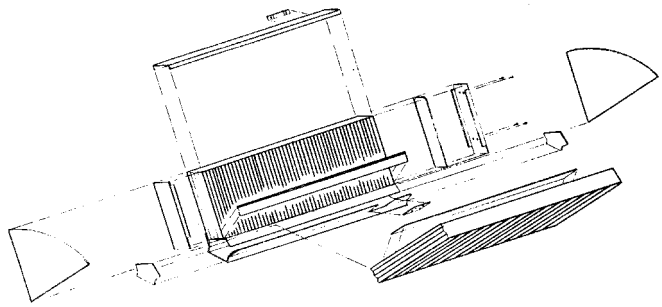


FIG. T-91 Exploded view of double V-bank retrofit design. (Source: Peerless.)

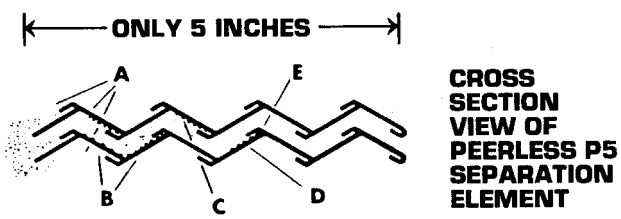


FIG. T-92 Separation element details. (Source: Peerless.)

It is significant to note that the liquid drainage in the vane-type mist extractor differs from the drainage in other impingement-type mist extractors, in that vane drainage occurs with the liquid out of the gas flow and at a right angle to the gas flow.

The individual vane corrugations, depth and size of the liquid pockets, and the vane spacing are critical features of the vane-type mist extractor. The slightest variation in any of these three features will materially decrease the capacity and performance of this type of separator.

See Table T-8.

Turbochargers*

Turbochargers (see Fig. T-93) are used to increase the operating pressure level of internal combustion engines, thereby increasing the power output of the engine. The turbocharger serves to uprate the engine or to restore sea-level performance at high altitudes. At the same time, a saving in specific fuel consumption is achieved.

Main industrial applications are on two- and four-cycle diesel engines, gas engines, and dual-fuel engines.

Performance parameters vary, and close cooperation between turbocharger and engine manufacturers is required in order to adjust the turbocharger to an individual application.

Basically, the turbocharger is a gas turbine consisting of a compressor and a turbine with the engine replacing the combustion chamber as shown in Fig. T-93. The air consumed by the engine is drawn from the atmosphere, compressed by

* Source: Demag Delaval, USA.

TABLE T-8 Application Matrix*

OEM Model No.	Steam Flow, lb·h	Operating Pressure, psig	Minimum† Vessel Diameter
SD-2-1.5	20,000	150	36
SD-2-3	20,000	300	36
SD-2-6	20,000	600	36
SD-2-10	20,000	1000	36
SD-5-1.5	50,000	150	42
SD-5-3	50,000	300	42
SD-5-6	50,000	600	42
SD-5-10	50,000	1000	36
SD-10-1.5	100,000	150	54
SD-10-3	100,000	300	48
SD-10-6	100,000	600	48
SD-10-10	100,000	1000	42
SD-15-1.5	150,000	150	60
SD-15-3	150,000	300	54
SD-15-6	150,000	600	54
SD-15-10	150,000	1000	48
SD-20-1.5	200,000	150	60
SD-20-3	200,000	300	60
SD-20-6	200,000	600	54
SD-20-10	200,000	1000	54

* Typical application guide; consult OEM representative for other variations.

† High-level shutdown is 4 in below vessel except for model SD-20-1.5, which requires 8 in.

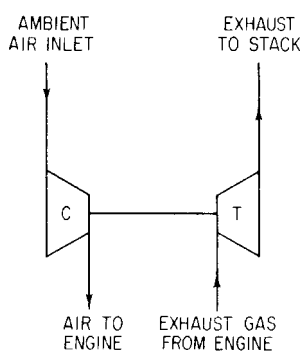


FIG. T-93 Turbocharger. (Source: Demag Delaval.)

compressor *C*, and discharged through a cooler in some designs into the intake manifold of the engine. The exhaust gas from the engine is expanded in turbine *T* and is exhausted to the atmosphere.

Typically, there is no mechanical connection between the shaft of the turbocharger and the engine. The power produced by the turbine matches the power absorbed by the compressor. This balance adjusts itself by speed variation.

Typical pressure ratios used were 1.5 to 3.0; lately, modern turbochargers have used pressure ratios of 3.2 to 3.5 and higher. The turbine pressure ratio is somewhat smaller than the compressor pressure ratio because of the pressure drop in the engine.

The compressor consists of a single centrifugal stage and the turbine of a single radial or axial stage that may be arranged between or outboard of the turbocharger

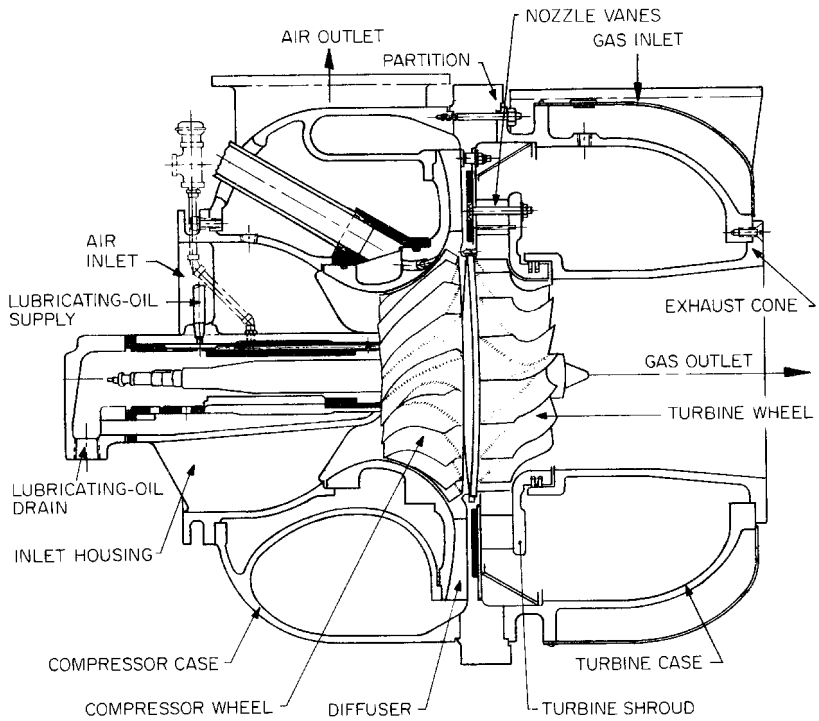


FIG. T-94 Turbocharger. (Source: Demag Delaval.)

bearings. If the pressure ratio exceeds the capability of the single stages, two turbochargers of different standardized sizes may be used in series.

A cross-section of a Delaval turbocharger is shown in Fig. T-94. A mixed-flow centrifugal-compressor stage and a mixed-flow radial-turbine stage are arranged back to back on one side of the bearing case.

Turboexpanders*

Expansion of gas in a turbine produces work and lowers the temperature of the gas stream as energy is removed. Turbines that produce work from the expansion of process gases and that serve the recovery of process waste energy are often called *expanders*. Some of these expanders are of considerable horsepower size. Representative gas conditions are inlet temperature = 1000°F, inlet pressure = 300 psia, and exhaust pressure = atmospheric or above.

Turboexpanders are part of low-temperature process equipment and refrigerators and are widely used in the cryogenic industry. Typical applications are air-separation plants for the production of gaseous and liquid oxygen and nitrogen when the turboexpander operates on an air or nitrogen stream down to the vicinity of –300°F. Applications involving the lowest temperature are helium liquefiers in which the turboexpander may operate at a temperature as low as –450°F.

The single-stage radial turbine has almost become a standard, although some axial turboexpanders have been built. The turbine is arranged outboard of the bearing case and separated from it by a seal. Most designs have oil-lubricated sleeve

* Source: Demag Delaval, USA.

bearings. The use of bearings lubricated by the cycle gas is very attractive, and several experimental units with gas-lubricated bearings have been built. The load horsepower is of secondary importance and is usually absorbed by a single-stage centrifugal compressor arranged outboard of the bearing case at the opposite end. The compressor may compress atmospheric air and dissipate the load by throttling, or some use may be made of the horsepower by compressing seal or process gas. Some larger units have been built with a load-absorbing generator driven through a reduction gear. Small units may dissipate the load by an oil brake.

Flow and horsepower sizes vary over a wide range. Turboexpanders with 4- to 6-in-diameter turbine wheels are typical. Turboexpanders have been built with wheel diameters over 17 in and as small as $^{5}/_{16}$ in. Miniature turboexpanders with wheel diameters below 1 in have a rotative speed above 100,000 rpm when only gas-lubricated bearings make a successful design possible.

References and Additional Reading

1. Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.
2. Bloch, H., and Soares, C. M., *Turboexpanders and Process Application*, Gulf/Butterworth-Heinemann, 2001.

U

Ultrasonic Cleaning*

The material presented in this section features models of ultrasonic cleaners that are unique to this manufacturer. Other sources' equivalent models' features should be considered for relevant applications.

Theory of Ultrasonic Cleaning

Everything that makes a sound vibrates, and everything that vibrates makes a sound; however, not all sounds are audible. Ultrasound literally means beyond sound—sound beyond the audible spectrum. Considering 18,000 Hz (cycles per second) as an approximate limit of human hearing, ultrasonics refers to sound above 18,000 Hz.

The ultrasonic power supply (generator) converts 50/60 Hz voltage to high frequency 20 or 40 kHz (20,000/40,000 cycles per second) electrical energy. This electrical energy is transmitted to the piezoelectric transducer within the converter, where it is changed to high-frequency mechanical vibration. The vibrations from the converter are amplified by the probe (horn), creating pressure waves in the liquid. This action forms millions of microscopic bubbles (cavities) that expand during the negative pressure excursion and implode violently during the positive excursion. It is this phenomenon, referred to as cavitation, that produces the powerful shearing action at the probe tip, and causes the molecules in the liquid to become intensely agitated.

Detailed description

Power supply. Depending on the model, the power supply transforms conventional 50/60 Hz electrical power into high-frequency electrical power at 20,000 or 40,000 Hz. Power supplies are typically rated in watts of output power. It should be noted that using a power supply with a higher wattage rating does not mean that more power will automatically be transmitted to the liquid. Rather, it is the resistance to the movement of the probe (load), that determines how much power will be delivered into the liquid. Load is determined by three factors: sample volume, sample viscosity, probe size and, in some cases, a pressurized environment. Under identical loading conditions, the wattage delivered by two power supplies with different power ratings will be the same (provided both have sufficient power capability).

The speed control on an automobile can, to a certain extent, be compared to an ultrasonic processor. The cruise control is designed to maintain a constant vehicle rate of travel. As the terrain changes, so do the power requirements. The speed control senses these requirements, and automatically adjusts the amount of power delivered by the engine in order to compensate for these ever-changing conditions. The steeper the incline, the greater the resistance to the movement of the vehicle

* Source: Sonics, USA. Adapted with permission.

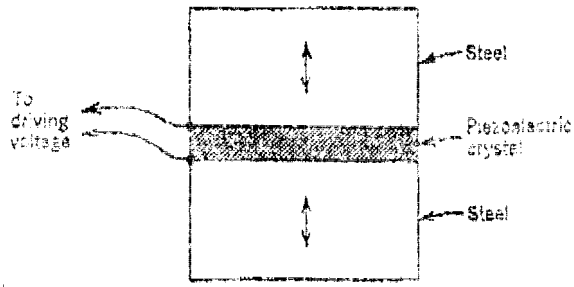


FIG. U-1 Converter. (Source: Sonics.)

and the greater the amount of power that will be delivered by the engine to overcome that resistance.

The ultrasonic processor is designed to deliver constant amplitude (peak-to-peak displacement at the probe tip). As the resistance to the movement of the probe increases, so do the power requirements. The power supply senses these requirements, and automatically increases the amount of power delivered in order to maintain the selected excursion at the probe tip constant.

The amplitude control allows the ultrasonic vibrations at the probe tip to be set to any desired level. Although the degree of cavitation required to process the sample can readily be determined by visual observation, the amount of power required cannot be predetermined. A sensing network continuously monitors the output requirements, and automatically adjusts the power to maintain the amplitude at the preselected level. Negligible power is required to keep an ultrasonic probe resonating when operated in air.

The greater the resistance to the movement of the probe due to higher viscosity, deeper immersion of the probe into the sample, larger probe diameter, or higher pressure, the greater the amount of power that will be delivered to the probe. Setting the amplitude control fully clockwise will not cause the maximum power to be delivered to the sample. The maximum power any ultrasonic processor is capable of delivering is only delivered when resistance to the movement of the probe is high enough to draw maximum wattage.

This phenomenon can be demonstrated as follows. Depress the probe down against a piece of wood. As the down pressure is increased and there is consequent increased resistance to the movement of the probe. Thus, there is a greater amount of power delivered by the power supply.

Converter. The converter receives high-frequency electrical energy from the power supply and converts it into mechanical vibration. Converters contain lead zirconate titanate piezoelectric ceramic discs. When an alternating voltage is applied to the opposing faces of the discs they expand and contract with the repeated change of polarity. Thus, when an alternating voltage at a frequency of 20 or 40 kHz is applied to the discs, they vibrate at that frequency. The transducer consists of PZT discs sandwiched between metal sections. The entire assembly is designed to resonate at a predetermined frequency, and its length is typically equal to one-half wavelength of the applied frequency. See Fig. U-1.

Probes. The probe (also a one-half-wavelength-long section) radiates and focuses the ultrasonic energy into the liquid. Probes with smaller tip diameters produce cavitation of greater intensity, but the energy released is restricted to a narrower,

more concentrated field immediately below the tip. Conversely, probes with larger tip diameters produce reduced intensity, but the energy is released over a greater area. In most cases the larger the tip diameter, the larger the volume that can be processed, but at reduced intensity. High gain probes produce much higher intensity than standard probes and are usually recommended for processing larger volumes or difficult applications. Probes are fabricated from high-grade titanium alloy (TI-6AL-4V) because of its good acoustical properties, low toxicity, high resistance to corrosion and excellent resistance to cavitation erosion. They are autoclavable and are available with threaded ends to accept replaceable tips, microtips, and extenders.

When driven at its resonant frequency, the end of the probe expands and contracts longitudinally about its center. However, no longitudinal motion occurs at the nodal plane (plane of no activity), allowing accessories to be connected to the probe at that point. Each probe has a gain ratio. The ratio of probe output amplitude to input amplitude is a function of the cross-sectional ratio between the input (top half) and output (bottom half) sections of the probe. The larger the upper section, the greater the amplitude of the tip of the probe.

Booster. For difficult applications, boosters are sometimes used to increase the amplitude by a factor of 2. The booster (also called an "amplitude transformer") is a one-half-wavelength long resonant section, mounted between the converter and the probe. See Fig. U-2.

Questions and answers

What are the differences between an ultrasonic processor and an ultrasonic bath? The intensity within a bath is fixed, low, location dependent, and inconsistent, due to the fluctuation in the level and temperature of the liquid.

With an ultrasonic processor, processing is fast and highly reproducible. The energy at the probe tip is high (at least 50 times that produced in a bath), focused, and adjustable.

With ultrasonic processing, are there any limitations? Yes, viscosity, temperature, and liquid characteristics.

The more viscous the material, the more difficult it is for the vibrations to be transmitted. Typically, the maximum viscosity at which a material can be effectively processed is 5000 cps.

With standard systems the practical upper limit on temperature is approximately 150°C? Solid probes can safely be used with both aqueous solutions and low surface tension liquids (e.g., solvents); however, probes with replaceable tips should *never* be used with low surface tension liquids.

Which instrument should I use? The 400, 500, and 600 watt units are the most versatile because they can process both large and small volumes—on a batch basis, as little as 200 μL with a microtip, or as much as 1 L with a 1-in (25-mm) probe; on a flow-through basis, up to 10 L/h.

However, since every instrument will perform equally well up to a certain volume, for samples up to 70 mL the 70-watt unit is recommended, and the 130-watt unit for samples up to 130 mL.

Which probe is best suited for my application? The larger the probe diameter, the larger the volume that can be processed, but at lesser intensity. See probe listings

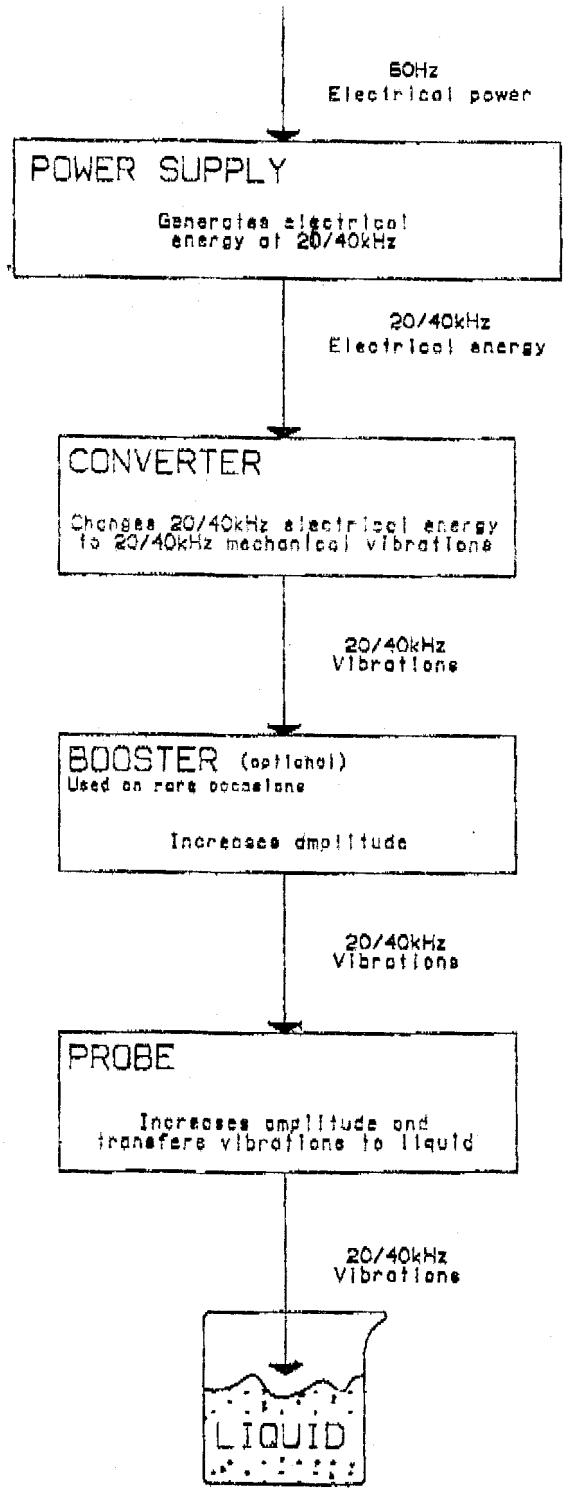


FIG. U-2 Booster or "amplitude transformer." (Source: Sonics.)

for recommendations; and for dependable performance, *always* use a solid probe when working with low surface tension liquids.

Do all manufacturers rate their instruments the same way? Unfortunately not: Sonics uses the RMS rating—the amount of power, measured in watts, that a unit is capable of delivering continuously. Others use peak power rating—the maximum amount of power, measured in watts, that a unit is capable of delivering only for a short time.

Ultrasonic cleaners are highly specialized pieces of equipment and generally correspond to a U.S. Environmental Protection Agency (EPA)—specified method. It is useful to look at specific cleaner models at this point.

To illustrate this, the components list for EPA environmental testing packages for extracting pesticides/PCBs and nonvolatile and semivolatile organic compounds from soil, sludge, and waste samples in accordance with EPA method SW 846-3550 is included here.

Special EPA Environmental Testing Packages

For extracting pesticides/PCB, and nonvolatile and semivolatile organic compounds from soil, sludge, and waste samples in accordance with EPA Method SW 846-3550.

	SINGLE TEST SYSTEM
1 each	Model VC 601—600-watt ultrasonic processor with converter and $^1/_2$ in (13 mm) probe with threaded end and replaceable tip.
1 each	Tapered microtip $^1/_8$ in (3 mm). Order number 630-0418.
1 each	$^3/_4$ in (19 mm) solid probe. Order number 630-0208.
	DUAL TEST SYSTEM—processes 2 samples simultaneously*
1 each	Model VC601—600-watt ultrasonic processor with converter and $^1/_2$ in (13 mm) probe with threaded end and replaceable tip.
1 each	Tapered microtip $^1/_8$ in (3 mm). Order number 630-0418.
1 each	Dual probe. Order number 630-0525.
	Note: The dual probe consists of an aluminum coupler, Order number 630-0526, and 2 special $^3/_4$ in (19 mm) solid probes, Order number 630-0527.

* Cannot be used with standard sound-abating enclosure.

All probes, including those with replaceable tips, are tuned to resonate at 20 kHz ± 100 Hz. If the replaceable tip is removed or isolated from the rest of the probe, that element will no longer resonate at 20 kHz and the power supply will fail. Organic solvents (e.g., methylene chloride) and low surface tension liquids will penetrate the interface between the probe and the replaceable tip and carry the particulates into the threaded section, isolating the tip from the probe. When working with organic solvents or low surface tension liquids, *always* use a solid probe or as an alternate a full wave 10-in (254-mm) probe or an extender. *Never* use a probe with a replaceable tip.

- Ultrasonic extraction is faster than soxhlet extraction, and requires 50 percent less solvent. Typical applications can be processed in less than 10 min compared to hours.
- The dual probe is a valuable time-saving tool capable of substantially reducing labor costs when processing a large number of samples.

Next, we look at the features and specifications of typical ultrasonic processors for high-volume applications.

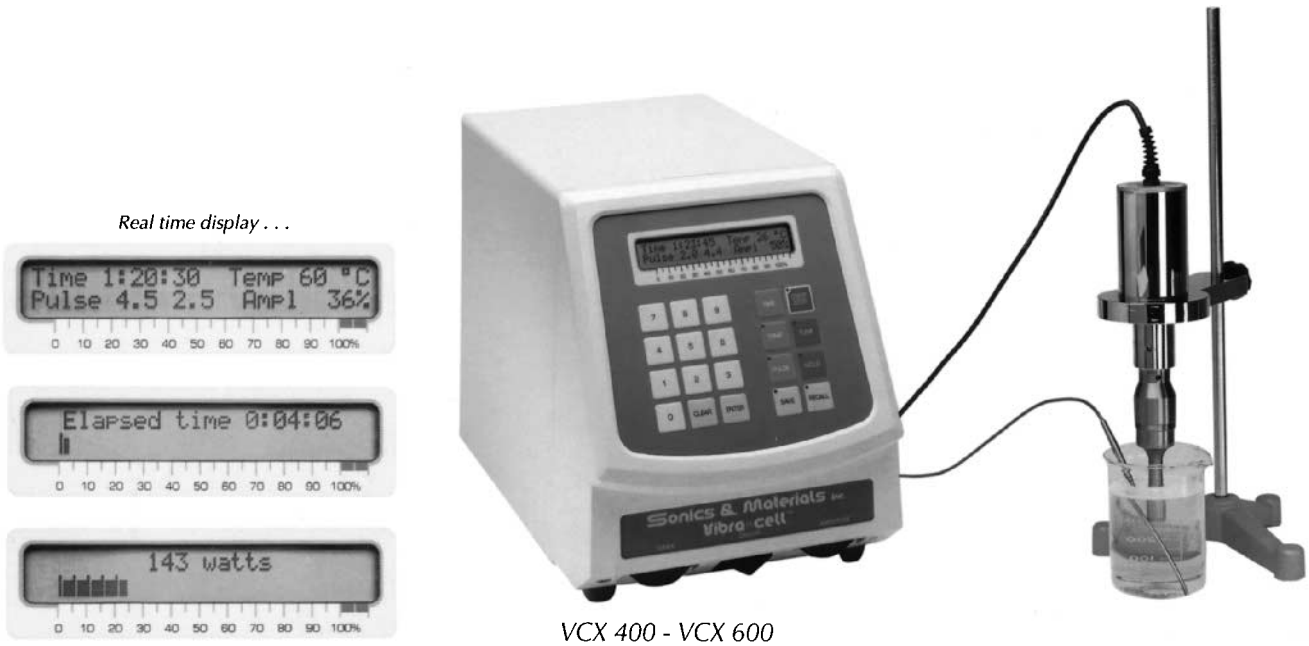


FIG. U-3 Ultrasonic cleaners (400–600 in). (Source: Sonics.)

400- and 600-Watt Ultrasonic Processors—250 microliters to liters (see Fig. U-3)

- Integrated temperature controller: Precludes harmful overheating of the sample and guarantees process integrity by terminating the ultrasonics when the sample temperature reaches a predetermined limit. Allows process control and monitoring from 1 to 100°C.
- Sealed converter: Inhibits failure due to humidity, dust, dirt, or corrosive fumes.
- Microprocessor based—programmable: Digital accuracy and repeatability ensures adherence to the most exacting protocol.
- Real-time display: Provides a window on the process. No more assumptions. No more approximations. All parameters are continuously displayed on the screen, providing operating mode confirmation without process interruption.
- Consistent reproducibility: Time-saving memory facilitates complex protocol duplication, automates repetitive tasks, and eliminates technician-to-technician method variability. Conveniently stores up to ten procedures.
- Automatic frequency control: Eliminates the need for constant adjustment of the power supply after initial setup.
- Variable amplitude control: Allows the ultrasonic vibrations at the probe tip to be set to any desired amplitude. Selected output level is clearly displayed on the screen.
- Automatic amplitude compensation: Ensures uniform probe amplitude regardless of the varying loading conditions encountered during the processing cycle.
- Wattmeter: Digitally displays the actual amount of power being delivered to the probe.
- Ten-hour process timer: Controls the processing time—from one second to ten hours.

- Independent on/off pulser: Enables safe treatment of temperature-sensitive samples at high intensity. Both on and off cycles are independently controllable from 0.1 to 10 s.
- Elapsed time indicator: Monitors both the elapsed time and the duration of processing.
- Operation: Tactile keypad, menu-driven prompts, and easy-to-use function keys combine to make for a user-friendly system.
- Space: Compact upright configuration minimizes bench space.
- Remote actuation compatible: Can be remotely actuated with a footswitch or a computer-controlled relay.
- Universal filtering: Proprietary filtering guarantees compliance with worldwide rules and regulations governing radio-frequency interference and leakage current.

Specifications

Power supply	Net power output: VCX 400—400 watts. VCX 600—600 watts. Frequency: 20 kHz. Dimensions (H × W × D): $9^1/_4 \times 7^1/_2 \times 13^1/_2$ in (235 × 190 × 340 mm). Weight: 15 lb (6.8 kg).
Sealed converter	Model CV 26. Type: Piezoelectric—PZT—lead zirconate titanate crystals. Diameter: $2^1/_2$ in (63.5 mm). Length: $7^1/_4$ in (183 mm). Weight: 2 lb (900 g). Cable length: 5 in (1.5 m).
Standard probe	Tip diameter: $^1/_2$ in (13 mm) solid or with threaded end with replaceable tip. Processing capability: 10 to 250 mL. Length: $5^3/_8$ in (136 mm). Weight: $^3/_4$ lb (340 g). Titanium alloy: TI-6AL-4V.
Temperature probe (optional)	Stainless steel—Order No. 830-00060
Electrical requirements	100, 117, 220, or 240 volts, 50/60 Hz. For export, please specify desired voltage option.

A typical specification for an ultrasonic cleaner used in industrial applications is as follows.

1500-Watt Ultrasonic Processor

- Automatic tuning
- Up to 15 L on a batch basis
- Up to 100 L/h on a flow-through basis

When used in conjunction with the optional high volume continuous flow cell, throughput rate is typically 100 L/h—variables are viscosity and desired degree of processing. The probe within the continuous flow cells is $^3/_4$ in (19 mm) in diameter with threaded end and replaceable tip.* Both the probe and the processing chamber

* Can be supplied with solid probe. Please specify. Probes with replaceable tip should *never* be used with solvents or low surface tension liquids.



FIG. U-4 Ultrasonic 1500-m cleaner. (Source: Sonics.)

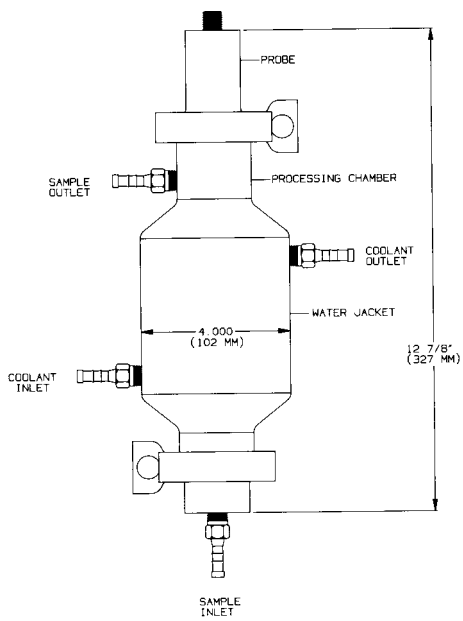


FIG. U-5 High volume continuous flow cell—all wetted parts are autoclavable. (Source: Sonics.)

are manufactured from titanium alloy TI-6AL-4V. The cell is manufactured of 316 stainless steel, and is capable of operating at pressures up to 50 psi (345 kPa/3.45 bar). The continuous flow cell is recommended for the treatment of low viscosity samples, where the required insonation time is relatively short. Designed primarily for dispersing and homogenizing. For optimum performance, a mechanical mixer should also be used to premix the sample when working on a flow-through basis. Easily disassembled for inspection and cleaning. See Figs. U-4 and U-5.

Specifications

Power supply	Net power output: 1500 watts. Frequency: 20 kHz. Dimensions (H × W × D): $6^1/_2 \times 17 \times 22^1/_2$ in ($165 \times 432 \times 571$ mm). Weight: 49 lb (22.2 kg).

Air-cooled converter	Type: Piezoelectric—PZT—lead zirconate titanate crystals. Diameter: $2^{13}/_{16}$ in (71 mm). Length: $6^{1}/_{2}$ in (165 mm). Weight: 2 lb (900 g). Cable length: 10 in (3 m).
Booster	BHNVC 15—increases amplitude by 50 percent.
Standard probe	Tip diameter: 1 in (25 mm). Solid or with threaded end and replaceable tip. Please specify. Processing capability: 2 L—up to 15 L when used in conjunction with an extender and a mechanical mixer. Length: 5 in (127 mm). Weight: 1 lb (450 g). Titanium alloy: TI-6AL-4V.
Electrical requirements	220 volts, 50/60 Hz
Continuous flow cell (optional)	Weight: 9 lb (4.1 kg). Housing: 316 stainless steel. Quick opening clamps. Probe: $^{3}/_{4}$ in (19 mm) with threaded end and replaceable tip. Titanium alloy TI-6AL-4V.

V

Valves (*see* Control Systems)

Vanes (*see* Metallurgy)

Vaporizers; Vaporizer Applications*

Applications are easier to discuss with specific reference to certain models. (See Figs. V-1 through V-6.)

Types of Vaporizers

- Vertical vaporizer ("vertical bayonet"): It is widely used for chlorine, ammonia, propane, methanol, sulfur dioxide, etc. Sizes range from 50,000 to 15,000,000 Btu/h (12,500 to 3,750,000 kcal/h). Very compact, high productivity, easily combined with built-in superheater with common control. Many heating media can be used, including steam, hot water, and heat transfer fluids such as Dowtherm, Therminol, etc. Electric heated vaporizers also available. Small footprint. (See Fig. V-7.)
- Indirect fluid heater: Very useful for high-pressure or corrosive fluids where special metallurgy (i.e., corrosion-resistant metals) can be used in smaller, less costly containment than traditional shellside boiling. Heating medium (steam/dowtherm/electric, etc.) heats an intermediate bath of water/NH3/Therminol or similar heat-transfer fluid that then heats a second coil at much lower cost than shellside heating or boiling. (See Fig. V-8.)
- Tubular low-temperature vaporizers/superheaters: Combination large flow rate liquid heatup and subsequent boiling or superheating of mixed fluids with diverse boiling points. Needs special stress analysis and mechanical design. Can preheat, boil, and superheat in same vessel. (See Fig. V-9.)
- Impedance electric heaters: Electric heater for process fluids. Lowest cost heater for life of equipment. Easily cleanable, very safe, very long life, simple maintenance, good for high temperature boiling/heat to 2000°F (1093°C), very useful for remote locations of corrosive fluids or gases. Electric current flows through the containment tube and generates heat that is transferred to the fluid. (See Fig. V-10.)
- Electric resistance vaporizers: Classic reboilers or submerged resistance heating elements in normal shells for pool boiling duty. Useful for low to medium capacity loads and more common metals of construction. Can have combined superheat coils in the same containment vessel. (See Fig. V-11.)
- Vaporizers with controls: Indirect fluid electric vaporizer with controls mounted. Very high-pressure heater of corrosive fluid. Fluid side 3175 psi (223 kg/cm^2 or 216 ATM) design pressure. (See Fig. V-12.)

---

* Source: Armstrong Engineering Associates, USA. Adapted with permission.



FIG. V-1 Very large low-temperature vaporizer/superheater, steam heated. Unit size 132 in (3353 mm) in diameter by 42 ft (13 cm) long. Duty to boil and superheat very low-temperature organic liquid. (Source: Armstrong Engineering Associates.)

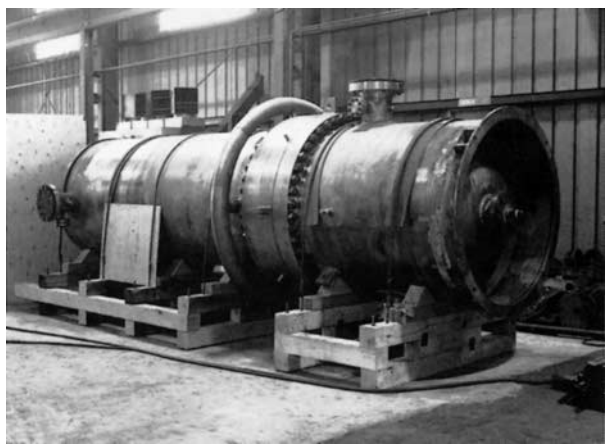


FIG. V-2 One of six all-stainless-steel vertical vaporizers for vertical location above tower in Mideastern refinery. Unit size 59 in (1500 mm) in diameter by 20 ft (6100 mm) high. (Source: Armstrong Engineering Associates.)

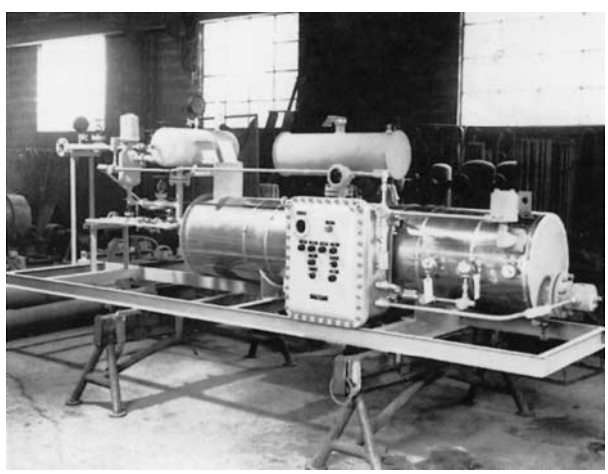


FIG. V-3 Indirect fluid electric heater insulated and mounted on skid with all controls in place. One of several at the same site in South America, vaporizing organic fluids. (Source: Armstrong Engineering Associates.)

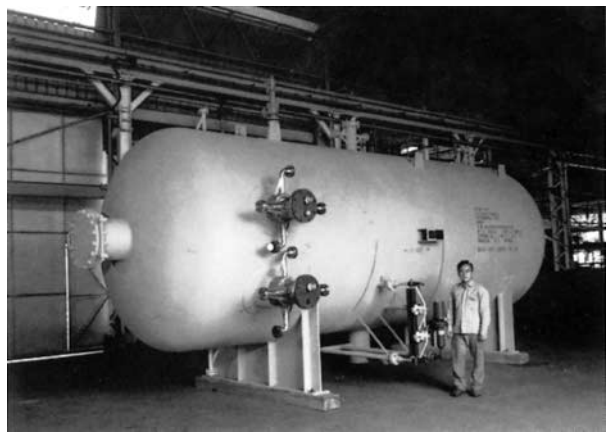


FIG. V-4 Large-sized ethylene vaporizer (steam in shell) with entering temperature of ethylene –67°F (–55°C) installed in India. Steam-heated steel shell with stainless steel heat-transfer surface. (Source: Armstrong Engineering Associates.)



FIG. V-5 Seven large hydrocarbon fluid vaporizers for chemical plant in Middle East. Unit size 36 in (914 mm) in diameter, overall height 135 in (3430 mm). (Source: Armstrong Engineering Associates.)

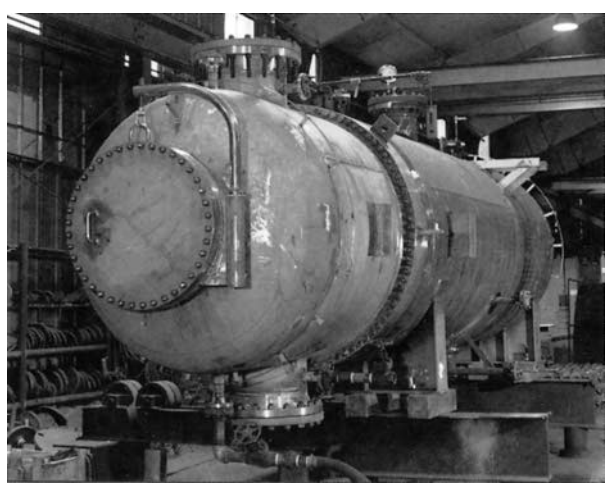


FIG. V-6 In-tube cryogenic vaporizer/superheater with steam in shell and mixed hydrocarbons on tubeside. Steel shell with balance stainless steel for U.S. installation. (Source: Armstrong Engineering Associates.)

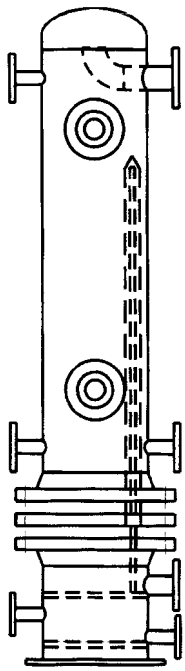


FIG. V-7 Vertical bayonet. (Source: Armstrong Engineering Associates.)

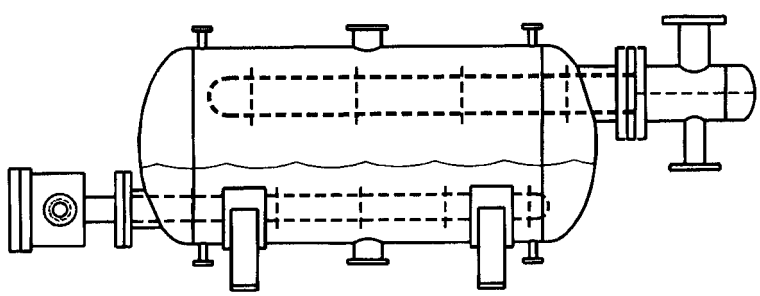


FIG. V-8 Indirect fluid heaters. (Source: Armstrong Engineering Associates.)

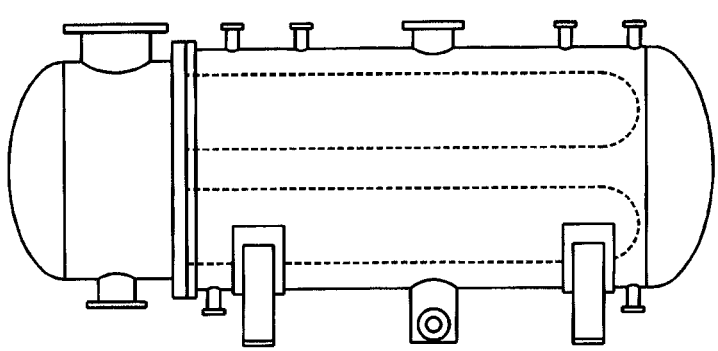


FIG. V-9 Tubular low-temperature vaporizers/superheaters. (Source: Armstrong Engineering Associates.)

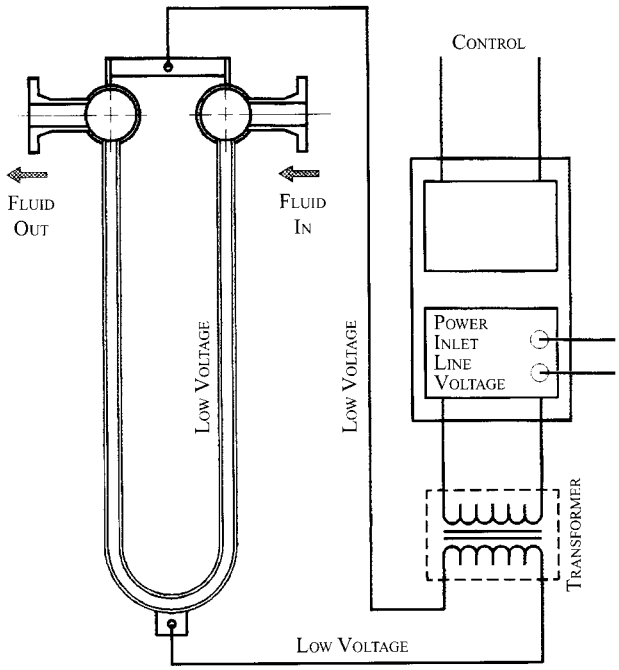


FIG. V-10 Impedance (electric) heaters. (Source: Armstrong Engineering Associates.)

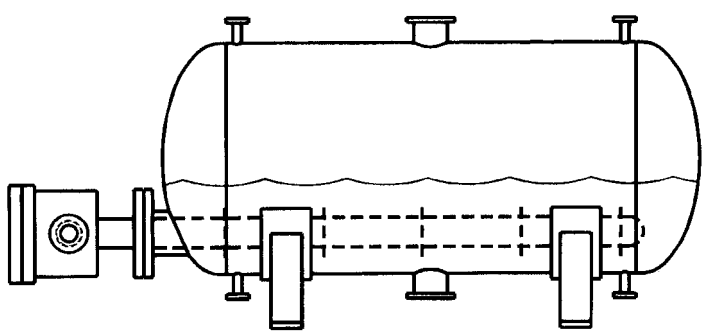


FIG. V-11 Electric resistance vaporizers. (Source: Armstrong Engineering Associates.)

- Electrical radiant furnaces: Radiant furnaces for high-temperature boiling levels of corrosive fluids or heating up to very high exit temperatures above fluid heating media capability [i.e., 2000°F (1093°C)]. Also for very high-pressure or corrosive fluids. Sizes from 12 to 50 ft (3.6 to 15.3 m) high. Can be very high capacity [some about 15,000 kW (12,900,000 kcal/h)] near nuclear site. (See Fig. V-13.)
- Cryogenic vaporizer: For boiling very low temperatures [–327°F (–200°C)]. Flare drum duty, to meet a few second startup emergency. Heating medium in shell and boiling fluid inside the tubes. Must be able to cope with thermal expansion and adjustments in a few seconds without damaging stresses. Also must avoid metallurgical problems including fatigue (cycling) for duties at high pressure such as ethylene, etc. Avoid freeze-up problems and heat up the fluid to required exit temperatures with no accompanying freezeup problems. Also, used to heat subzero fluids being distributed on service grids to multiple users and cold fluids

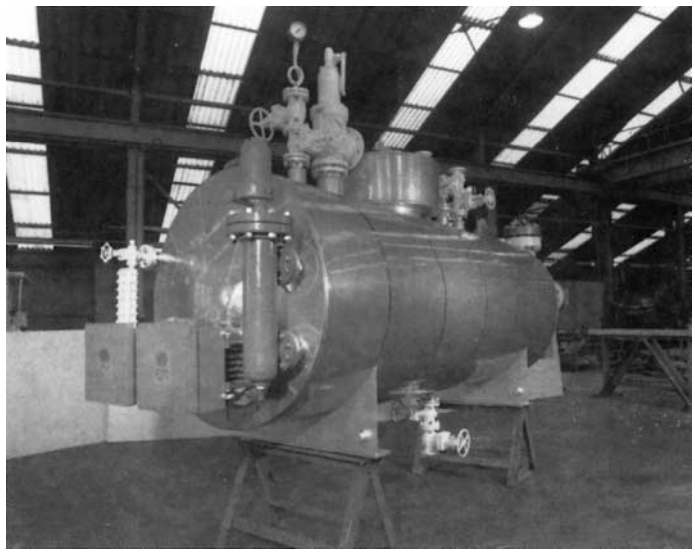


FIG. V-12 Vaporizers with controls or on skids with controls mounted. (Source: Armstrong Engineering Associates.)

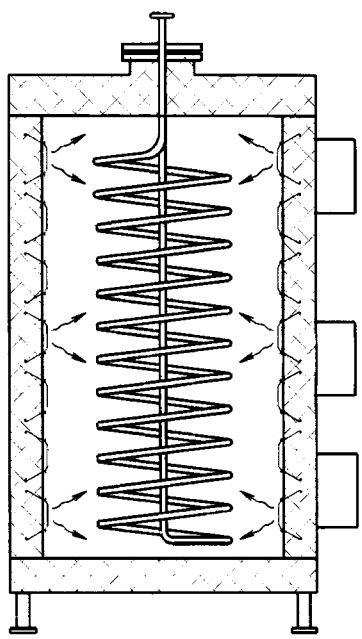


FIG. V-13 Electrical radiant furnaces. (Source: Armstrong Engineering Associates.)

from ships or rail cars needing heatup to avoid fracture of steel or other nonductile piping systems of user. Sizes can be up to 12 ft (3.6 m) in diameter and 40 ft (12 m) long. Shells often steel with tubes of stainless steels 304/316, etc. (See Fig. V-14.)

Vaporizer Specifications and Process Parameters

See Tables V-1 and V-2.

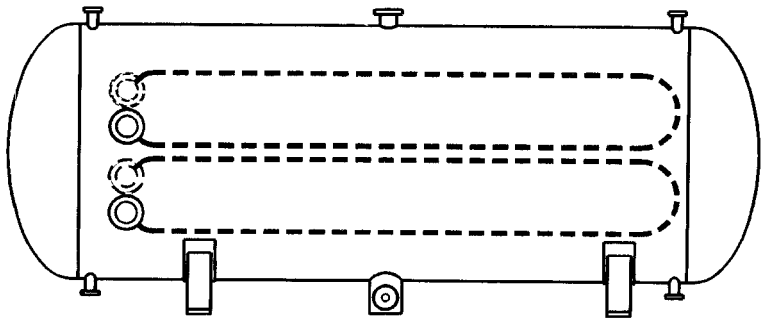


FIG. V-14 Extremely low temperature. (Source: Armstrong Engineering Associates.)

TABLE V-1 Useful Conversion Factors

	U.S. or British Units	SI and Metric Units	Multiplier to Convert	
			SI/Metric to British	British to SI/Metric
Heat duty	Btu/h	Watts	3.4144	0.29288
	Btu/h	kcal/h	3.9683	0.2520
Pressure	psi	kg/cm^2	14.223	0.0703
	psi	Bars	14.504	0.0689
	psi	Pa	$1.4504(10)^{-4}$	$6.8948(10)^{3}$
Velocity	ft/sec	m/s	3.2808	0.3048
Volumetric flow rate	gal/min (US)	m^3/s	$1.585(10)^{4}$	$6.309(10)^{-5}$
Mass flow rate	lb/hr	kg/s	$7.936(10)^{3}$	$1.260(10)^{-4}$
Density	lb/ft^3	kg/m^3	0.06242	16.018
Heat capacity	Btu/lb—F	J/kg—K	$2.3901(10)^{-4}$	$4.1840(10)^{3}$
Enthalpy	Btu/lb	J/kg	$4.302(10)^{-4}$	$2.324(10)^{3}$
	Btu/lb	kcal/kg	1.8000	0.55556
Viscosity	$lb_m/hr—ft$	cp	2.419	0.4134
Thermal conductivity	Btu/h—ft—F	W/m—K	0.57818	1.7296
	Btu/h—ft—F	kcal/h—m—C	0.67197	1.4882
Heat flux	$Btu/h—ft^2$	W/m^2	0.3172	3.1525
	$Btu/h—ft^2$	$kcal/h—m^2$	0.3686	2.7125
Heat-transfer coefficient	$Btu/h—ft^2—F$	$kcal/h—m^2—C$	0.2048	4.8824
	$Btu/h—ft^2—F$	$W/m^2—K$	0.17623	5.6745

Terminology: cm = centimeter, C = degrees Celsius, cp = centipoise, F = degrees Fahrenheit, ft = feet, gal = U.S. gallons, h = hour, J = joules, K = degrees Kelvin, kcal = kilocalories, kg = kilograms, m = meter, min = minute, Pa = pascals, s = seconds, W = watts.

Example: To convert heat duty in Btu/h to kcal/h, multiply Btu/h × 0.252, e.g., 15,000,000 Btu/h × 0.252 = 3,780,000 kcal/h. To convert velocity in ft/s to m/s, multiply by 0.3048: Velocity of 5 ft/s × 0.3048 = 1.524 m/s.

Vertical bayonet vaporizers

See Fig. V-15 and Table V-3.

Specifications

Shell. Ruggedly fabricated welded steel. Shells 24 in (610 mm) and below are made of SA-106 Gr.B pipe. Larger shells are welded of steel plate of SA-516 Gr.70 normally. Tubesheets are normally of SA-516 Gr.70 material, but are also available in stainlesses, nickel alloys, Hastelloy, etc.

Tube bundle. Removable on all sizes if required. Standard design of size "A" and "B" units may not have a removable tube bundle. Tubes are normally 1 in (25.4 mm)

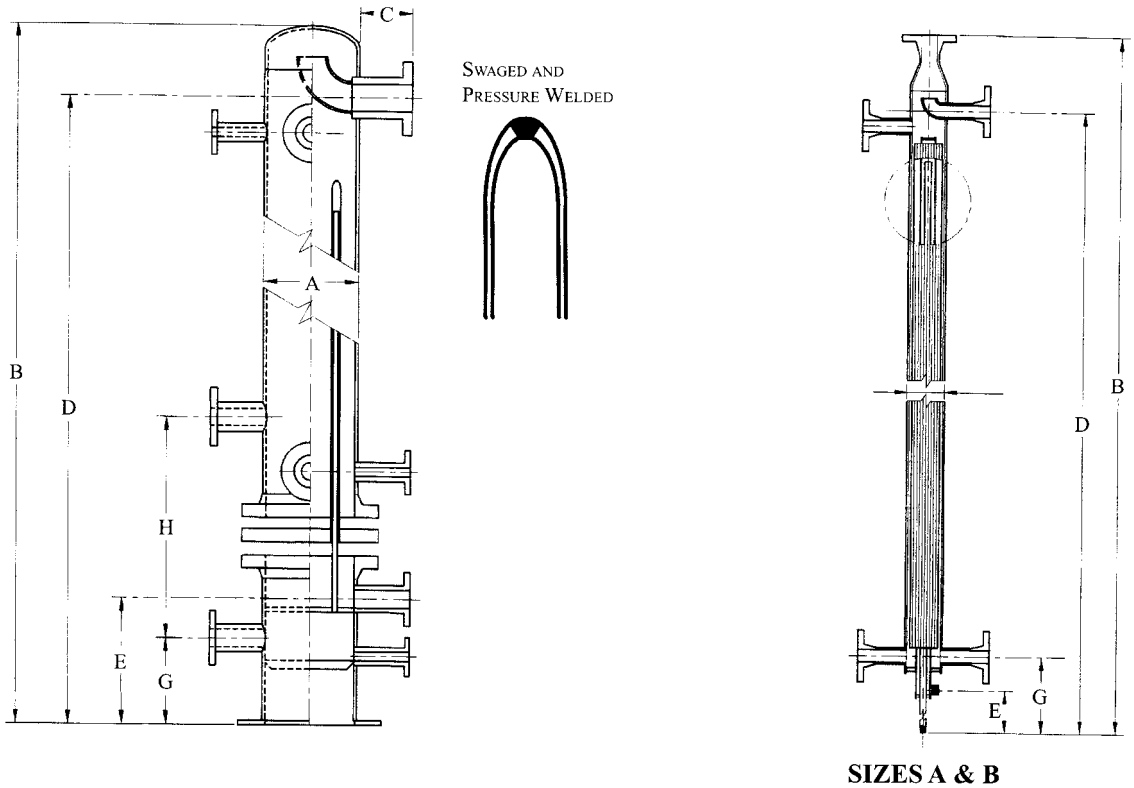


FIG. V-15 Vertical bayonet vaporizer dimensions. (Source: Armstrong Engineering Associates.)

TABLE V-2 Comparison of American, British, German, and Japanese Material Specifications

Material	American (ASME/ASTM)	German (DIN)	British (BS)	Japanese (JIS)
Plates	SA 516 Gr 60–70	A St 45–52 DIN 17135	BS 1501-224-490	JIS G 3118 SGV 49
	SA 515 Gr 60	H II DIN 17155	BS 1501-161-430	JIS G 3103 SB 42
Pipes	SA 53 Gr B	ST 45 DIN 1629	BS 3601 HFS-430	STPG42-G3454
(seamless)	SA 106 Gr A	ST 35.8 DIN 17175	BS 3602 HPS-360	STPT 38-G3456
	SA 106 Gr B	ST 45.8 DIN 17175	BS 3602 HFS-430	STPT42-G3456
Tubes	SA 214	ST 37.2 DIN 1626	BS 3059 ERW-320	STB 35E-G3461
	SA 179	ST 35.8 DIN 17175	BS 3059 CDS-320	STB 30S-G3461
Forgings	SA 105	C22.8Vd TUEV 350/3	BS 1503-221-490	S25C-G4051
Studs	SA 193 GR B7	21CrMoV57 DIN 17240	BS 4882 Gr B7	SNB7-G4107
Nuts	SA 194 GR 2H	24CrMo5 DIN 17240	BS 4882 Gr 2H	S45C-G4051

O.D., 0.083 in (2.1 mm) Bwg. but can be changed to meet customer specifications. Top ends of tubes are securely welded shut on all units. Tubes on sizes A and B normally have external longitudinal fins in contact with liquid being vaporized, multiplying the external surface about eight times, but can also be supplied with bare internal heating tubes.

Tubes are welded to the tubesheet and then rolled and expanded for additional holding power. Rolled joints alone are not sufficient for extended periods of service. For special services, these tubes can be of steel, stainless steel, or other materials. Bayonet tubes are roller expanded into lower tube plate.

TABLE V-3 Approximate Dimensions* (for Steam-Heated Vessels) (Dimensions Are in Millimeters)

Size	Base Plate Dia.	Shell O.D. "A"	Overall Height "B"	Outlet Projection "C"	Height of Outlet "D"	Cond. Height "E"	Inlet Height "G"	Outlet Height "H"	Steam Inlet	Cond. Outlet	Liquid Inlet	Vapor Outlet	Float & Safety
A	N/A	89	1867	152	1715	64	165	n/a	19	13	25	38	38
B	N/A	114	1981	152	1880	127	229	n/a	13	13	38	38	38
BT	203	114	1930	152	1753	191	127	343	25	25	38	38	38
C	305	168	2134	152	1930	337	270	378	38	38	51	51	51
D	305	219	2134	152	1930	356	270	378	38	38	51	51	51
E	413	273	2134	152	1930	356	270	378	51	51	64	64	51
F	413	324	2159	152	1930	330	229	419	76	76	76	76	51
G	508	406	2134	152	1930	311	203	479	102	51	102	102	51
H	660	508	2337	152	2070	368	270	498	102	51	102	102	51
I	762	610	2337	152	2057	368	270	537	102	51	102	102	51
J	889	762	2946	203	2578	648	508	737	152	76	102	152	51
K	1041	914	3200	203	2718	737	559	864	203	102	102	203	51
L	1194	1067	3404	203	2832	889	660	826	254	102	152	254	51
M	1346	1219	3556	203	2972	914	660	864	305	152	152	305	51

* Dimensions and outlet sizes may be varied to suit individual job conditions. Gauge connection is 3/4 in. All nozzles $2^1/_2$ in and over are flanged.

NOTE: Outlets with screwed connections also available upon request (at lower cost).

TABLE V-4 Approximate Steam Consumption

U.S. units: One Pound of Steam per Hour Will Vaporize		Metric units: One kg/h Steam Will Vaporize
1.0 U.S. gallon/h	Propane Butane	8.4 L/h
2 lb/h	Ammonia	2 kg/h
8 lb/h	Chlorine	8 kg/h
6 lb/h	Sulfur dioxide	6 kg/h

Design working pressure. On the process side, normally 250 psi (17.6 kg/cm^2). In steam or hot water space, 100 psi (7 kg/cm^2) (higher pressures available if needed). All vaporizers built in the U.S. are designed, inspected, and National Board stamped in accordance with the ASME Code. Vaporizers built outside the U.S. may be supplied per ASME, TUV, Stoomwezen or other local codes as required.

When operating pressure goes below 25 psi on propane or butane, check with the factory to avoid difficulty from pressure drop through nozzles. (See Table V-4.)

The 1-in-O.D. (25.4 mm) × 0.083 in (2.1 mm) wall tubes, seal-welded and rolled, give more clearance for condensate and steam. Thicker tube used in design adds to the life of the bundle. The seal-welding, and rolling gives strength needed against the fairly rapid variations in pressure and temperature encountered under some conditions. To avoid a loss of process fluid through leakage (and the peril of a potential explosion), the OEM seal-welds the tubes to the tubesheet.

See Figs. V-16 through V-18.

Design features

The tubes are free to expand or contract. Since the tubes are only secured at the bottom end, there is no tendency for the tubes to flex or twist from temperature stress. This is a marked advantage over units with tubesheets at both ends, where repeated temperature stress may cause failure at the tube end.



FIG. V-16 16-in-diameter (406-mm) chlorine vaporizer. (Source: Armstrong Engineering Associates.)



FIG. V-17 Four typical vertical vaporizers for large Mideastern refiner. Note the inlet belts on three of the vaporizers, often used to improve shell side distribution for improved boiling and excessive tube impingement. (Source: Armstrong Engineering Associates.)

Bottom steam feed protects against freezeup. The condensate is constantly warmed by incoming hot steam or hot water (if that is the heating medium). Even though the vaporizing temperature in the shell falls below freezing temperature, the condensate does not run the risk of freezing with consequent bursting of a tube. For boiling below 25°F (–4°C), consult the information source.

The tube bundle is removable and can be replaced in the field. It is no longer necessary to remove the whole unit in the event that the tubes begin to corrode out. A replacement bundle can be bought and installed in the field with a minimum amount of downtime.

How the vaporizer works—unlimited built-in turndown

1. The vaporizer takes its feed from a storage tank or process plant output and the boiling liquid rises in the shell until the vapor outlet generated by the load is matched by the heat transfer to the submerged surface level at the time. At that point, it stabilizes and continues to boil at that level until the load changes. If the load rises, the level of the fluid goes up to give the added output needed. If

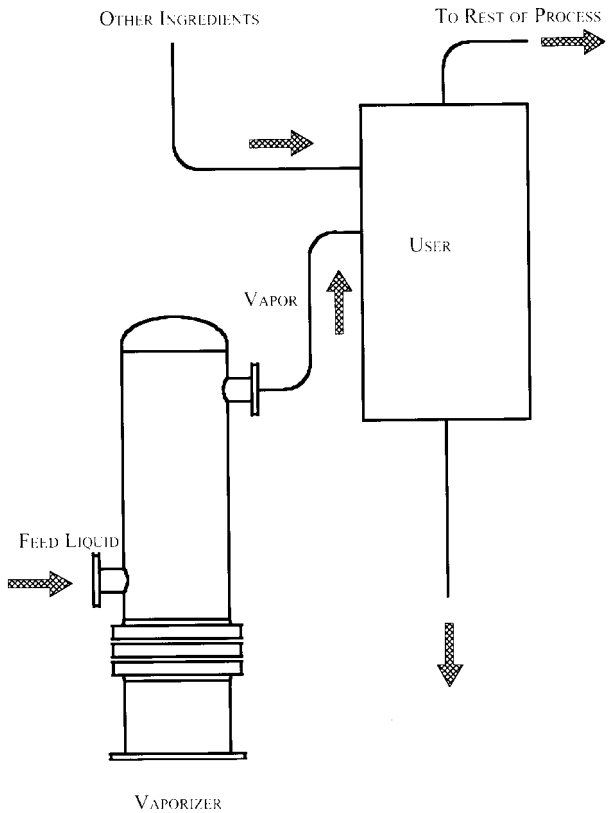


FIG. V-18 Vaporizer in process flow. (Source: Armstrong Engineering Associates.)

the load drops, the fluid level in the vaporizer drops until the output matches demand. This automatic turndown applies to any operating level in the vertical vaporizer. No special turndown control is needed.

2. It is easy to include a superheat section by adding height to the bundle (added surface) sufficient to achieve the desired superheat. This is impossible in either jacketed shell or reboiler-type vaporizers without the addition of a separate superheating element at substantial added cost.
3. These units protect against freezeup when boiling near the freezing point of the steam of other heating medium. The vaporizer can operate with boiling temperatures somewhat below the freezing point. The bottom steam feed protects the tubes against freezing so that the vaporizer can operate at boiling temperatures below the freezing point of the heating medium condensate. Consult factory for specific design figures in such cases.
4. The hold-up volume of process fluid is well below that in other types of vaporizers.
5. The footprint of the vaporizer is less than any horizontal unit. It is also normally lower than any other type of vertical in-tube or jacketed unit because of the greater output of the vaporizer.
6. The vaporizer is a standardized design and preliminary layout drawings are available early to enable plant layout to proceed quickly.
7. The tube bundle is removable and can easily be replaced or changed in the field.

8. The tubes are secured only at one end and are free to expand or contract so there is no thermal stress originating due to temperature variations in the bundle.
9. Code approval is normally easy since almost all code supervision agencies in the world have experienced submissions of vaporizers in past years.

Freeze-up protection with bayonet-type vaporizers

Controls and recommendations. During normal operation, the vertical bayonet design is excellent for vaporizing fluids at temperatures of 32°F (0°C) or several degrees lower. The leaving condensate is constantly warmed by incoming hot steam. The following recommendations are based on operating experience of vaporizers for propane, ammonia, chlorine, etc.

Precautions against freezing of steam condensate

Steam failure. The steam controls should be arranged such that the steam cannot be shut off at any time when cold process liquid can be in the shell at or below 32°F (0°C) and the operating instructions to personnel should stress this fact.

As an example, if a thermostatic steam valve or similar control is used in the inlet steam line, it should be limited in such a way that it cannot shut off completely when the process fluid in the shell is below 32°F (0°C). A hand valve in the steam line as a bypass around the control valve may be used to provide a positive steam supply.

Startup procedure would be to first establish steam supply to the unit before permitting cold process liquid to enter the shell, and shutdown procedure would be to first stop the process fluid flow before stopping the steam. If there is a failure of the steam supply, some precaution is desirable to stop the process fluid flow and to immediately remove the cold process fluid from the shell.

Suggestions would include a temperature control switch in the condensate line to sound an alarm and/or stop process fluid flow. A control indicating steam pressure failure may also be used.

Condensate backup. The steam and condensate lines must be free draining. In the case of a condensate return line to the boiler, care must be taken that the steam pressure is high enough to avoid a static head in the condensate line, which may result in backing up of condensate into the steam space of the vaporizer. This condensate may then freeze if cold process fluid is present in the shell.

Steam trap. The steam trap must be adequately sized to avoid backup. Also, a trap with minimum holdup of condensate is preferable. If the steam fails, condensate will re-evaporate and return to the tubes, so an absolute minimum condensate volume in the trap is desirable. Thermostatic traps have proven satisfactory for many applications.

Separate trap on steam chamber. A separate trap is recommended to carry away condensate that forms in the steam feed line and in the steam chamber.

Trap not too high. The trap on the main condensate outlet should be installed enough below the vaporizer condensate outlet connection to avoid backing up of condensate inside the vaporizer due to equalizing loads.

Strainers on traps. The traps should be equipped with strainers to ensure foreign materials will not plug the trap.

Positive steam pressure. The steam should be operated at a high enough pressure to overcome any pressure loss in lines, valves, fittings, etc., and to ensure operation

of the steam trap. Typically a pressure of 5 to 15 psig (0.35 to 1.05 kg/cm^2) is used as a minimum.

Steam trap stoppage is, arguably, the single most prevalent cause of freeze-up in vaporizers. For critical installations, duplicate traps may be installed in parallel.

Superheating outlet vapor methods and reasons

Basic design of vaporizers includes:

- Sensible heat to warm up the liquid from storage temperature to the boiling temperature in the vaporizer.
- Latent heat to boil the liquid at vaporizer temperature and pressure.
- Superheat required to heat vapor from saturation temperature to some desired gas outlet temperature.

The vaporizer usually has enough surface, figured to operate below the liquid level, to preheat the liquid and boil it.

Any surface required to superheat must be above the boiling area.

Three basic approaches to superheat are used:

1. Extension of tube bundle above liquid boiling level to add superheating surface.
2. A completely separate external superheater can be used.
3. For many fluids, reducing the pressure of discharged saturated vapor will produce some superheat. This method may invite surging. Consult factory.

Extension of the tube bundle. This usually requires more surface than #2 (above) as the vapor velocity is lower. Control of the gas outlet temperature is somewhat difficult since there is only one steam supply. However, by maintaining the boiling in the vaporizer at a fixed pressure, and setting a steam control to a fixed gas outlet temperature, control is possible.

This method is often somewhat more compact and also less costly if the amount of superheat is not too great.

Reasons for superheating

- Superheat may be required where outlet vapor lines are long, uninsulated, or exposed to low temperatures, so that recondensation could take place. Initial superheat allows for line temperature losses and the vapor can be delivered intact at the pipeline outlet end.
- Some controls contain elements subject to freeze-up or damage at temperatures below 32°F (0°C). Superheat of the vapor will avoid this danger.
- Controlled superheat may often be required for process reasons.

Separate external superheater. This is most desirable in cases where a large amount of superheat is needed. Superheaters are often made with finned tubes, which give a less costly heat exchanger than one with bare tubes for this duty.

Control is simple.

See Figs. V-19 and V-20.

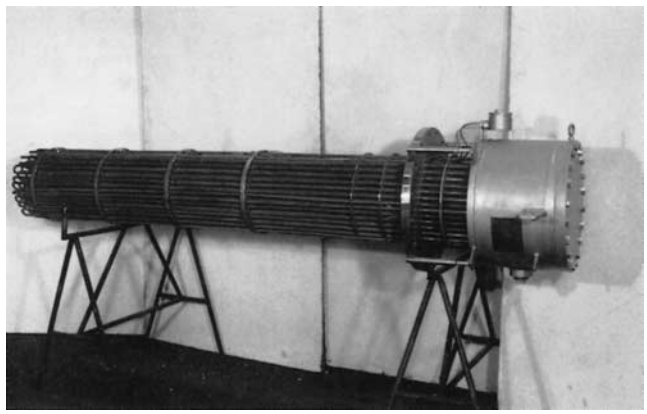


FIG. V-19 Typical electric superheater. (Source: Armstrong Engineering Associates.)

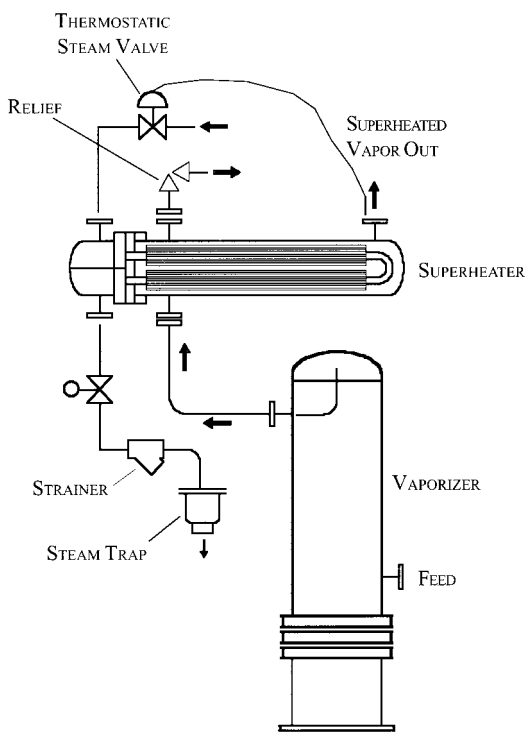


FIG. V-20 The above control hookup shows a typical external superheater. The advantage of the separate superheater is twofold. First, by having a separate steam feed on the superheater controlled by gas outlet temperature, definite temperature control can be gained. Second, the superheating surface is much cheaper as finned tubing than it would be in adding bare tube surface to the vaporizer bundle for boiling. Every installation using this system has been quite successful, although there are a number of jobs on which superheat was obtained merely by incomplete immersion of the bundle. This does take quite a bit more surface; however, each job should be individually calculated. (Source: Armstrong Engineering Associates.)

Some vaporizer applications

Liquid Vaporized	End Product	Typical Process
Acetaldehyde	N-Butyl Alcohol	Acetaldehyde based
Acetic acid	Acetic anhydride	Acetylene
	Acetaldehyde	Acetic acid based
	Vinyl acetate	Acetylene based
Ammonia	Acryonitrile	Sohio
	Ethanolamines	Ethylene oxide based
	Fertilizers	Various
	Nitric Acid	Ammonia based
Benzene	Maleic anhydride	Various
Butadiene	1,4-Dichloro-2-Butylene	Butadiene based
Butane	Acetaldehyde	
	Acetic acid	Oxidation of butane
	Acetone	
	Formaldehyde	
	MEK, Methanol	
	Propanol	
Butylene	Isoprene rubber	Prins
Chlorine	Aldrin, dieldrin	
	Endrin, isodrin	Shell
	Carbon tetrachloride	
	Perchloroethylene	Propane chlorination
	Chlorine dioxide	Various
	Chlorobenzenes	Various
	Chloromethanes	Methane based
	1,4-Dichloro-2-Butylene	Butadiene based
	Ethyl chloride	Ethane, ethylene based
	Ethylene dichloride	Ethylene based
	Ethylene oxide	Epichlorohydrin
	Glycerine	Propylene based
	Phosgene	Carbon monoxide based
	Propylene oxide	Epichlorohydrin
	Titanium dioxide	Various
	Vinylidene chloride	Acetylene based
Ethanol	Acetaldehyde	Ethanol based
Ethylene dighloride	Vinyl chloride	Ethylene dichloride based
Ethylene oxide	Ethanolamines	Ethylene oxide based
Formaldehyde	Isoprene rubber	Prins
Hydrogen chloride	Bisphenol	Various
	Ethyl chloride	Ethylene based
	Methyl chloride	Methanol based
	Neoprene	Acetylene based
	Trichloroethylene	
	Perchloroethylene	Acetylene based
	Vinyl chloride	Wulff
Hydrogen cyanide	Acrylonitrile	Acetylene based
	Acrylonitrile	Ethylene oxide based
Methanol	Dimethyl terephthalate	Hercules-Witten
	Formaldehyde	Methanol based
	Methyl chloride	Methanol based
Phosgene	2,4-Toluene diisocyanate	Toluene diamine

Propane	Acetaldehyde	
	Acetic acid	
	Acetone	Oxidation of propane
	Formaldehyde	
	MEK, Methanol	
	Propanol	
	Carbon tetrachloride	Propane chlorination
	Perchloroethylene	Propane chlorination
	Nitromethane	
	Nitroethane	Propane nitration
Propylene	Acetone	Wacker
	Acrolein	Propylene based
	Acrylonitrile	Sohio
	Glycerine	Propylene based
	Polypropylene	Various
Propylene oxide	Glycerine	Propylene oxide based
Sulfur trioxide	Alkyl benzene sulfonates	Various
Vinyl chloride	Vinylidene chloride	Acetylene based

Other fluids vaporized in vaporizers

Butanol	Methyl chloride
1-Butene	Methylene chloride
Butylamine	Nitrogen
Bromine	Oxygen
Carbon dioxide	Paracymene
Diethyl ether	Pyridine
Dimethyl ether	Refrigerant 11
Ethane	Refrigerant 12
Ethylene	Refrigerant 13
Hydrogen bromide	Refrigerant 13
Hydrogen fluoride	Refrigerant 223
Hydrogen sulfide	Sulfur dioxide
Isobutylene	Sulfur hexafluoride
LPG mixtures	Turbine fuels
Methylamine	Vinylidene chloride

Design criteria for various fluids. Capacity ratings and flow paths of vaporizers are based upon extensive field measurements of plant scale installations and also on much in-house testing of miscellaneous fluids over a long period of years.

Chlorine. Vertical vaporizers, often with built-in superheat capability, are used. Special instrumentation is required, specifically designed for chlorine service. Base designs are carbon steel. However, if during steamout or cleaning, etc., water is left in the chlorine space, acid is formed that will cause extremely rapid corrosion (hours), often resulting in failure. For this reason, tubes or tubesheets may be used in nickel alloys such as Monel, Inconel 600, and Incoloy 800.

Pamphlet 9 of the Chlorine Institute gives very useful recommendations for application of chlorine vaporizers including some references such as autoignition (rapid corrosion of steel at high temperatures when chlorine encounters a hydrocarbon at the steel surface). Consult the information source about possible difficulties due to concentration of nitrogen trichloride over a long period of time.

Ammonia. Vertical vaporizers, often with built-in superheat capability. Normally steel, but sometimes with stainless steel tubes or tubesheets, particularly if the feed material may have trace elements of a corrosive nature.

TABLE V-5 Atmospheric Boiling Temperatures of Typical Liquids Handled in Information Source's Vaporizers

Fluid	°F	°C
Ammonia	–28.0	–33.3
Chorine	–30.1	–34.5
Hydrogen chloride	–121.0	–85.0
Hydrogen sulfide	–75.3	–59.6
Sulfur dioxide	14.0	–10.0
Methane	–258.7	–161.5
Ethane	–127.8	–88.8
Propane	–43.7	–42.1
N-Butane	31.1	–0.5
Ethylene	–154.7	–103.7
Propylene	–53.9	–47.7

Miscellaneous liquids. Methanol, sulfur dioxide, refrigerants, random hydrocarbon mixtures, etc. Mostly handled in vertical vaporizers, often with built-in superheat capability. *Metals as required to suit individual needs*. Typical metals include Hastelloy, various stainless steels, Carpenter 20, Incoloy 800, Monel, Inconel 600, and various nickel alloys. Low temperature nickel steels are seldom used because of unavailability of small quantities of metal on short notice, plus costly fabricating practices.

Liquified petroleum (propane, butane). Normally in vertical units, sometimes in indirect water bath vaporizers. Normally all steel equipment.

Lethal fluids. Such as hydrogen sulfide, phosgene, hydrogen cyanide, xylol bromide, etc. Normally in vertical units, can be all welded (no gaskets anywhere) if preferred. Usually 100 percent radiographically inspected and heat treated after fabrication. Also available as heavy duty shell design with removable bundles.

Ethylene. Usually fed at low temperature to horizontal shell and coil type vaporizers at very low temperatures (–155°F or –104°C) so stainless steel or other high impact value material is normally used. Typical designs include steel shells with stainless steel vaporizing bundle designed to avoid surging and withstand thermal shock conditions.

Liquefied natural gas. Usually substantially methane, handled like ethylene. Sometimes vertical installations are required for shipboard application, involving approval for such codes as ABS, United States Coast Guard, Lloyds, Veritas, etc.

Cryogenics. Nitrogen, oxygen, low boiling hydrocarbons, etc. Similar to ethylene except sometimes temperatures for direct steam heating may be as low as –325°F (–198°C). Most materials are stainlesses. Special cleaning may be required for oxygen processes.

Freezeup protection. For details, see discussion later in this section. (See Table V-5 for atmospheric boiling temperatures of some typical liquids.)

Cryogenic vaporizers (direct steam heated)

Uses of cryogenic vaporizers. These are used on vaporizing process upset fluids such as ethylene, propylene, etc., on flare systems, where quantities exceed normal capacity of flare drums, or to vaporize ethylene, nitrogen, etc., for consumption out of atmospheric storage systems. They all also used on LNG tankers to vaporize nitrogen, for padding, for loading or transfer of cargo, or in areas between tanks to reduce explosion hazard.

See Figs. V-21 and V-22.

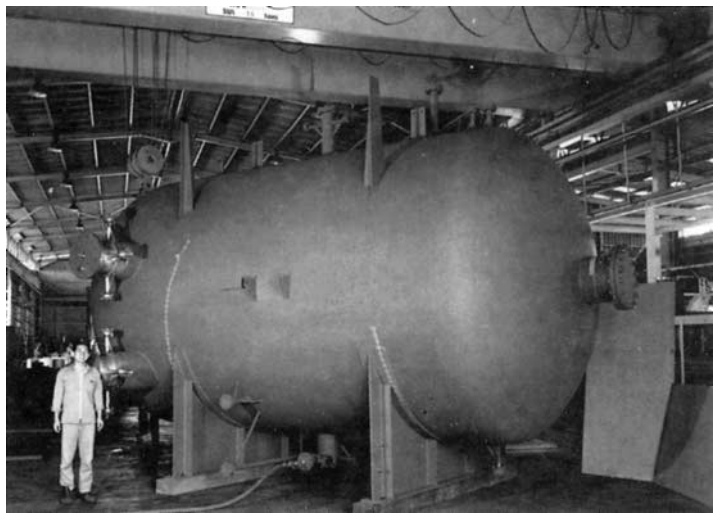


FIG. V-21 Large mixed hydrocarbon vaporizer for feed temperature approximately −150°F (−101°C) for installation in Eastern Gulf refinery. Unit size 132 in (3353 mm) in diameter × 23 ft (7000 mm) long. (Source: Armstrong Engineering Associates.)

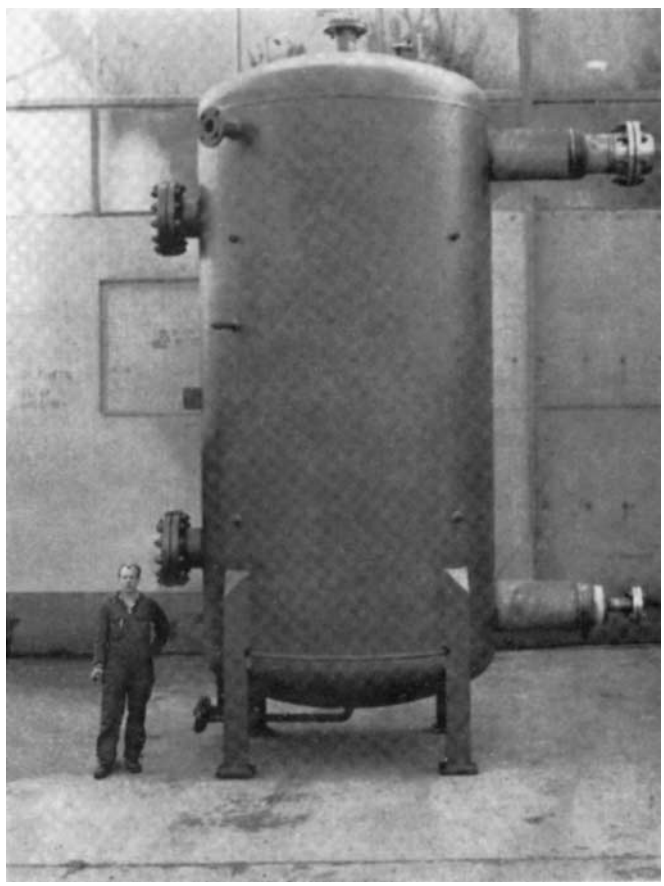


FIG. V-22 Vertical vaporizer/superheater with internal helical coil. Shell is steel with internals of stainless steel. 1412-kW (1,215,000-kcal/h) unit installed on LNG tanker to heat product and assist pump transfer from hold. Vertical format reduces footprint when necessary. (Source: Armstrong Engineering Associates.)

Direct steam heated vaporizers for fluids boiling as low as –327°F (–200°C). Ultra-low temperature vaporizers are designed to avoid freezeup of steam condensate since the steam is on the shellside and condensate is always in contact with the steam.

Indirect systems often require relatively larger equipment and much more costly instrumentation plus the maintenance and supervision that goes with that instrumentation. Direct heated vaporizers require simpler controls than indirect heaters plus avoid any danger of condensate freezeup except if the steam trap gets blocked and traps condensate inside the shell. Desuperheaters can be added to give maximum flexibility to exit vapor control temperature.

Direct heated cryogenic vaporizers have a long history of successful field experience, using direct steam as a heating medium to boil and superheat fluids from as low as –327°F (–200°C). None of these many direct-heated vaporizers, which are in a variety of fluid duties, has ever failed from freezeup of steam condensate to our knowledge.

Vaporizers have considerable antifatigue designs built in for grid loadout duties where the vaporizer may operate on highly fluctuating/cyclic flow rates.

The design and method is thoroughly proven from a great many field installations of sizes up to 40,000,000 Btu/h (10,000,000 kcal/h) per individual vaporizer.

Transfer heaters for very cold liquids antifreeze designs

Typical duties

1. Heating of liquid ammonia, ethylene, LPG, etc., for transfer from ship or other low-temperature storage to ordinary steel pipelines or shipping tanks.
2. Defrosting of fluid circulating systems.
3. Vaporizing oxygen, nitrogen, etc., at temperatures down to –327°F (–200°C) and pressures up to 6000 psi (422 kg/cm^2) using steam or hot water as a heating medium.
4. Heating of very corrosive fluids in separate tube bundles of metal such as Monel, Nickel 200, Inconel 600, Incoloy 800, silicon bronze, etc.
5. Very high-pressure liquid flow can be up to 10,000 psi (703 kg/cm^2 or 680 atm).

Often it is convenient to use a combined two bundle unit in a single shell, or two separate shells using steam to vaporize an intermediate fluid, for example methanol. The vaporized methanol then rises to the top bundle, heating up the fluid passing through the tubes. Since methanol is not subject to freezing, and the boiling temperature of the methanol is kept above 32°F (0°C), there is no freezeup hazard. See Figs. V-23 and V-24.

Small electric indirectly heated vaporizers

Electrically indirect heated vaporizers are suitable for boiling of ammonia and a number of other fluids. Shells are usually of steel, and the heating elements are often copper, although many other metals, such as stainless steels, Monel, Inconel, Hastelloy, and Incoloy can be supplied when requested.

These vaporizers can be supplied with or without controls. They offer a solution to vaporization in outlying areas where steam is not available.

Typical duties include vaporizing HF, H_2S, bromine, CO_2, SO_2, CH_3Cl, Cl_2, HCl, NH_3, LPG, C_3H_8, C_4H_{10}, etc.

See Figs. V-25 through V-28.

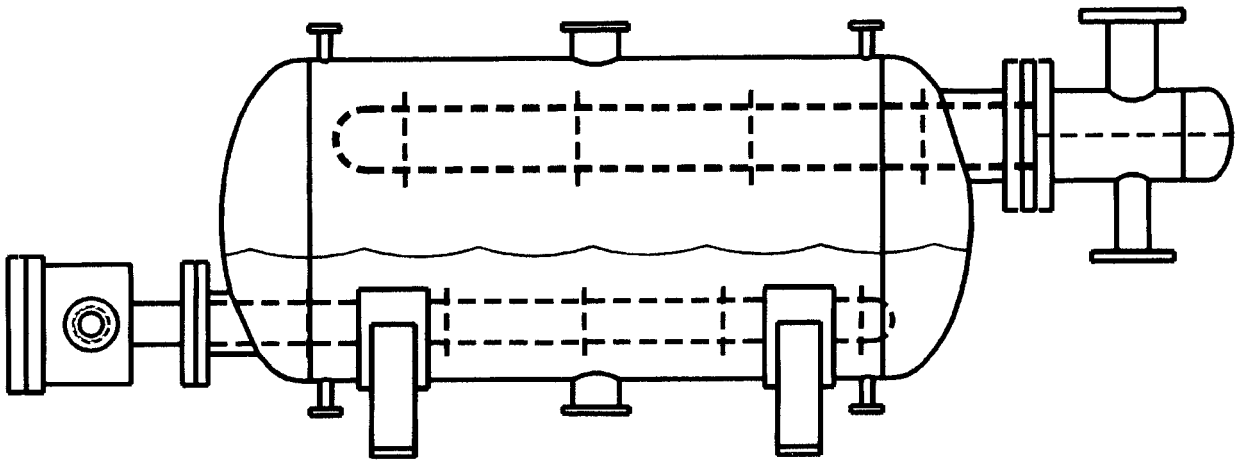


FIG. V-23 Intermediate fluid transfer antifreeze heater. (Source: Armstrong Engineering Associates.)

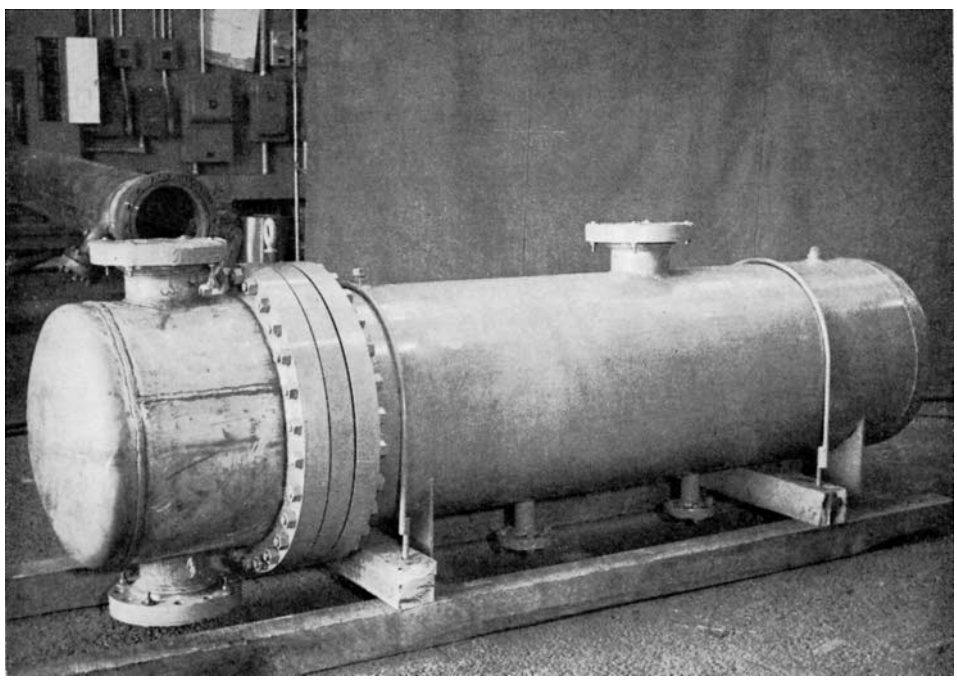


FIG. V-24 Direct heated transfer heater. Horizontal antifreeze heater for cold ammonia, entering temperature −28°F (−33°C) (or below if necessary). Steam heated. Steel and stainless steel construction. Removable bundle. Can also be electric heated. (Source: Armstrong Engineering Associates.)

Combination electric indirect vaporizers

In working with highly inflammable or explosive fluids, it may not be safe to put the electric element directly in contact with the fluid. Therefore, the information source supplies a combination unit consisting of a vaporizer, an electric water heater, and a pump, with controls.

The electrically heated water is circulated into the vaporizer to furnish the heat required for boiling. This same arrangement may be used for chlorine, or for any

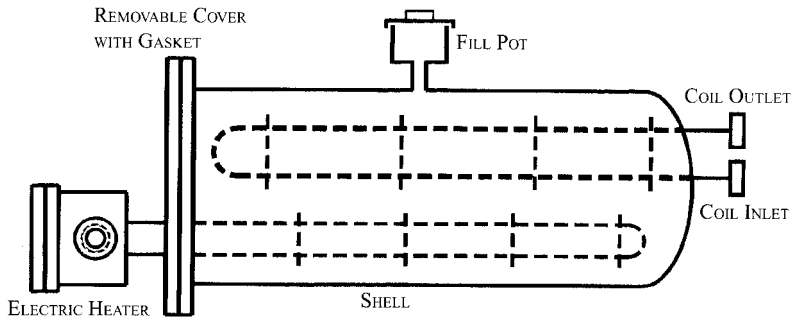


FIG. V-25 Typical arrangement for electric water heater skid mounted unit to vaporize whatever fluid desired. (Source: Armstrong Engineering Associates.)

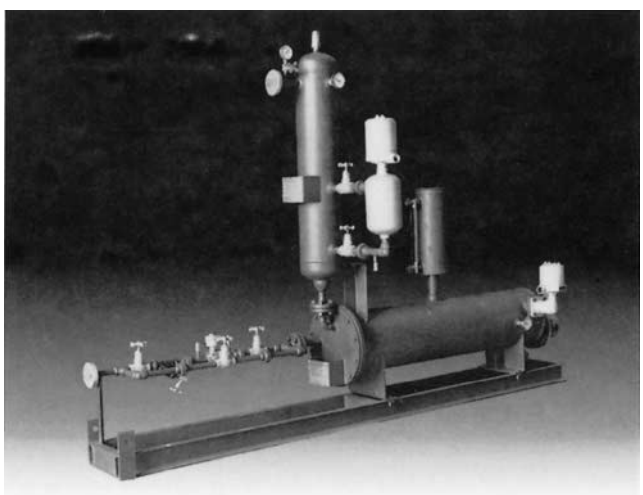


FIG. V-26 Electric indirect heated (glycol bath) vaporizer, for boiling of propane, LPG, HCl, SO_2, etc. (Source: Armstrong Engineering Associates.)

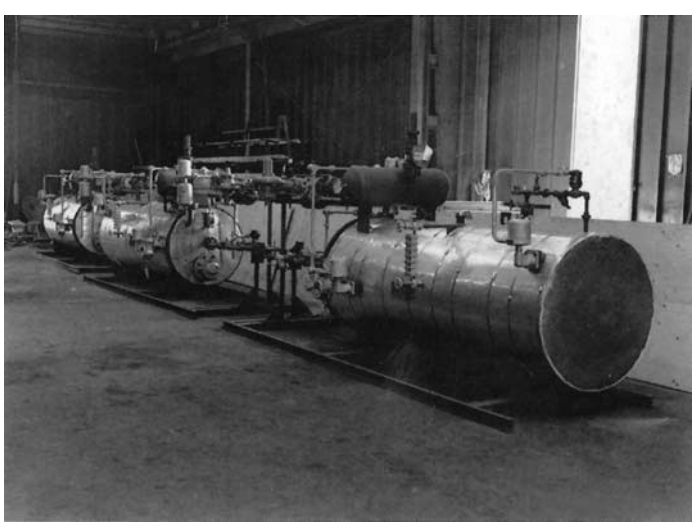


FIG. V-27 Line of 36 in (914 mm) × 8 in (2438 mm) indirect heated LPG vaporizers showing insulation and controls. Installed in large apartment complex in Hong Kong. (Source: Armstrong Engineering Associates.)

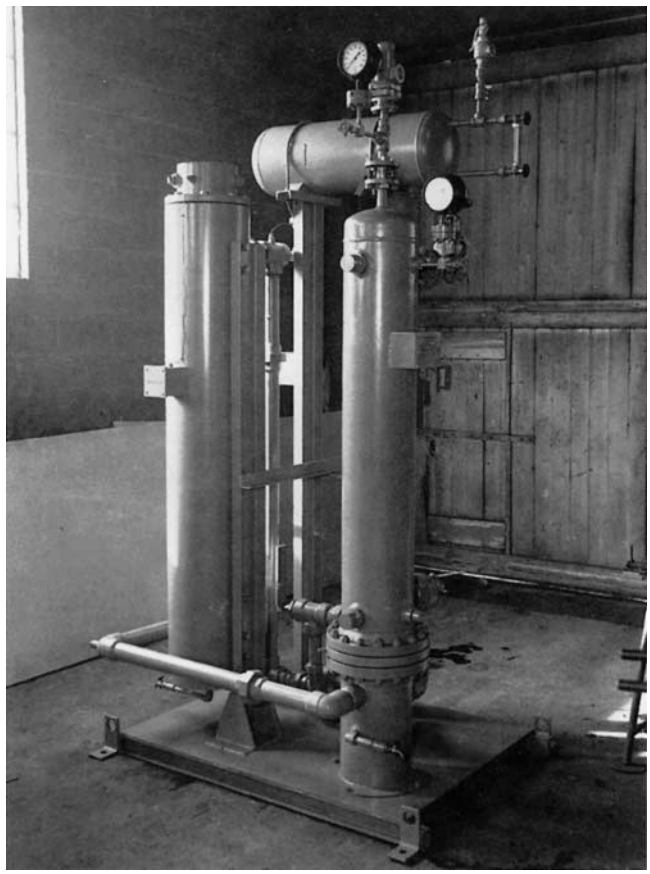


FIG. V-28 Electric heated vaporizer: Circulating pumped water bath. System includes electric immersion water heater, pump, piping, controls, and expansion tank, all factory piped up and delivered on skid ready to operate. (Source: Armstrong Engineering Associates.)

fluid that might offer severe corrosive attack to the sheath metal of the electric heater. This typed unit is often used for C_3H_8, Cl_2, C_4H_{10}, SO_2, Freons, etc.

See Figs. V-29 and V-30.

Pressurizing storage tanks

Where the outdoor storage temperature is very low, the saturation pressure of the stored liquid may get so low there is not sufficient pressure to deliver the liquid or vapor past the piping and valve resistances required. In such cases, a vaporizer may be used to furnish the heat required to keep the stored liquid at a desired temperature and corresponding pressure, even though the outdoor ambient temperature may be quite low. See Fig. V-31.

The most prevalent type of vaporizer for this purpose is the vertical bayonet. The main point to consider is whether there is a likelihood of damming or stoppage of the condensate inside the tube at a time when surrounding liquid is below 32°F (0°C). In that event, freezing is a danger and in such times, the antifreeze aspect of the vertical bayonet vaporizer becomes desirable.

The internal condensing capacity of the storage tank, even with only a fraction of the surface not covered by liquid, is tremendous, and would require both a very large vaporizer and also a very much larger boiler or heat source. Therefore, the only practicable way to approach this problem is to figure on boiling the liquid at

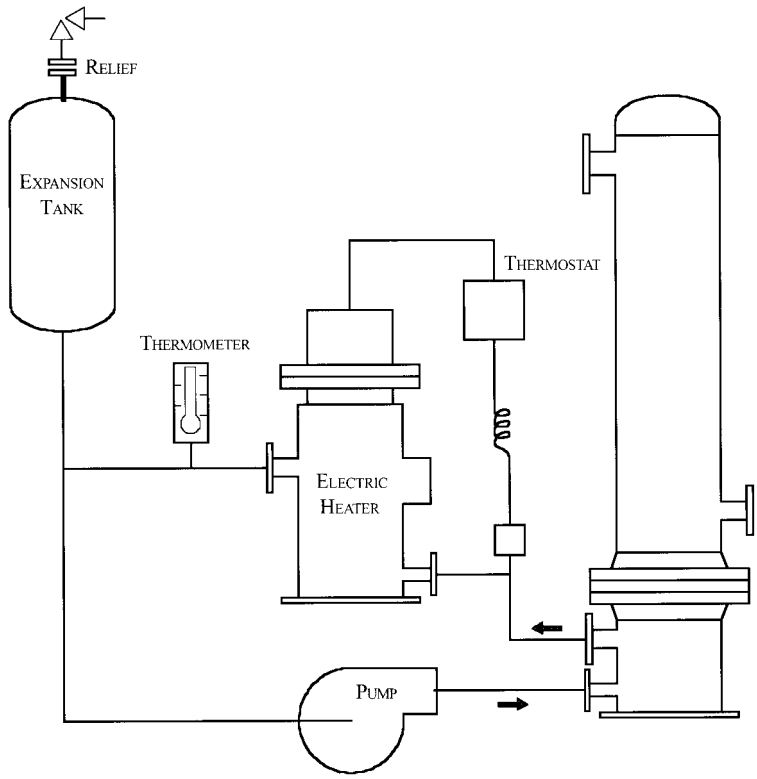


FIG. V-29 Electric indirect vaporizer in process and flow. (Source: Armstrong Engineering Associates.)

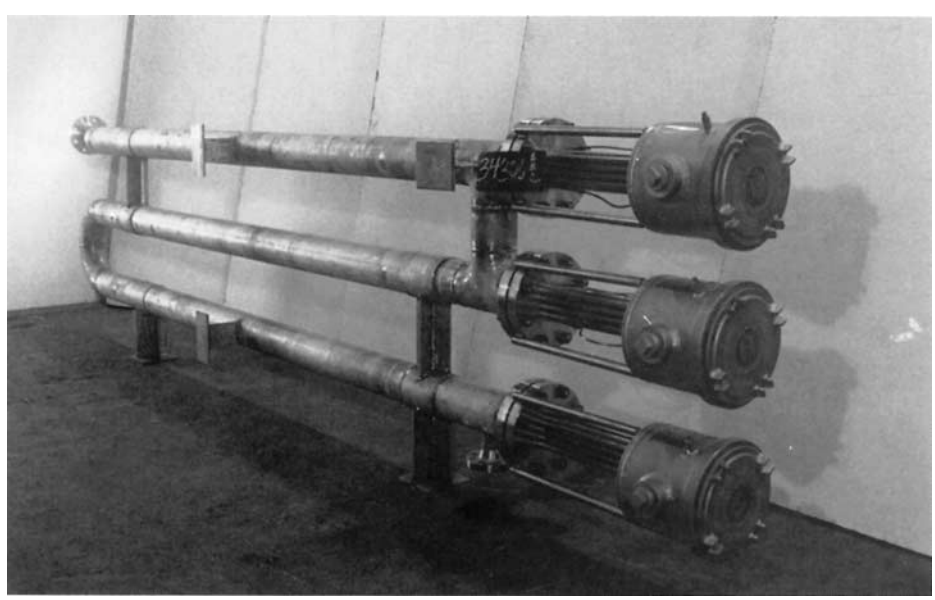


FIG. V-30 Direct electrically heated ammonia vaporizer. (Source: Armstrong Engineering Associates.)

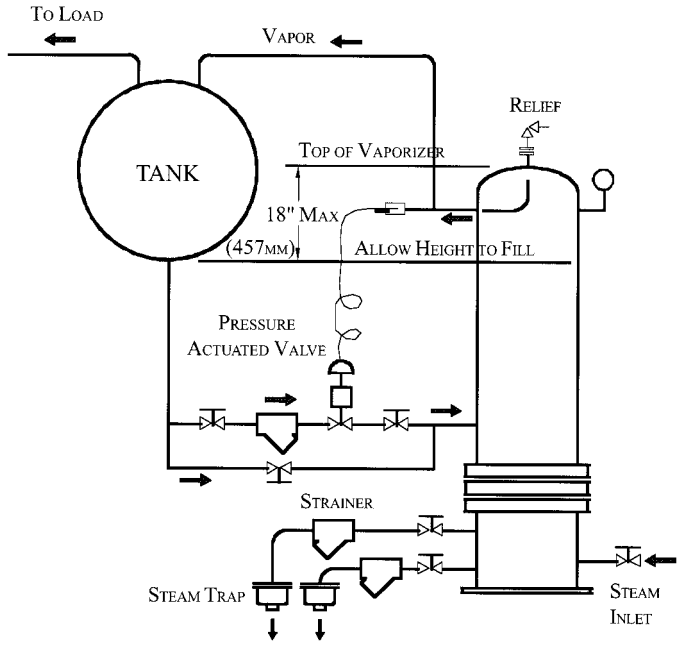


FIG. V-31 The above vaporizer hookup shows an arrangement that is often used for ammonia and other gases in addition to propane. The liquid level is important to make sure the liquid will flow by gravity or otherwise into the vaporizer. The pipelines must be large enough to overcome any hydraulic loss in the flow system, to make sure that the vaporizer tube bundle will be covered. Otherwise, at low levels, the vaporizer will not have full capacity. The pressure-actuated valve may not be absolutely necessary, but is desirable in many cases. Since it is usually better to keep the steam on at all times, this avoids excess boiling at times when the tank may already have adequate pressure. (Source: Armstrong Engineering Associates.)

the beginning of operation at the low temperature; get the whole body of the tank and its stored liquid up to the operating temperature, say 70°F (21°C), and then furnish enough heat to overcome the convection heat loss from the outside surface of the tank to the ambient air. Also, sufficient capacity is needed to heat the fresh incoming liquid as it arrives by tank car, assuming that it also has cooled down to the outdoor temperature.

To illustrate the application, take an example of a 30,000 U.S. gallon uninsulated storage tank, 9 ft in diameter by 63 ft long, to be filled with 10,000 U.S. gallons of liquid anhydrous ammonia in a 10-h day, assuming 70°F required tank temperature, and with an ambient outdoor temperature of –30°F. The outside surface area of the tank is 1827 ft^2. The convection heat loss can be taken conservatively as 5 Btu/h/ft^2/°F at a wind speed of 20 mph.

Ambient heat loss is then

$$1827 \times 5 \times [70 - (-30)] = 915{,}000 \text{ Btu/h}$$

Incoming fresh liquid (loading the tank):

$$1000 \text{ gph} \times 5.6 \text{ lb/U.S. gal} \times (120.5 - 10.7) = 615{,}000 \text{ Btu/h}$$

Note: 120.5 is enthalpy of liquid at 70°F (21°C); 10.7 is enthalpy of liquid at –30°F (–34°C).

Ambient heat load	913,500 Btu/h
Fresh liquid heatup	615,000 Btu/h
Total	1,528,500 Btu/h

To illustrate another application, we take an example of a 114-m^3 uninsulated storage tank, 2.74 m in diameter by 19.2 m long, to be filled with 38 m^3 of liquid anhydrous ammonia in a 10-h day, assuming 21°C required tank temperature (8.0 kg/cm^2G required pressure), and with an ambient temperature of –34°C. The outside surface area of the tank is 170 m^2. The convection heat loss can be taken conservatively as 24.4 kcal/h/m^2/°C at a wind speed of 8.9 m/s. Ambient heat loss is then

$$(170) \times (24.4) \times [(21) - (-34)] = 228{,}140 \text{ kcal/h}$$

Incoming fresh liquid (loading the tank in 10 h):

$$3.8 \text{ m}^3 \times 683.17 \text{ kg/m}^3 \times (66.94 - 5.94) = 158{,}360 \text{ kcal/h}$$

Note: 66.94 kcal/kg is enthalpy of liquid ammonia at 21°C; 5.94 kcal/kg is enthalpy of liquid ammonia at –34°C.

Ambient heat load	228,140 kcal/h
Fresh liquid heatup	158,360 kcal/h
Total	386,500 kcal/h

This duty is added to the amount of vapor withdrawn from the tank to process, if any.

Vaporizers with controls

Setting of float valves on vaporizers. The fluid that is boiling inside the vaporizer is not entirely liquid, but a mixture of liquid and vapor. As a result, its density is usually appreciably lower than the density of the liquid column, which acts upon the float in the float chamber, which is not subject to such vigorous boiling. See Figs. V-32 through V-39.

Therefore, the height at which the operating center of gravity of the float should be set should be lower than the actual boiling level in the vaporizer by an amount equal to the ratios of the densities of the liquids in the vaporizer to that in the float column. A good practice is to start at about 2/3 the height of the vaporizer level above its bottom, and then adjust the float level in the field to give best performance. Note that on vertical vaporizers, the float acts only as a limit to capacity. When the vaporizer is operating at less than its capacity, it automatically operates with a liquid level lower than the top of the tubes.

On all installations, if the propane or other liquid in the shell should get down to temperatures below 32°F (0°C), it is very important not to have a steam fallure. In this case, the vapor can condense and freeze and possibly burst a tube.

This is important to consider in any control that will shut the steam off and also on any system where the steam may fail for external reasons.

When desirable, vaporizers can be supplied with the controls mounted. We caution the prospective buyer in this case:

1. Since there is wide variety in preferences as to method, make, and type of control, it often takes several times as long to work up quotations on units with controls as it does to quote bare vaporizers.

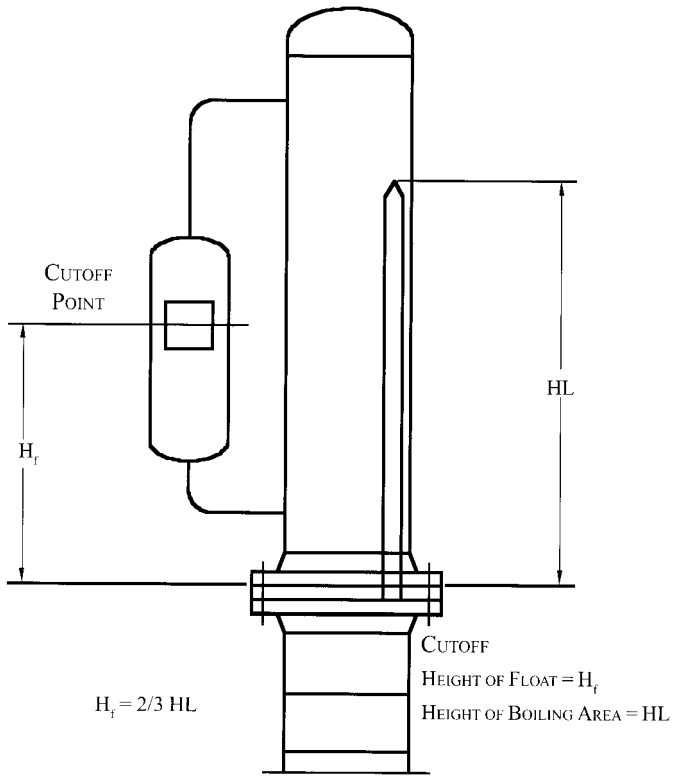


FIG. V-32 Float setting. (Source: Armstrong Engineering Associates.)

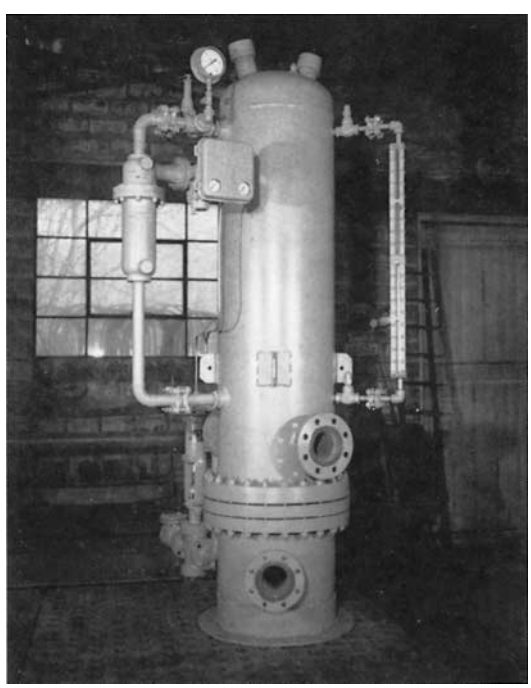


FIG. V-33 Process vaporizer with controls factory mounted. Note float and float-operated valve, liquid level gauge, thermostatic steam valve, condensate traps, bursting disc relief valves, and miscellaneous hand valves, including bypasses and strainers. (Source: Armstrong Engineering Associates.)

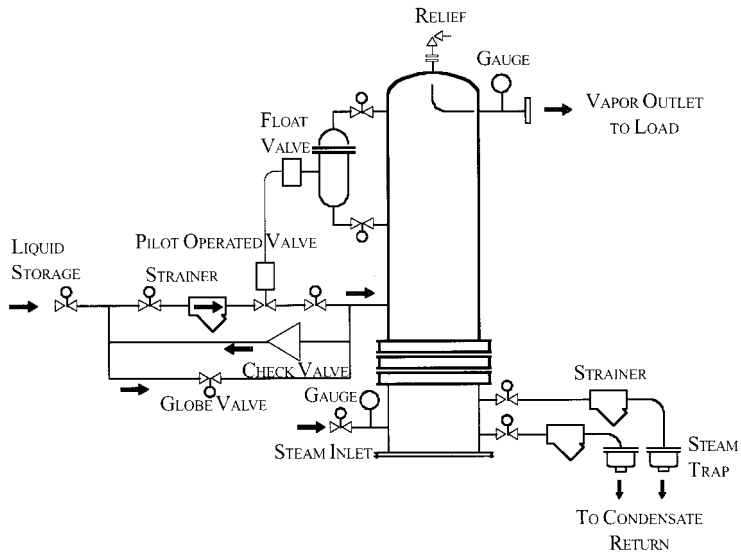


FIG. V-34 Typical control hookup for routine fluid vaporizing. Note presence of shellside float valve. This operates to stop liquid feed when flow level gets too high due to excessive draw of fluid from the vaporizer. The valve shuts down flow of the liquid feed to the vaporizer to avoid carryover of liquid into the exit line. (Source: Armstrong Engineering Associates.)

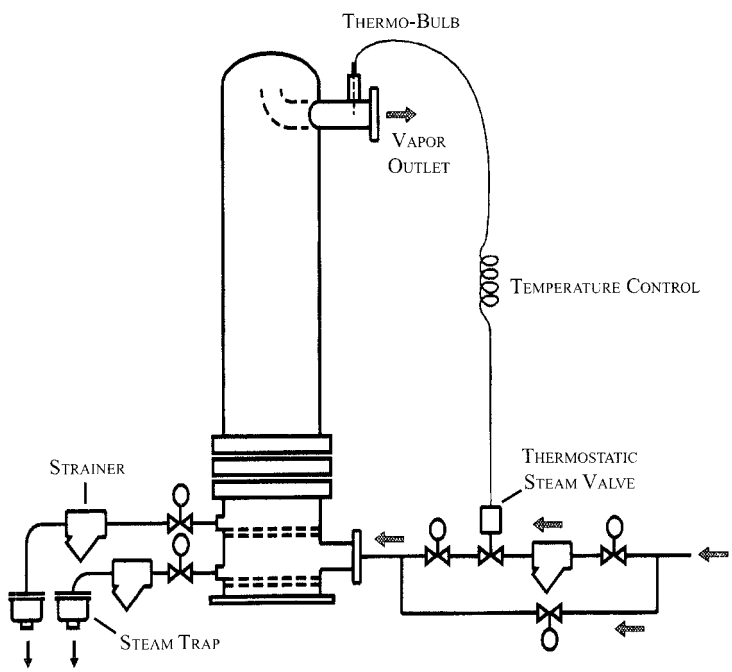


FIG. V-35 Typical control setup when specific superheated temperature is required. For normal chlorine vaporizing see separate diagram. (Source: Armstrong Engineering Associates.)

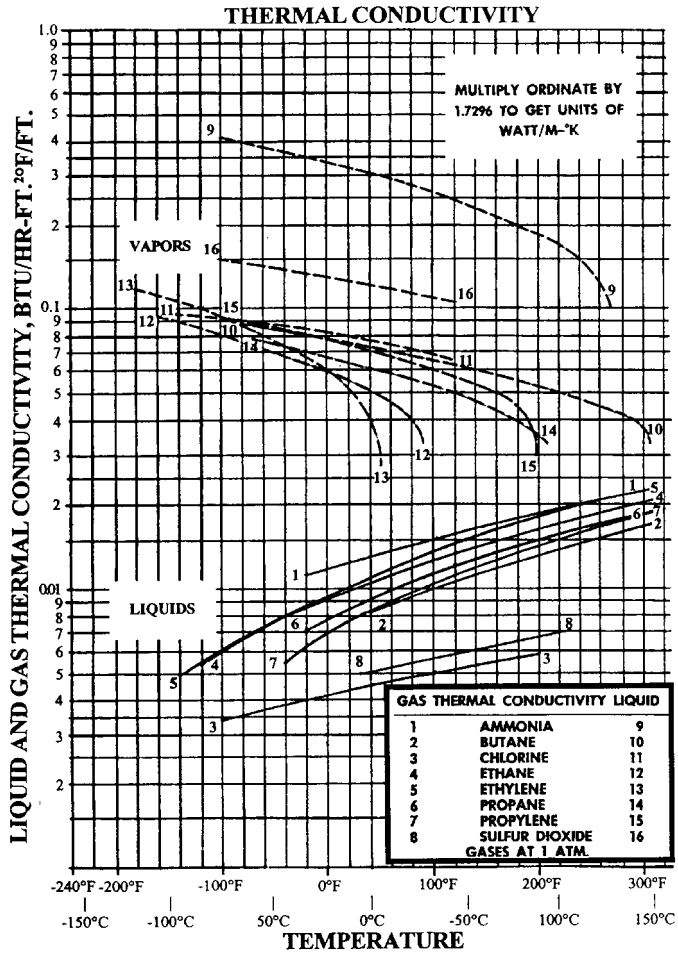


FIG. V-36 Conductivity—temperature for various fluids. (Source: Armstrong Engineering Associates.)

2. It usually takes several weeks longer to supply vaporizers with controls than it does to complete a bare vaporizer.

Liquid to vaporizer–vapor direct to load (vapor does not return to storage tank). Figure V-34 shows the most common dependable hookup for automatic operation. Liquid from the tank enters either by gravity, differential pressure, or may be fed by liquid pump when necessary to ensure adequate pressure to go through the reducing valves. This liquid enters the three lines and may go alternatively through the valve under normal operation, or through the hand bypass if the pilot valve is out. The pilot valve is controlled by a float on the shell that limits the level to which the liquid can rise above the tubesheet face.

There may or may not be a pressure-reducing valve between the storage tank and the controls shown in the figure, and depending on the duty, there may be a further vapor pressure-reducing valve after the outlet, before the load.

In most cases, the steam is fed by hand feed only, and is left on all the time. A trap may be used to drain the regular condensate outlet, and another one to also drain the condensate that forms in the steam chamber.

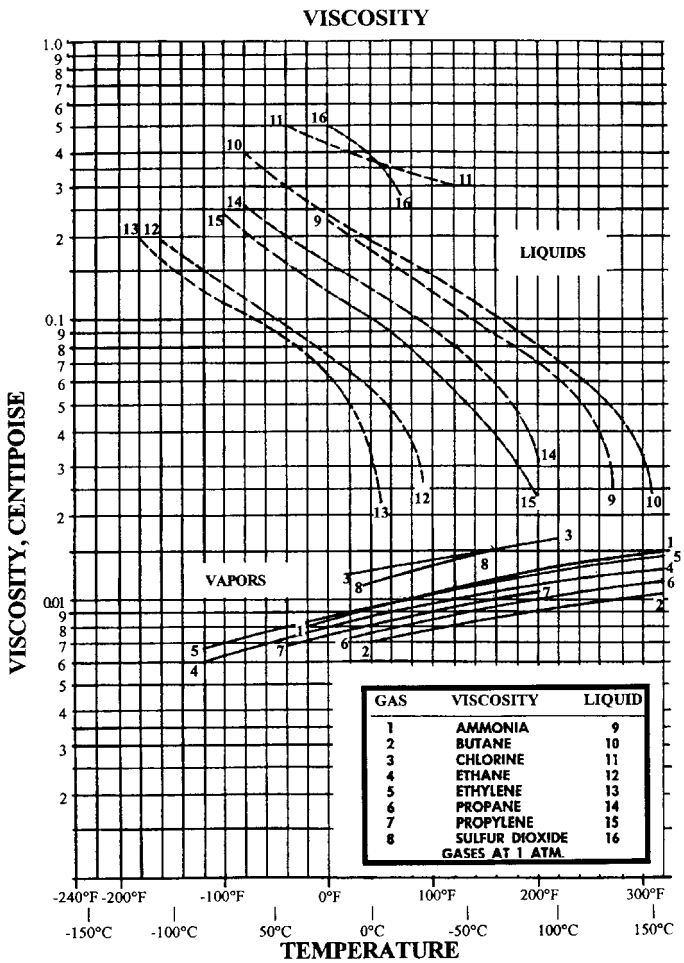


FIG. V-37 Viscosity temperature for various fluids. (Source: Armstrong Engineering Associates.)

If the unit is operating normally and the vapor outlet line is suddenly closed due to load variations, the steam being on, the pressure would normally build up rapidly with the liquid being confined in the vaporizer. This might cause the relief valve to blow and lose fluid. To avoid this, the center bypass check is installed—called a "blowback." When the vaporizer pressure exceeds the feed pressure, the flow will reverse, allowing the liquid to leave the vaporizer and return to the tank. Obviously, this blowback line must take into consideration the particular design of pump and pressure relief valves, if either are present, and must be arranged to bypass them if they will not permit reverse flow.

Constant vapor outlet temperature. The hookup illustrated in Fig. V-35 shows an alternate arrangement that is sometimes used to give a fairly constant vapor outlet temperature. It may be combined with a constant pressure arrangement through use of pumps and reducing valves if the pressure must also be controlled to obtain constant gas density for the requirements of mixing valves.

This sketch also shows an alternate arrangement sometimes used to get a constant gas density for an air-gas mixing valve. The thermostatic valve will maintain a constant gas-off temperature and a constant propane pressure in the vaporizer.

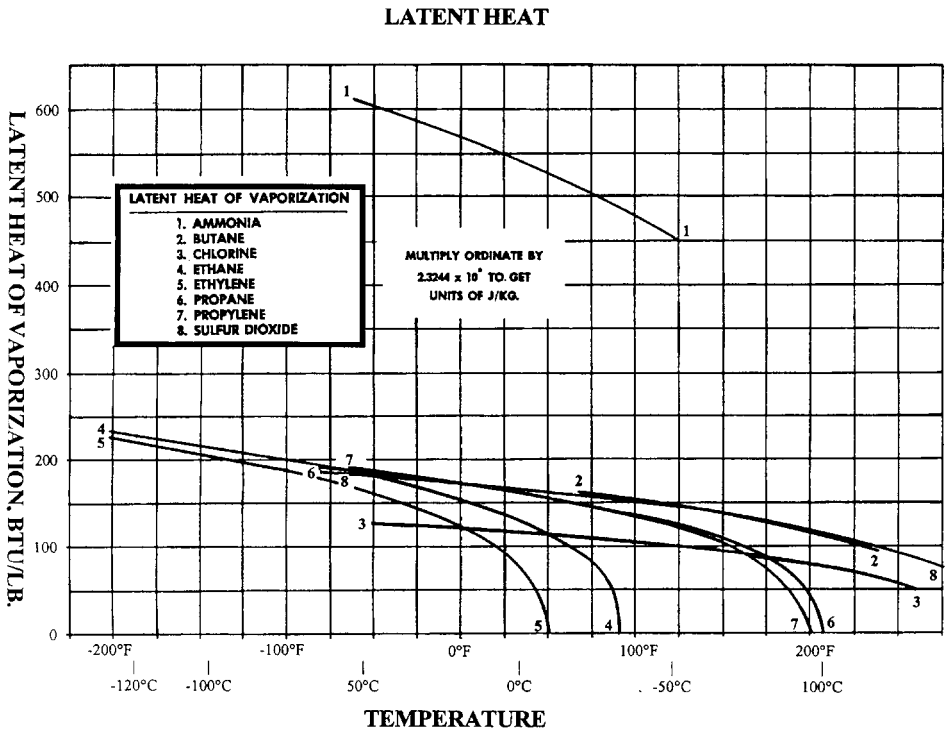


FIG. V-38 Latent heat—temperature for various fluids. (Source: Armstrong Engineering Associates.)

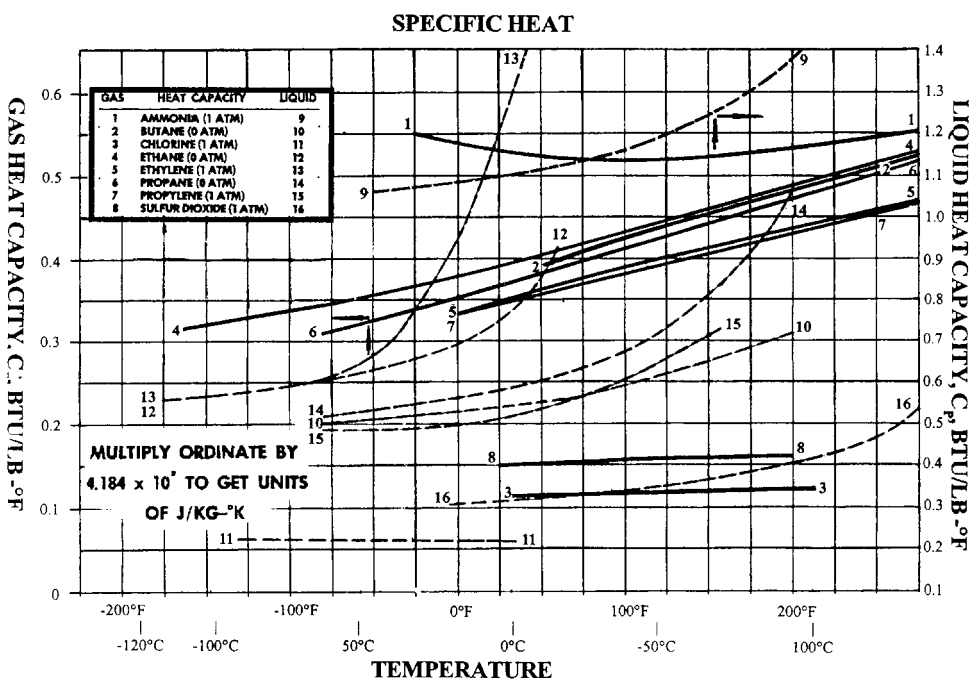


FIG. V-39 Heat capacity—specific heat—temperature for various fluids. (Source: Armstrong Engineering Associates.)

Vaporizers will furnish superheat if operated at lower ratings than those shown. The amount of superheat available by operating at reduced liquid level and using the upper part of the bundle for superheating, however, is limited. If a large amount of superheat is required, then a separate superheater may be advisable from a cost standpoint, as it often takes a great deal more surface to superheat the gas than to vaporize the liquid.

Vents and Flame Arrestors

Vents are provided on a storage tank so that the tank can "breathe." "Inhale" happens if a temperature drop causes a liquid volume reduction. It also can occur if liquid is withdrawn from the tank. "Exhale" happens in the reverse circumstances. Required thermal venting capacity is determined from tank capacity and fluid flashpoint. A tank supplier will ask what fluid your tanks will hold and the steady temperature and pressure at which the fluid is to be held. Vent capacity can then be determined. Conservation vent valves have just a slightly different design to minimize evaporative losses.

Flame arrestors are provided on tanks storing fluids with a flashpoint of less than 110°F. Flame arrestors should also be provided on storage tanks for liquids with a flashpoint above 110°F, where a combustible environment exists or other potentially hazardous condition, such as the heating of heavy oils.

Vibration Measurement (*see* Condition Monitoring)

W

Waste Heat Recovery (*see* Cogeneration)

Waste Management* (*see also* Pollutants, Chemical)

Waste management is a specialized science in itself and also has its own series of handbooks. However, if waste contaminates a site, it is possible that a process engineer may be held liable for the pollution, depending on whether mishaps in his or her plant's process caused it. For that reason, a definition of contaminated site classifications and a user's guide to classification of contaminated sites is included here.

Site Classification Categories

Sites will not be ranked relative to one another. Sites will be classified on their individual characteristics and will be placed into classes (Class 1, 2, 3, or N, according to their priority for action, or Class I for sites that require further information before they can be classified). The classification groupings are as follows:

Class 1 (score 70 to 100): Action required

The available information indicates that action (e.g., further site characterization, risk management, remediation, etc.) is required to address existing concerns. Typically, Class 1 sites show a propensity to high concern for several factors, and measured or observed impacts have been documented.

Class 2 (score 50 to 69.9): Action likely required

The available information indicates that there is high potential for adverse off-site impacts, although the threat to human health and the environment is generally not imminent. There is probably no indication of off-site contamination; however, the potential for this was rated high and therefore some action is likely required.

Class 3 (score 37 to 49.9): Action may be required

The available information indicates that this site is currently not a high concern. However, additional investigation may be carried out to confirm the site classification, and some degree of action may be required.

Class N (score <37): Action not likely required

The available information indicates there is probably no significant environmental impact or human health threats. There is likely no need for action unless new information becomes available indicating greater concerns, in which case the site should be reexamined.

---

* Source: Environment Canada.

TABLE W-1 User's Guide

Category	Evaluation Factor	Scoring Guideline
I. Contaminant(s) characteristics	A. Degree of hazard	
	■ High concern contaminants—high concentration	14
	■ High concern contaminants—low concentration	11
	■ Medium concern contaminants—high concentration	8
	■ Medium concern contaminants—low concentration	5
	■ Low concern contaminants	3
	B. Contaminant quantity (area/volume of site contamination)	
	■ >10 ha or >1000 m^3 or drums of liquid	10
	■ 2 to 10 ha, or 100 to 1000 m^3	6
	■ <2 ha or <100 m^3	2
	C. Physical state of contaminants	
	■ Liquid/gas	9
	■ Sludge	7
	■ Solid	3
	Special considerations	–6 to +6
II. Exposure pathways	A. Groundwater	
	1. Known contamination at or beyond property boundary	
	■ Groundwater significantly exceeds Canadian Drinking Water Guidelines (CDWG) by >2× or known contact of contaminants with groundwater	11
	■ Between 1 and 2× CDWG or probable contact with groundwater	6
	■ Meets Canadian Drinking Water Guidelines	0

Rationale	Method of Evaluation	Sources of Information
In determining the degree of hazard of a waste, it is recognized that a listed hazardous waste is generally of greater concern than a liquid or solid industrial waste. These are in turn of greater concern than other solid wastes. Municipal and organic wastes are considered medium concern contaminants due to their putrescible nature (production of methane and other landfill gases). Household wastes may contain hazardous materials (e.g., batteries, medical wastes, paints, etc).	Determine the level of hazard according to the following table of typical contaminants and definition of high concentrations: *High concern contaminants* ■ Materials defined as dangerous goods in the Transport of Dangerous Goods Act and Regulations ■ Materials identified by Province as hazardous waste (pesticides, herbicides, paint sludge, acid and alkaline solutions, solvents, etc.) ■ Materials regulated by the Canadian Environmental Protection Act (e.g., PCBs) ■ Institutional waste (lab, schools hospitals, etc.) ■ Pathological wastes and animal carcasses ■ Radioactive wastes *Medium concern contaminants* ■ Liquid waste not referred to in above, petroleum products septic tank pumpings, agricultural and chemical containers ■ Food processing wastes ■ Nonhazardous incinerator residues ■ Municipal solid (household) wastes ■ Organic and vegetable wastes ■ Mining residues *Low concern contaminants* ■ Industrial and commercial solid wastes, (e.g., construction materials such as wood, metal, hay, sand/silt piles, etc.) ■ Other nearly inert wastes (e.g., foundry sands) *High concentration of contaminants* ■ Contaminant concentrations in soil, groundwater, or surface water exceed Canadian Environmental Quality Criteria for Contaminated Sites (>2× commercial/ industrial level); or material that was deposited in highly concentrated form (e.g., >5000 ppm)	Transport of Dangerous Goods Act; Provincial/ Territorial Hazardous Wastes lists; regulations under Canadian Environmental Protection Act; Canadian Environmental Quality Criteria for Contaminated Sites; etc.
Little information is known about the quantity of wastes at abandoned sites in Canada. Therefore, waste interpreted from area or quantity information.	Measure or estimate the area or quantity of potential contamination. *Note:* Any number of drums abandoned or disposed is considered a high concern.	
Contaminants in liquid form are more mobile in the ground and water than solids. However, certain water-soluble solid wastes are more mobile than viscous liquids, and these should be evaluated on a case-by-case basis.	Determine the state of the contaminant when it was disposed or deposited. Technical judgment.	
The legislative basis for most jurisdictions is to prevent off-site migration of contamination.	Review chemical data and evaluate groundwater quality. If contamination at or beyond the property boundary exceeds Canadian Drinking Water Guidelines (CDWG) or applicable provincial/territorial guidelines or policies, or if contaminants are known to be in contact with groundwater, then evaluate the site as high.	Canadian Water Quality Guidelines; Provincial/ Territorial Water Quality Guidelines or policies: Guidelines for Canadian Drinking Water Quality.

TABLE W-1 User's Guide (*Continued*)

Category	Evaluation Factor	Scoring Guideline
	2. Potential for groundwater contamination	
	(a) Engineered subsurface containment	
	■ No containment	4
	■ Partial containment	2
	■ Full containment	0
	(b) Thickness of confining layer over aquifer(s) of concern	
	■ 3 m or less	1.5
	■ 3 to 10 m	1
	■ >10 m	0
	(c) Hydraulic conductivity of the confining layer	
	■ $>10^{-4}$ cm/s	1.5
	■ 10^{-4} to 10^{-6} cm/s	1
	■ $<10^{-6}$ cm/s	0.5
	(d) Annual rainfall	
	■ >1000 mm	1
	■ 600 mm	0.6
	■ 400 mm	0.4
	■ 200 mm	0.2
	(e) Hydraulic conductivity of aquifer(s) of concern	
	■ $>10^{-2}$ cm/s	3
	■ 10^{-2}–10^{-4} cm/s	1.5
	■ $<10^{-4}$ cm/s	0.5
	3. Special considerations	–4 to +4
B. Surface water		
	1. Observed or measured contamination of water/effluent discharged from site	
	■ Known or strongly suspected to exceed Canadian Water Quality Guidelines (CWQE) by >2×	11
	■ Known or strongly suspected to be between 1 – 2 × CWQE	6
	■ Meets Canadian Water Quality Guidelines	0
	2. Potential for surface water contamination	
	(a) Surface containment	
	■ No containment	5
	■ Partial containment	3
	■ Full containment	0.5
	(b) Distance to perennial surface water	
	■ 0 to <100 m	3
	■ 100 to 300 m	2
	■ >300 m	0.5

Rationale	Method of Evaluation	Sources of Information
Well contained sites have minimal potential for pollution. Potential for pollution decreases with increasing containment.	Review the existing engineered systems and relate these structures to hydrogeology of the site and determine if full containment is achieved. Full containment is defined as an engineered system, monitored as being effective, which provides for the capture and treatment of contaminants. If there is no system, this factor is evaluated high. If there is less than full containment or if uncertain then evaluate as medium. Typical engineered systems include leachate collection systems and low permeability liners.	
The thickness of a confining layer (e.g., clay, shale, etc.) between contaminants and any aquifers of concern will affect the attenuation of contaminants and hence the quantity and quality of contaminants reaching the aquifers.	Measure or estimate thickness of any confining layer (e.g., clay, shale, etc.) over all aquifers of concern from existing well records or from a general knowledge of local conditions. If possible, an estimate of the continuity of the confining layer should be made from borehole well record information. Note: an aquifer is defined as a geologic material that will yield groundwater in usable quantities.	Historical geologic maps, well records, government hydrogeologist or local consultants.
The rate at which contaminants migrate through the confining layer will affect attenuation and the contaminant loading to the aquifers.	Determine the nature of geologic materials and estimate hydraulic conductivity from published material. Clays, granite, shales should be scored low. Silts etc. should be scored medium. Sand, gravel, and limestone should be scored high.	Freeze and Cherry, 1979, and other groundwater texts.
The quantity of rainfall affects the quantity of leachate produced. Higher leachate quantities have a higher impact on the environment	Refer to Environment Canada rainfall records for relevant areas. Use 30-year average rainfall for evaluation purposes. Divide rainfall by 1000 and round to nearest tenth (e.g., 667 mm = 0.7 score).	Hydrological Atlas of Canada (Fisheries and Environment Canada, 1978).
Aquifers with high hydraulic conductivity can transport contaminants at high velocity over great distances, e.g., solution limestones, highly fractured rocks or gravel deposits.	Determine the nature of geologic materials and estimate hydraulic conductivity of all aquifers of concern from published material.` Technical judgment.	Freeze and Cherry, 1979.
The legislative basis in all jurisdictions is not to contaminate surface water beyond established limits.	Collect all available information on quality of surface water near to site. Evaluate available data against Canadian Water Quality Guidelines (select appropriate guidelines based on local water use, e.g., recreational, irrigation, freshwater aquatic life, etc.) and relevant provincial/ territorial water quality objectives.	CCME Canadian Water Quality Guidelines; Relevant provincial/ territorial and federal legislation and regulations.
The level and type of engineered containment will affect the potential for contaminants to be released to surface water.	Review the existing engineered systems and relate these structures to site conditions and proximity to surface water and determine if full containment is achieved; e.g., evaluate low if there is full containment such as capping, berms, dikes; evaluate medium if there is partial containment such as natural barriers, trees, ditches, sedimentation ponds; evaluate high if there are no intervening barriers between the site and nearby surface water.	Site inspection reports, air photos, etc.
The distance to surface water will affect the probability of contaminants reaching the watercourse. The Ontario Ministry of the Environment has established a classification for immediate impact zone at 50 m. For conservatism, this zone has been broadened to 100 m.	Review available mapping and survey data to determine distance to nearest surface water bodies.	

TABLE W-1 User's Guide (*Continued*)

Category	Evaluation Factor	Scoring Guideline
	(c) Topography	
	■ Contaminants above ground level and slope is steep	1.5
	■ Contaminants at or below ground level and slope is steep	1.2
	■ Contaminants above ground level and slope is flat	0.8
	■ Contaminants at or below ground level and slope is flat	0
	(d) Run-off potential	
	■ >1000 mm rainfall and low permeability surface material	1
	■ 500 to 1000 mm rainfall and moderately permeable surface material	0.6
	■ <500 mm rainfall and highly permeable surface material	0.2
	(e) Flood potential	
	■ 1 in 2 years	0.5
	■ 1 in 10 years	0.3
	■ 1 in 50 years	0.1
	3. Special considerations	–4 to +4
	C. Direct contact	
	1. Known contamination of media off-site	
	■ Known contamination of soil, sediment, or air off-site due to contact with contaminated soil, dust, air, etc. (vector transported should also be considered).	11
	■ Strongly suspected contamination of media off-site	6
	■ No contamination of media off-site	0
	2. Potential for direct human and/or animal contact	
	(a) Airborne emissions (gases, vapors, ducts, etc.)	
	■ Known or suspected airborne emissions impacting on neighboring properties	5
	■ Airborne emissions generally restricted to site	3
	■ No airborne emissions	0
	(b) Accessibility of site (ability to contact materials)	
	■ Limited or no barriers to prevent site access; contaminants not covered	4
	■ Moderate accessibility or intervening barriers; contaminants are covered	3
	■ Controlled access or remote location and contaminants are covered	0
	(c) Hazardous soil gas migration	
	■ Contaminants are putrescible and soil permeability is high	2
	■ Soil contaminants are putrescible but soil permeability is low and/or groundwater is <2 m from surface	1
	■ No putrescible contaminants at the site.	0
	3. Special considerations	–4 to +4
III. Receptors	A. Human and animal uses	
	1. Known adverse impact on humans or domestic animals as a result of the contaminated site	
	■ Known adverse effect on humans or domestic animals	18
	■ Strongly suspected adverse effect on humans or domestic animals	15

Rationale	Method of Evaluation	Sources of Information
Water can run off (and therefore potentially contaminate surface water) with greater ease from elevated sites on slopes.	Review engineering documents on the topography of the site and the slope of surrounding terrain. ■ Steep slope = >50% ■ Flat slope = <5% *Note:* Type of fill placement (e.g., trench, above ground, etc.)	
Run-off transports contaminants into water bodies. Water run-off is a function of precipitation and the rate of infiltration (less permeable soils will allow greater turn-off).	Refer to Environment Canada precipitation records for relevant areas. Use 30-year average precipitation for evaluation purposes.	Hydrological Atlas of Canada (Fisheries and Environment Canada, 1978).
The potential for large quantities and concentrations of contaminants to be released to surface water courses over a short period of time will be affected by the flood potential of a water course near the site.	Review published data such as flood plain mapping or flood potential (e.g., spring or mountain run-off) and Conservation Authority records to evaluate flood potential of nearby water course both up and down gradient. Rate zero if site not in flood plain. Technical judgment.	Established flood plain guidelines/ maps; provincial/ territorial soil survey maps.
Known or measured contamination off-site is an important consideration for determining impact of contaminants.	Record known or measured contamination of soil, sediment, or air on or off-site. Note any presence of soil gas, such as methane, associated with site.	
If air emissions are evident off-site, there is a great hazard for direct contamination of neighboring biota and/or resources.	Review available site information to determine if there have been complaints off-site (due to vapors, gas, dust, etc.). Reports for these problems are not likely available for most abandoned sites. Review regulatory site inspection reports. If airborne emissions are known to be impacting neighboring properties and possibly endangering the public, some immediate action (including characterization of emissions) should be initiated to curtail hazardous emissions or otherwise reduce or eliminate exposure.	Site inspection reports, etc.
The greater the accessibility to a site and to contaminants, the greater the chance for contamination of human and animal life by direct contact.	Review location and engineering of the site and determine if there are intervening barriers between the site and humans or animals. A low rating should be assigned to a (covered) site surrounded by a locked chain link fence or in a remote location, whereas a high score should be assigned to a site that has no cover, fence, natural barriers, or buffer.	
Methane gas migration has been known to cause explosions adjacent to abandoned landfills.	Consider presence of organic material on site, the depth to water table, soil hydraulic conductivity, vegetative stress, odors, etc. Technical judgment.	
Contamination from a site that causes a measurable impact on humans is a great concern.	Review and evaluate reports of impact(s) of site contamination (e.g., increased heavy metal levels measured in blood of nearby residents as a result of site contamination). Any site assigned 15 or more points for this factor should automatically be classified as Class 1. An adverse effect is considered to be any one or more of the following: (i) impairment of the quality of the natural environment for any use that can be made of it, (ii) injury or damage to property or to plant or animal life, (iii) harm or material discomfort to any person, (iv) impairment of the safety of any person, (v) rendering any property or plant or animal life unfit for use by humans, (vi) loss of enjoyment of normal use of property, and (vii) interference with the normal conduct of business (from Ontario Environmental Protection Act, 1980).	

TABLE W-1 User's Guide (*Continued*)

Category	Evaluation Factor	Scoring Guideline
	2. Potential for impact on humans or animals	
	(a) Drinking water supply	
	(i) Known impact on drinking water supply; drinking water supply is known to be adversely affected as a result of site contamination	
	■ Known contamination of drinking water supply to levels above CDWG	9
	■ Strongly suspected contamination of drinking water supply	7
	■ Drinking water supply is known not to be contaminated	0
	(ii) Potential for impact on drinking water supply; proximity to drinking water supply	
	■ 0 to <100 m	6
	■ 100 to <300 m	5
	■ 300 m to <1 km	4
	■ 1 to 5 km	3
	"Availability" of alternate drinking water supply	
	■ Alternate drinking water supply is not available	3
	■ Alternate drinking water supply would be difficult to obtain	2
	■ Alternate drinking water supply available	0.5
	b) Other water resources	
	(i) Known impact on used water resource; water resource (used for recreational purposes, commercial food preparation, livestock watering, irrigation, or other food chain uses) is known to be adversely affected as a result of site contamination.	
	■ Water resource is known to be contaminated above CWQG	4
	■ Water resource is strongly suspected to be contaminated above CWQG	3
	■ Water resource is known not to be contaminated	0
	(ii) Potential for impact on water resources; proximity to water resources used for activities listed above	
	■ 0 to <100 m	2
	■ 100 to <300 m	1.5
	■ 300 m to <1 km	1
	■ 1 to 5 km	0.5
	Use of water resources—if multiple uses, give highest score (use following table)	0.2–2

Water Use	Frequency of Use	
	Frequent	Occasional
Recreational (swimming, fishing)	2	1
Commercial food preparation	1.5	0.8
Livestock watering	1	0.5
Irrigation	1	0.5
Other domestic or food chain uses	0.5	0.3
Not currently used but likely future use	0.5	0.2

	(c) Direct human exposure	
	(i) Known contamination of land used by humans	
	■ Known contamination of land used for agricultural or residential/parkland/school purposes above AG or R/P EQC values	5

Rationale	Method of Evaluation	Sources of Information
Water used for drinking should be protected against contamination from any site.	Review available site data (inspection reports, assessment documentation) to determine if drinking water (groundwater, surface water, private, commercial, or municipal supply) is known or suspected to be contaminated above Guidelines for Canadian Drinking Water Quality or applicable provincial/territorial guidelines or policies. If drinking water supply is known to be contaminated above these guidelines, some immediate action (e.g., provision of alternate drinking water supply) should be initiated to reduce or eliminate exposure.	Guidelines for Canadian Drinking Water Quality; other drinking water guidelines developed by recognized agencies (e.g., other Health and Welfare Canada guidelines, U.S. EPA, etc.).
The nearer a drinking water well is to a contaminant source, the greater the potential for contamination. Well water used for irrigation/agricultural purposes should also be included as it may be used for human consumption.	Review provincial/territorial base mapping or air photos and measure the distance to the nearest resident or drinking water supply. Judge whether the water is being used as a drinking water source. Commonly rural areas use groundwater for drinking purposes. For urban sites, contact the local Public Utilities Commission to determine water source and location.	
This factor takes into account the availability of replacement water supplies, and is used in the technical sense as a factor to indicate the degree of urgency, not as a sociopolitical consideration.	Determine availability of alternate drinking water supply or distance to alternate source.	
The water used for these purposes (groundwater or surface water) should be protected against contamination.	Review documentation for reported or suspected contamination of water used for recreation or food chain uses, and refer to Canadian Water Quality Guidelines or other relevant guidelines (select appropriate guidelines based on local water used) to determine if supply is considered contaminated.	CCME Canadian Water Quality Guidelines; provincial/ territorial water quality guidelines and objectives; etc.
The nearer a water resource is to a site, the greater the risk of contamination.	Determine distance from the site to the nearest recreational or food chain used water resource.	
Potential for impact due to use of water resource is related to the type and frequency of use. Human uses are of the highest cencern.	Assess water users adjacent to the site from maps and directories.	
Hazards associated with soil contamination are directly related to land use.	Review zoning and land use maps for lands adjacent the site. Evaluate levels of soil contamination against Canadian Environmental Quality Criteria (EQC) for Contaminated Site (AG = agricultural level; R/P = residential/parkland	CCME Canadian Environmental Quality Criteria for Contaminated Sites.

TABLE W-1 User's Guide (*Continued*)

Category	Evaluation Factor/Guideline	Scoring Rationale
	■ Known contamination of land used for commercial or industrial purposes above C/I EQC values	3.5
	■ Land is known not to be contaminated	0
	(ii) Potential human exposure through land use	
	■ Use of land at and surrounding site (use following table; give highest score to worst case scenario)	0.5–5

	Distance from Site		
Lane use (current or future)	0–300 m	300 m–1 km	1–5 km
Residential	5	4.5	3
Agricultural	5	4	2.5
Parkland/school	4	3	1.5
Commercial/industrial	3	1	0.5

3. Special considerations	–5 to +5
B. Environment	
1. Known adverse impact on a sensitive environment as a result of the contaminated site	
■ Known adverse impact on sensitive environment	16
■ Evidence of stress on aquatic species or vegetative stress on trees, crops, or plant life located on properties neighboring the site	14
■ Strongly suspected adverse impact on sensitive environment	12
2. Potential for impact on sensitive environments	
(a) Distance from site to nearest sensitive environment (e.g., sensitive aquatic environment, nature preserve, habitat for endangered species, sensitive forest reserves, national parks or forests, etc.)	
■ 0 to <500 m	10
■ 500 m to <2 km	6
■ 2 to <5 km	2
■ 5 to 10 km	0.5
(b) Groundwater—distance to important or susceptible groundwater resource(s)	
■ 0 to <500 m	6
■ 500 m to <2 km	4
■ 2 to <5 km	2
■ 5 to 10 km	1
3. Special considerations	–5 to +5

Method of Evaluation	Information	Sources of
	level; C/I = commercial/industrial level). If soil is known to be contaminated above these levels and possibly endangering public health, some immediate action (e.g., fencing the area, limiting public access, etc.) should be initiated to reduce or eliminate the exposure.	
Hazards associated with soil contamination are directly related to land use and distance of the used land from the site. Residential and agricultural land uses are of highest concern because humans are situated at these locations for longer periods.	Review zoning and land use maps over the distances indicated. If the proposed future land use is more "sensitive" than the current land use, evaluate this factor assuming the proposed future use is in place (indicate in the worksheet that future land use is the consideration). Agricultural land use is defined as uses of land where the activities are related to the productive capability of the land or facility (e.g., greenhouse) and are agricultural in nature, or activities related to the feeding and housing of animals as livestock. Residential/Parkland land uses are defined as uses of land on which dwelling on a permanent, temporary, or seasonal basis is the activity (residential), as well as uses on which the activities are recreational in nature and require the natural or human designed capability of the land to sustain that activity (parkland). Commercial/Industrial land uses are defined as land on which the activities are related to the buying, selling, or trading of merchandise or services (commercial), as well as land uses that are related to the production, manufacture, or storage of materials (industrial). Technical judgment.	
The environment should be to protected against site contamination. Evidence of impact(s) shows that protection is lacking.	Review records for evidence of vegetative stress or impairment of any nearby sensitive environments. A sensitive environment is defined as a sensitive aquatic environment, nature preserve, habitat for endangered species, sensitive forest reserves, national parks or forests, etc. An adverse effect is considered to be any one or more of the following: (i) impairment of the quality of the natural environment for any use that can be made of it, (ii) injury or damage to property or to plant or animal life, (iii) harm or material discomfort to any person, (iv) impairment of the safety of any person, (v) rendering any property or plant or animal life unfit for use by humans, (vi) loss of enjoyment of normal use of property, and (vii) interference with the normal conduct of business (from Ontario Environmental Protection Act, 1980).	
It is considered that within approximately 1 km of the site there is immediate concern for contamination. Therefore, an environmentally sensitive area located within this area of the site will be subject to concern. It is also generally considered that any sensitive area located greater than 10 km from the site will not be impacted.	Review Conservation Authority mapping and literature. Also review Ministry of Natural Resources records and Federal Land Capability maps. Identify provincial/territorial and federal designated environmentally sensitive areas.	Relevant provincial/ territorial and federal maps of sensitive environments.
The closer a site is to a discharge or recharge area, the greater the potential for contamination of a groundwater or surface water resource.	Review groundwater contour maps, if available, and other available reports. Otherwise use established hydrogeologic principles. Technical judgment.	Local groundwater maps, etc.

Class I (estimated score ≥15): Insufficient information

There is insufficient information to classify the site. In this event, additional information is required to address data gaps.

Within each priority category, further refinement of the relative classification of sites may be necessary. The National Classification System is a screening tool only. Firm conclusions about the need for remedial action will still depend on a number of factors (including planned long-term use or redevelopment of the site, application of contaminated site criteria, and relevant/site-specific objectives of the jurisdiction in which the site is located, local issues, availability of technology, remediation costs, etc.). These factors are beyond the scope of this section, however, and are not meant to be addressed.

See Table W-1.

Reference and Additional Reading

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.

Welding (*see* Metallurgy)

Some Commonly Used Specifications, Codes, Standards, and Texts

Texts

No handbook is all encompassing, and the process engineer's field is a vast one. The following texts may help process engineers, depending on their industry sector and duties, especially if used in conjunction with this one.

Agriculture

1. Hsu, D. S., *Ultrahigh Temperature Processing and Aseptic Packaging of Dairy Products; Dairy and Food Technology Handbook*, Damana Tech, 1970.
2. *North American Food Processing and Trade Dictionary*, expanded NAFTA edition, Allied Publishers International, 1997.
3. *Handbook of the Food Processing, Distribution and Retailing Sectors*, Rizvi SMH, Marketing and Economics Branch, Agriculture Canada.
4. *North American Food Processing Dictionary*, Allied Publishing International, 1996.

Biotechnology (*see* Environment)

Chemical Engineering; Chemicals

1. *Handbook of Chemical Engineering Calculations*, McGraw-Hill, New York, 1984.
2. Wells, G. M., *Handbook of Petrochemicals and Processes*, Gower, New York, 1999.
3. Perry, R. H., ed., *Perry's Chemical Engineering Handbook*, 7th ed., McGraw-Hill, New York, 1997.

Civil Engineering

1. Scott, J. S., *Dictionary of Civil Engineering*, 3d ed., Halstead Press, 1981.
2. *Standard Handbook for Civil Engineers*, 4th ed., McGraw-Hill, 1995.
3. ASCE Manuals and Reports on Engineering Practice, ASCE, 1962.

Cleaning (*see* Agriculture; Chemical Engineering; *appropriate field*)

Control Systems; Instrumentation

1. *Process/Industrial Instrumentation and Controls Handbook*, 4th ed., McGraw-Hill, New York, 1993.
2. Lenk, J. D., *Handbook of Controls and Instrumentation*, Prentice Hall, Upper Saddle River, N.J., 1980.
3. *Handbook of Instrumentation and Controls: A Practical Design and Applications Manual for the Mechanical Services Covering Steam Plants, Power Plants, Heating Systems, Air Conditioning Systems, Ventilation Systems, Diesel Plants, Refrigeration and Water Treatment*, McGraw-Hill, New York, 1996.

Conversion of Units (*see* Metric System Conversions)

Environment

1. Soares, C. M., *Environmental Technology and Economics: Sustainable Development in Industry*, Butterworth-Heinemann, 1999.
2. Saylor and Sanseverino, eds., *Biotechnology in the Sustainable Environment*, Plenum Press, New York, 1997.

3. Ronneau and Bitchaeva, eds., *Biotechnology for Waste Management and Site Restoration: Technology, Education, Business and Political Aspects*, Kluwer Academic, 1997.
4. Sikdar and Irvine, *Bioremediation: Principles and Practice*, Techromic Publishing Co., 1998.
5. Eweis, *Bioremediation Principles*, McGraw-Hill, New York.
6. *Standard Handbook of Hazardous Waste Treatment and Disposal*, 2d ed., McGraw-Hill, New York, 1998.
7. *Hazardous Waste Materials Compliance Handbook*, 2d ed., Van Nostrand Reinhold, New York, 1997.
8. *Handbook of Solid Waste Management*, McGraw-Hill, New York, 1996.

Food Processing (*see* Agriculture)

Health (*see* Safety)

Instrumentation; Measurement (*see* Control Systems)

Machinery (*see* Rotating Machinery)

Machining and Tooling (*see* Metallurgy)

Manufacturers

1. *Thomas Register of American Manufacturers*, Thomas Publishing Company, New York.

Medical (*see* Pharmaceuticals)

Metallurgy; Metals; Machining; Tooling, Mining

1. *American Society of Tool and Manufacturing Engineering, Handbook of Fixture Design*, ASTM, Philadelphia, Pa., 1962.
2. *Thesaurus of Mineral Processing and Extractive Metallurgy Terms*, 2d ed., Energy Mines and Resources Canada, 1985.
3. *International Dictionary of Materials, Mineralogy, Geology, Mining and Oil Industries*, Technoprint International, Milano, 1968.
4. *Concise Metals Engineering Data Book*, ASM International, 1997.
5. Walsh, R. A., *Machining and Metalworking Handbook*, 2d ed., McGraw-Hill, New York, 1998.
6. Brown, J., *Advanced Machining Technology Handbook*, McGraw-Hill, New York, 1998.
7. Madison, J., *CNC Machining Handbook: Basic Theory, Production Data and Machining Procedures*, 1st ed., Industrial Press, 1996.
8. *Smithells Metals Reference Book*, 7th ed., Butterworth-Heinemann, 1998.
9. *Welders Handbook: A Complete Guide*, HP Books, 1997.
10. *Welding Handbook*, American Welding Society, 1968.

Metals (*see* Metallurgy)

Metric System Conversions

1. *McGraw-Hill Metrication Manual*, McGraw-Hill, New York, 1972.

Mining (*see* Metallurgy)

Pharmaceuticals

1. *CPS: Compendium of Pharmaceuticals and Specialties*, Canadian Pharmaceutical Association, 1967.
2. *Chemical Stability of Pharmaceuticals: A Handbook for Pharmaceuticals*, John Wiley, New York, 1979.

Plastics

1. *Handbook of Plastic Materials and Technology*, John Wiley, New York, 1990.
2. *Plastics Handbook*, McGraw-Hill, New York, 1994.

Process Plant Machinery (*see* Rotating Machinery)

Pulp and Paper

1. Britt, K. W., ed., *Handbook of Pulp and Paper Technology*, Van Nostrand Reinhold, New York, 1970.
2. *Process Engineering Design Criteria Handbook*, Technical Association of Pulp and Paper Industries, TAPPI Press, 1996.

Rotating Machinery

Bloch, H., and Soares, C. M., *Process Plant Machinery*, 2d ed., Butterworth-Heinemann, 1998.

Safety

1. *Handbook of Environmental Health and Safety Principles and Practices*, National Environmental Health Association, 1991.
2. Woodside, G., *Environmental Health and Safety Portable Handbook*, McGraw-Hill, New York, 1998.

Waste Management (*see* Environment)

Welding (*see* Metallurgy)

A List of American Petroleum Institute (API) Specifications and Publications

From the Library of Congress Index, 138 titles (subjects include, for instance, petroleum engineering—specifications; and rotating machinery, such as compressors, pumps, and so forth).

A List of Available ASME Codes, Standards, and Specifications

Consult ASME (800-THE ASME) at the time of purchase for any changes to list prices and/or additional specifications available.

Key to Updating Service Symbols and Reaffirmation (R) Designation

Codes and standards with updating service are indicated by the following symbols:

A = Addenda: Additions and revisions that are sent automatically to purchasers of code or standard.

I = Interpretations: Written replies to inquiries concerning interpretation of technical aspects of code or standard. Sent automatically to purchasers of code or standard.

C = Cases: Codes and standards committees meet regularly to consider proposed additions and revisions to code or standard and to formulate Cases to clarify the intent of existing requirements or provide, when the need is urgent, rules for materials or constructions not covered by existing rules. Sent automatically to purchasers of code or standard.

S = Supplement: Same as "A," Addenda.

Year (e.g., A I 2000): Next publication date of code or standard. Updating service includes Addenda, Cases, Interpretations, or Supplements up to this date.

Reaffirmation (e.g., R1997): Committee has determined that no changes are warranted in code or standard.

L = List: List of certificate holders.

AUTHORIZED INSPECTION

QAI-1-1995
Qualifications for Authorized Inspection
A I L 1999
Order No. J09595
$38.00

AUTOMOTIVE LIFTING DEVICES
Safety standard for portable automotive lifting devices, covering requirements for safety, health, design, production, construction, maintenance, performance or operation of equipment, or qualification of personnel.

PALD-1997
Safety Standard for Portable Automotive Lifting Devices
A I 2002
Order No. J13597
$89.00

BOILERS

Boiler and Pressure Vessel Code

CHAINS
Standards for the dimensions of transmission and conveyor chains.

B29.1M-1993
Precision Power Transmission Roller Chains, Attachments and Sprockets
Order No. K12793
$35.00

B29.2M-1982(R1994)
Inverted Tooth (Silent) Chains and Sprockets
Order No. K00111
$29.00

B29.3M-1994
Double-Pitch Power Transmission Roller Chains, and Sprockets
Order No. K07594
$30.00

B29.4M-1994
Double-Pitch Conveyor Roller Chains, Attachments and Sprockets
Order No. M07294
$32.00

B29.6M-1993
Steel Detachable Link Chains, Attachments, and Sprockets
Order No. M06693
$39.00

B29.8M-1993
Leaf Chains, Clevises and Sheaves
Order No. M00393
$29.00

B29.10M-1997
Heavy Duty Offset Sidebar Power Transmission Roller Chains and Sprocket Teeth
Order No. M07897
$33.00

B29.11M-1994
Combination Chains, Attachments, and Sprocket Teeth
Order No. K08194
$29.00

B29.12M-1997
Steel Bushed Rollerless Chains, Attachments, and Sprocket Teeth
Order No. K03097
$32.00

B29.14M-1996
"H" Type Mill Chains, Attachments, and Sprocket Teeth
Order No. M06296
$29.00

B29.15M-1997
Steel Roller Type Conveyor Chains, Attachments, and Sprocket Teeth
Order No. M07997
$34.00

B29.16M-1995
Welded Steel Type Mill Chains, Attachments, and Sprocket Teeth
Order No. K07995
$29.00

B29.17M-1983(R1995)
Hinge Type Flat Top Conveyor Chains and Sprocket Teeth
Order No. K00078
$29.00

B29.18M-1994
Welded Steel-Type Drag Chains, Attachments, and Sprocket Teeth
Order No. K08093
$29.00

B29.19M-1996
Agricultural Roller Chains, Attachments, and Sprockets
Order No. K08596
$29.00

B29.21M-1996
700 Class Welded Steel and Cast Chains, Attachments, and Sprockets for Water and Sewage Treatment Plants
Order No. K10596
$29.00

B29.22M-1995
Drop Forged Rivetless Chains, Sprocket Teeth Drive Chain/Drive Dogs
Order No. K10495
$27.00

B29.23M-1985(R1995)
Flexible Chain Couplings
Order No. K00114
$29.00

B29.24M-1995
Roller Load Chains for Overhead Hoists
Order No. K12695
$29.00

B29.25M-1994
Open Barrel Steel Pintle Type Conveyor Chains, Attachments, and Sprockets
Order No. K12494
$28.00

B29.26M-1996
Fatigue Testing Power Transmission Roller Chain
Order No. M16096
$32.00

COMPRESSORS
Safety standards for the construction, installation, operation, and maintenance of air and gas compression equipment.

B19.1-1995
Safety Standard for Air Compressor Systems
A I 2000
Order No. J09795
$48.00

B19.3-1991
Safety Standard for Compressors for Process Industries
A I 1996
Order No. JX7091
$37.00

CONTROLS
Safety requirements for the assembly, maintenance, and operation of controls and safety devices installed on automatically operated boilers directly fired with gas, oil, gas/oil, or electricity.

CSD-1-1998
Controls and Safety Devices for Automatically Fired Boilers
A I 2001
Order No. A09198
$52.00

Form CSD-1
Manufacturer's/ Installing Contractor's Report for ASME CSD-1
Order No. A091F1
$18.00

CONVEYORS
Safety standard for the design, construction, installation, maintenance, inspection, and operation of conveyors and conveying systems in relation to accident hazards.

B20.1-1996
Safety Standard for Conveyors and Related Equipment
A I 1999
Order No. J08897
$58.00

CRANES AND HOISTS
Safety standards for the construction, installation, operation, inspection, and maintenance of jacks, cranes, monorails, derricks, hoists, hooks, slings, and cableways; and specification standards for hoists.

B30.1-1998
Jacks
A I 2003
Order No. J06998
$57.00

B30.2-1996
Overhead and Gantry Cranes (Top Running Bridge, Single or Multiple Girder, Top Running Trolley Hoist)
A I 2001
Order No. J08996
$52.00

B30.3-1996
Construction Tower Cranes
A I 2001
Order No. J03396
$52.00

B30.4-1996
Portal, Tower, and Pedestal Cranes
A I 2001
Order No. J00196
$45.00

B30.5-1994
Mobile and Locomotive Cranes
A I 1999
Order No. J08394
$58.00

B30.6-1995
Derricks
A I 2000
Order No. J09395
$45.00

B30.7-1994
Base Mounted Drum Hoists
A I 1994
Order No. JX9094
$54.00

B30.8-1993
Floating Cranes and Floating Derricks
AI 1998
Order No. J12493
$52.00

B30.9-1996
Slings
A I 2001
Order No. J09296
$75.00

B30.10-1993
Hooks
A I 1998
Order No. J04893
$45.00

B30.11-1998
Monorails and Underhung Cranes
A I 2003
Order No. J12098
$52.00

B30.12-1992
Handling Loads Suspended From Rotorcraft
A I 1997
Order No. J04792
$47.00

B30.13-1996
Storage/Retrieval (S/R) Machines and Associated Equipment
A I 2001
Order No. J05296
$48.00

B30.14-1996
Side Room Tractors
A I 2001
Order No. J05996
$42.00

B30.16-1993
Overhead Hoists (Underhung)
A I 1998
Order No. J11693
$52.00

B30.17-1992
Overhead and Gantry Cranes (Top Running Bridge, Single Girder, Underhung Hoist)
A I 1997
Order No. JX6092
$40.00

B30.18-1993
Stacker Cranes (Top or Under Running Bridge, Multiple Girder with Top or Under Running Trolley Hoist)
A I 1998
Order No. J11393
$45.00

B30.19-1993
Cableways
A I 1998
Order No. J10893
$47.00

CRANES AND HOISTS cont'd

B30.20-1993
Below-the-Hook Lifting Devices
A I 1998
Order No. J10093
$53.00

B30.21-1994
Manually Lever Operated Hoists
A I 1999
Order No. J12694
$49.00

B30.22-1993
Articulating Boom Cranes
A I 1998
Order No. J11793
$53.00

B30.25-1998
Scrap and Material Handlers
A I 2003
Order No. J14698
$50.00

HST-1M-1989(R1995)
Performance Standard for Electric Chain Hoists
Order No. J07889
$29.00

HST-2M-1989(R1995)
Performance Standard for Hand Chain Manually Operated Chain Hoists
Order No. J08789
$29.00

HST-3M-1991(R1996)
Performance Standard for Manually Lever Operated Chain Hoists
Order No. J09891
$29.00

HST-4M-1991(R1996)
Performance Standard for Overhead Electric Wire Rope Hoists
Order No. J09991
$29.00

HST-5M-1991(R1996)
Performance Standard for Air Chain Hoists
Order No. J10291
$29.00

HST-6M-1986(R1995)
Performance Standard for Air Wire Rope Hoists
Order No. J00106
$29.00

DIMENSIONS
Standards for limits and fits, preferred sizes, and pipe dimensions.

B4.1-1967(R1994)
Preferred Limits and Fits for Cylindrical Parts
Order No. L00022
$30.00

B4.2-1978(R1994)
Preferred Metric Limits and Fits
Order No. L00054
$32.00

B4.3-1978(R1994)
General Tolerances for Metric Dimensioned Products
Order No. L00055
$30.00

B4.4M-1981(R1994)
Inspection of Workpieces
Order No. L00062
$30.00

B32.1-1952(R1994)
Preferred Thickness for Uncoated, Thin, Flat Metals (Under 0.250/in.)
Order No. L00025
$29.00

B32.2-1969(R1994)
Preferred Diameters for Round Wire—0.500 Inches and Under
Order No. M00012
$29.00

B32.3M-1984(R1994)
Preferred Metric Sizes for Flat Metal Products
Order No. L00045
$29.00

B32.4M-1980(R1994)
Preferred Metric Sizes for Round, Square, Rectangle and Hexagon Metal Products
Order No. L00046
$29.00

B32.5-1977(R1994)
Preferred Metric Sizes for Tubular Metal Products other than Pipe
Order No. L00051
$29.00

B32.6M-1984(R1994)
Preferred Metric Equivalents of Inch Sizes for Tubular Metal Products other than Pipe
Order No. L00052
$29.00

B36.10M-1996
Welded and Seamless Wrought Steel Pipe
Order No. M03196
$35.00

B36.19M-1985(R1994)
Stainless Steel Pipe
Order No. M00013
$29.00

DRAWINGS AND TERMINOLOGY
Standards for drafting, abbreviations, letter symbols, graphs, and charts.

Abbreviations
Y1.1-1989
Abbreviations—for Use on Drawings and in Text
Order No. J00389
$49.00

Drafting Standards
Y14.1-1995
Decimal Inch Drawing Sheet Size and Format
Order No. N00195
$32.00

Y14.1M-1995
Metric Drawing Sheet Size and Format
Order No. N12995
$32.00

Y14.2M-1992(R1998)
Line Conventions and Lettering
Order No. N00292
$32.00

Y14.3M-1994
Multi and Sectional View Drawings
Order No. N00394
$40.00

Y14.4M-1989(R1994)
Pictorial Drawing
Order No. N00489
$29.00

Y14.5M-1994
Dimensioning and Tolerancing
Order No. N00594
$93.00

Y14.5.1M-1994
Mathematical Definition of Dimensioning and Tolerancing
Order No. N13294
$49.00

Y14.5M-1994 PLUS Y14.5.1M-1994
Special Offer
Order No. YX1494
$120.00

Y14.5.2-1995
Certification of Geometric Dimensioning and Tolerancing Professionals
Order No. N13495
$32.00

Y14.6-1978(R1993)
Screw Thread Representation
Order No. N00006
$29.00

DRAWINGS AND TERMINOLOGY
cont'd

Y14.6M-1981(R1998)
Screw Thread Representation (Metric Supplement)
Order No. N00097
$29.00

Y14.7.1-1971(R1998)
Gear Drawing Standards—Part 1: For Spur, Helical, Double Helical and Rack
Order No. K00067
$29.00

Y14.7.2-1978(R1994)
Gear and Spline Drawing Standards—Part 2: Bevel and Hypoid Gears
Order No. N00081
$29.00

Y14.8M-1996
Castings and Forgings
Order No. N12096
$46.00

Y14.13M-1981(R1998)
Mechanical Spring Representation
Order No. N00098
$29.00

Y14.18M-1986(R1998)
Optical Parts
Order No. N00115
$29.00

Y14.24M-1989(R1996)
Types and Applications of Engineering Drawings
Order No. N00122
$62.00

Y14.32.1M-1994
Chassis Frames Passenger Car and Light Truck—Ground Vehicle Practices
Order No. N13394
$32.00

Y14.34M-1996
Associated Lists
Order No. N10296
$35.00

Y14.35M-1997
Revision of Engineering Drawings and Associated Documents
Order No. N13097
$35.00

Y14.36M-1996
Surface Texture Symbols
Order No. N08096
$44.00

Y14.100M-1998
Engineering Drawing Practices
Order No. N13898
$45.00

Y14
Technical Report 4-1989 A Structural Language Format for Basic Shape Description
Order No. N11989
$94.00

Graphic Symbols Standards
Y32.2.3-1949(R1994)
Redesignation of Z32.2.3-1949(R1953) Graphic Symbols for Pipe Fittings, Valves and Piping
Order No. K00006
$30.00

Y32.2.4-1949(R1998)
Graphic Symbols for Heating, Ventilating, and Air Conditioning
Order No. K00005
$29.00

Y32.2.6-1950(R1998)
Graphic Symbols for Heat-Power Apparatus
Order No. K00004
$29.00

Y32.4-1997(R1994)
Graphic Symbols for Plumbing Fixtures for Diagrams Used in Architecture and Building Construction
Order No. K00007
$29.00

Y32.7-1972(R1994)
Graphic Symbols for Railroad Maps and Profiles
Order No. K00008
$29.00

Y32.10-1967(R1994)
Graphic Symbols for Fluid Power Diagrams
Order No. N00022
$29.00

Y32.11-1961(R1998)
Graphic Symbols for Process Flow Diagrams in Petroleum and Chemical Industries
Order No. K00040
$29.00

Y32.18-1972(R1998)
Symbols for Mechanical and Acoustical Elements as Used in Schematic Diagrams
Order No. K00011
$29.00

ELEVATORS AND ESCALATORS
Safety codes and standards for the design, construction, installation, operation, inspection, testing, maintenance, alteration, and repair of elevators, dumbwaiters, escalators, moving walks, material lifts and dumbwaiters with automatic transfer devices, wheelchair lifts, and inclined-stairway chair lifts.

A17 CD-ROM for Elevators and Escalators
Order No. A00144
$495.00

A17.1-1996
Safety Code for Elevators and Escalators
A I 1999
Order No. A09696
$125.00

A17.1-1993
Safety Code for Elevators and Escalators
A I 1996
Order No. A09693
$114.00

A17.1-1990
Safety Code for Elevators and Escalators
Order No. AX9690
$103.00

Handbook on A17.1 Safety Code for Elevators and Escalators-1996 Edition
Order No. A11296
$105.00

A17.1 special Offer-A17.1-1996 Plus A17.1 Handbook (1996 Edition)
Order No. AX1796
$205.00

A17.2.1-1996
Inspector's Manual for Electric Elevators
A I 2000
Order No. A13496
$52.00

A17.2.2-1997
Inspector's Manual for Hydraulic Elevators
A I 2000
Order No. A13997
$54.00

A17.2.3-1994
Inspector's Manual for Escalators and Moving Walks
A I 1997
Order No. A14194
$42.00

A17.3-1996
Safety Code for Existing Elevators and Escalators
A I 2001
Order No. A12396
$55.00

A17.3-1990
Safety Code for Existing Elevators and Escalators
Order No. A12390
$47.00

A17.4-1991
Guide for Emergency Evacuation of Passengers from Elevators
Order No. A12591
$25.00

A17.5-1996
Elevator and Escalator Electrical Equipment
A I 2000
Order No. A13296
$69.00

ELEVATORS AND ESCALATORS
cont'd

Interpretations: A17 Documents 1972–1979
Order No. A00108
$25.00

A17 Interpretations Nos. 2–13 1989
Order No. A96187
$39.00

QEI-1-1997
Standard for the Qualification of Elevator Inspectors
A I L 2000
Order No. A12097
$49.00

Checklist for Inspection and Test of Elevators Used for Construction*
Order No. A49C10
$41.00

Checklist for Inspection of Electric Elevators*
Order No. A134C1
$39.00

Checklist for Inspection of Hydraulic Elevators*
Order No. A050C7
$28.00

Checklist for Inspection of Escalators and Moving Walks*
Order No. A141C1
$29.00

**In pad form (100 per pad)*

FASTENERS
Standards for bolts, nuts, rivets, washers, keys, eyebolts, pins, and retaining rings.

FAP-1-1995
Quality Assurance Program Requirements for Fastener Manufacturers and Distributors and Testing Laboratories
A I L 1999
Order No. M13995
$46.00

Metric Fasteners
B18.1.3M-1983 (R1995)
Metric Small Solid Rivets
Order No. M00131
$29.00

B18.2.3.1M-1979 (R1995)
Metric Hex Cap Screws
Order No. M00100
$29.00

B18.2.3.2M-1979 (R1995)
Metric Formed Hex Screws
Order No. M00101
$29.00

B18.2.3.3M-1979 (R1995)
Metric Heavy Hex Screws
Order No. M00102
$29.00

B18.2.3.4M-1984 (R1995)
Metric Hex Flange Screws
Order No. M00103
$29.00

B18.2.3.5M-1979 (R1995)
Metric Hex Bolts
Order No. M00104
$29.00

B18.2.3.6M-1979 (R1995)
Metric Heavy Hex Bolts
Order No. M00105
$29.00

B18.2.3.7M-1979 (R1995)
Metric Heavy Hex Structural Bolts
Order No. M00106
$29.00

B18.2.3.8M-1981 (R1991)
Metric Hex Lag Screws
Order No. M00107
$29.00

B18.2.3.9M-1984 (R1995)
Metric Heavy Hex Flange Screws
Order No. M00133
$29.00

B18.2.3.10M-1996
Square Head Bolts (Metric Series)
Order No. M15896
$28.00

B18.2.4.1M-1979 (R1995)
Metric Hex Nuts, Style 1
Order No. M00114
$29.00

B18.2.4.2M-1979 (R1995)
Metric Hex Nuts, Style 2
Order No. M00117
$29.00

B18.2.4.3M-1979 (R1995)
Metric Slotted Hex Nuts
Order No. M00119
$29.00

B18.2.4.4M-1982 (R1993)
Metric Hex Flange Nuts
Order No. M00122
$29.00

B18.2.4.5M-1979 (R1998)
Metric Hex Jam Nuts
Order No. M00115
$29.00

B18.2.4.6M-1979 (R1998)
Metric Heavy Hex Nuts
Order No. M00116
$29.00

B18.3.1M-1986 (R1993)
Socket Head Cap Screws
Order No. M00094
$29.00

B18.3.2M-1979 (R1998)
Metric Series Hexagon Keys and Bits
Order No. M00108
$29.00

B18.3.3M-1986 (R1993)
Hexagon Socket Head Shoulder Screws
Order No. M00109
$29.00

B18.3.4M-1986 (R1993)
Hexagon Socket Button Head Cap Screws
Order No. M00110
$29.00

B18.3.5M-1986 (R1993)
Hexagon Socket Flat Countersunk Head Cap Screws
Order No. M00130
$29.00

B18.3.6M-1986 (R1993)
Metric Series Socket Set Screws
Order No. M00111
$29.00

B18.5.2.1M-1996
Metric Round Head Short Square Neck Bolts
Order No. L06496
$32.00

B18.5.2.2M-1982 (R1993)
Metric Round Head Square Neck Bolts
Order No. M00124
$29.00

B18.5.2.3M-1990 (R1998)
Round Head Square Neck Bolts with Large Head
Order No. M00141
$29.00

B18.6.5M-1986 (R1993)
Metric Thread Forming and Thread Cutting Tapping Screws
Order No. N00110
$30.00

B18.6.7M-1985 (R1993)
Metric Machine Screws
Order No. M00136
$29.00

FASTENERS
cont'd

B18.7.1M-1984
(R1992)
Metric General Purpose Semi-Tubular Rivets
Order No. N00106
$29.00

B18.8.3M-1995
Spring Pins—Coiled Type
Order No. M14295
$34.00

B18.8.4M-1994
Spring Pins—Slotted
Order No. M14794
$32.00

B18.8.5M-1994
Machine Dowel Pins—Hardened Ground
Order No. M14894
$32.00

B18.8.6M-1995
Cotter Pins (Metric Series)
Order No. M14695
$32.00

B18.8.7M-1994
Headless Clevis Pins
Order No. M14994
$32.00

B18.8.8M-1994
Headed Clevis Pins
Order No. M15094
$32.00

B18.8.9M-1998
Grooved Pins (Metric Series)
Order No. M16598
$34.00

B18.12-1962(R1991)
Glossary of Terms for Mechanical Fasteners
Order No. M00046
$29.00

B18.13.1M-1991
Screw and Washer Assemblies—Sems
A I 1996
Order No. N00124
$42.00

B18.16.1M-1979
(R1995)
Mechanical and Performance Requirements for Prevailing Torque-Type Steel Metric Hex Nuts and Hex Flange Nuts
Order No. M00118
$29.00

B18.16.2M-1979
(R1995)
Torque-Tension Test Requirements for Prevailing-Torque Type Steel Metric Hex Nuts and Hex Flange Nuts
Order No. M00113
$29.00

B18.16.3M-1982
(R1993)
Dimensional Requirements for Prevailing-Torque Type Steel Metric Hex Nuts and Hex Flange Nuts
Order No. M00123
$29.00

B18.18.1M-1987
(R1993)
Inspection and Quality Assurance for General Purpose Fasteners
Order No. M00125
$29.00

B18.18.2M-1987
(R1993)
Inspection and Quality Assurance for High-Volume Machine Assembly Fasteners
Order No. M00126
$29.00

Supplement to B18.18.2
Order No. M0126S
$29.00

B18.18.3M-1987
(R1993)
Inspection and Quality Assurance for Special Purpose Fasteners
Order No. M00127
$29.00

B18.18.4M-1987
(R1993)
Inspection and Quality Assurance for Fasteners for Highly Specialized Engineered Applications
Order No. M00128
$29.00

B18.18.5M-1998
Inspection and Quality Assurance Plan Requiring In-Process Inspection and Control
Order No. M16298
$32.00

B18.18.6M-1998
Quality Assurance Plan for Fasteners Produced in a Third Party Accreditation System
Order No. M16398
$28.00

B18.18.7M-1998
Quality Assurance Plan for Fasteners Produced in a Customer Approved Control Plan
Order No. M16498
$29.00

B18.21.2M-1994
Lock Washers
Order No. M14094
$32.00

B18.22M-1981(R1990)
Metric Plain Washers
Order No. L00063
$29.00

B18.24.1-1996
Part Identifying Number (PIN) Code System Standard for B18 Externally Threaded Products
A I 2002
Order No. M15996
$55.00

B18.25.1M-1996
Square and Rectangular Keys and Keyways
Order No. M15696
$32.00

B18.25.2M-1996
Woodruff Keys and Keyways
Order No. M15796
$32.00

B27.7-1977(R1993)
General Purpose Tapered and Reduced Cross Section Retaining Rings (Metric)
Order No. L00053
$29.00

B27.8M-1978(R1993)
General Purpose Metric Tapered and Reduced Cross Section Retaining Rings-Type 3DM1-Heavy Duty External Rings, Type 3EM1-Reinforced E″ Rings, Type 3FM1-8C Type Rings
Order No. L00060
$29.00

Fasteners in Customary Units
B18.1.1-1972(R1995)
Small Solid Rivets
Order No. M00006
$29.00

B18.1.2-1972(R1995)
Large Rivets
Order No. K00023
$29.00

B18.2.1-996
Square and Hex Bolts and Screws (Inch Series)
A I 1999
Order No. M04496
$47.00

B18.2.2-1987(R1993)
Square and Hex Nuts (Inch Series)
Order No. M00043
$29.00

B18.2.6-1996
Fasteners for Use in Structural Applications
A I 1999
Order No. M16196
$34.00

FASTENERS
cont'd

B18.3-1998
Socket Cap, Shoulder, and Set Screws, Hex and Spline Keys (Inch Series)
Order No. M00798
$39.00

B18.5-1990(R1998)
Round Head Bolts (Inch Series)
Order No. L03290
$30.00

B18.6.1-1981(R1997)
Wood Screws (Inch Series)
Order No. M0005
$29.00

B18.6.2-1972(R1993)
Slotted Head Cap Screws, Square Head Set Screws and Slotted Headless Set Screws
Order No. K00012
$29.00

B18.6.3-1972(R1997)
Machine Screws and Machine Screw Nuts
Order No. M00004
$29.00

B18.6.4-1981(R1997)
Thread Forming and Thread Cutting Tapping Screws and Metallic Drive Screws (Inch Series)
Order No. N00021
$30.00

B18.7-1972(R1992)
General Purpose Semi-Tubular Rivets, Full Tubular Rivets, Split Rivets and Rivet Caps
Order No. N00059
$29.00

B18.8.1-1994
Clevis Pins and Cotter Pins
Order No. L02494
$32.00

B18.8.2-1995
Taper Pins, Dowel Pins, Straight Pins, Grooved Pins and Spring Pins (Inch Series)
Order No. L05695
$39.00

B18.9-1996
Plow Bolts (Inch Series)
Order No. L02196
$32.00

B18.10-1982(R1992)
Track Bolts and Nuts
Order No. M00025
$29.00

B18.11-1961(R1992)
Miniature Screws
Order No. K00037
$29.00

B18.13-1996
Screw and Washer Assemblies—Sems (Inch Series)
Order No. N04396
$34.00

B18.15-1985(R1995)
Forged Eyebolts
Order No. M00037
$29.00

B18.21.1-1994
Lock Washers (Inch Series)
Order No. L00294
$32.00

B18.22.1-1965(R1998)
Plain Washers (Redesignation of B27.2-1965)
Order No. L00019
$29.00

B18.29.1-1993
Helical Coil Screw Thread Inserts—Free Running and Screw Locking (Inch Series)
Order No. M14392
$32.00

B27.6-1972(R1993)
General Purpose Uniform Cross Section Spiral Retaining Rings
Order No. L00037
$29.00

FLOW MEASUREMENT
Standards for fluid flow volume measurements in pipes.

MFC-1M-1991
Glossary of Terms Used in the Measurement of Fluid Flow in Pipes
Order No. J06591
$29.00

MFC-2M-1983(R1988)
Measurement Uncertainty for Fluid Flow in Closed Conduits
Order No. K00112
$33.00

MFC-3M-1989(R1995)
Measurement of Fluid Flow in Pipes Using Orifice, Nozzle and Venturi (Not an American National Standard)
Order No. K11389
$40.00

MFC-4M-1986(R1997)
Measurement of Gas Flow by Turbine Meters
Order No. K00118
$29.00

MFC-5M-1985(R1994)
Measurement of Liquid Flow in Closed Conduits Using Transit-Time Ultrasonic Flowmeters
Order No. K00115
$29.00

MFC-6M-1998
Measurement of Fluid Flow in Pipes Using Vortex Flow Meters
Order No. K11798
$32.00

MFC-7M-1987(R1992)
Measurement of Gas Flow by Means of Critical Flow Venturi Nozzles
Order No. K00119
$29.00

MFC-8M-1988
Fluid Flow in Closed Conduits—Connections for Pressure Signal Transmissions Between Primary and Secondary Devices
Order No. K12188
$29.00

MFC-9M-1988
Measurement of Liquid Flow in Closed Conduits by Weighing Method
Order No. K12588
$29.00

MFC-10M-1994
Method for Establishing Installation Effects on Flowmeters
Order No. K12394
$29.00

MFC-11M-1989 (R1994)
Measurement of Fluid Flow by Means of Coriolis Mass Flowmeters
Order No. K12989
$29.00

MFC-14M-1995
Measurement of Fluid Flow Using Small Bore Precision Orifice Meters
Order No. K13795
$30.00

MFC-16M-1995
Measurement of Fluid Flow in Closed Conduits by Means of Electromagnetic Flowmeters
Order No. K13895
$30.00

GAGE BLANKS
Standard for dimensions of standard blanks and components.

B47.1-1988(R1994)
Gage Blanks
Order No. M08288
$70.00

GAUGES
Standardizes principal features of gauges.

B40.1-1991
Gauges—Pressure Indicating Dial Type—Elastic Element
Order No. K01591
$28.00

B40.2-1991
Diaphragm Seals
Order No. K13191
$29.00

B40.3-1997
Bimetallic Actuated Thermometers
Order No. K13097
$32.00

B40.5-1995
Snubbers
Order No. K13695
$29.00

B40.6-1994
Pressure Limiter Valves
Order No. K13594
$29.00

HIGH PRESSURE SYSTEMS

HPS-1994
High Pressure Systems
A I 1999
Order No. A14094
$75.00

INDUSTRIAL TRUCKS
Safety standards relating to the elements of design, operation, and maintenance; also the standardization relating to principal dimensions to facilitate interchangeability, test methods, and test procedures of industrial trucks.

B56.1-1993
Safety Standard for Low Lift and High Lift Trucks
A I 1998
Order No. J08593
$69.00

B56.5-1993
Safety Standard for Guided Industrial Vehicles and Automated Functions of Manned Industrial Vehicles
A I 1998
Order No. J05793
$33.00

B56.6-1992
Safety Standard for Rough Terrain Forklift Trucks
A I 1997
Order No. J05692
$55.00

B56.7-1987(R1992)
Safety Standard for Industrial Crane Trucks
A I 1997
Order No. J07387
$40.00

B56.8-1993
Safety Standard for Personnel and Burden Carriers
A I 1996
Order No. J06893
$38.00

B56.9-1992
Safety Standard for Operator Controlled Industrial Tow Tractors
A I 1997
Order No. J11492
$40.00

B56.10-1992
Safety Standard for Manually Propelled High Lift Industrial Trucks
A I 1997
Order No. K13292
$50.00

B56.11.1-1992
Double Race or Bi-Level Swivel and Rigid Industrial Casters
A I 1995
Order No. K05192
$35.00

B56.11.3-1992
Load Handling Symbols for Powered Industrial Trucks
A I 1995
Order No. K07392
$32.00

B56.11.4-1992
Hook-Type Forks and Fork Carriers for Powered Industrial Forklift Trucks
A I 1995
Order No. K07492
$34.00

B56.11.5-1992
Measurement of Sound Emitted by Low Lift, High Lift, and Rough Terrain Powered Industrial Trucks
A I 1995
Order No. K12892
$34.00

B56.11.6-1992
Evaluation of Visibility from Powered Industrial Trucks
A I 1995
Order No. K13392
$35.00

KEYS
Standards for keys and keyseats.

B17.1-1967(R1998)
Keys and Keyseats
Order No. J00038
$29.00

B17.2-1967(R1998)
Woodruff Keys and Keyseats
Order No. J00037
$29.00

MACHINE GUARDING
Safety standard applying to any source of hazard to personnel from the operation of mechanical power transmission apparatus on machines, equipment, or systems that are stationary in their use, other than the point of operation.

B15.1-1996
Safety Standard For Mechanical Power Transmission Apparatus
A I 1999
Order No. J09496
$32.00

MANLIFTS
Safety standard for the manufacture, installation, maintenance, inspection, and operation of endless belt manlifts.

A90.1-1997
Safety Standard for Belt Manlifts
A I 2002
Order No. J10197
$39.00

MEASUREMENT
Standards for dimensional metrology and calibration of instruments.

B89
Technical Paper 1990, Space Plate Test Recommendations for Coordinate Measuring Machines
Order No. L07190
$29.00

B89
Technical Report 1990, Parametric Calibration of Coordinate Measuring Machines
Order No. L00070
$29.00

B89.1.2M-1991
Calibration of Gage Blocks by Contact Comparison Methods (Through 20 in. and 500 mm)
Order No. L07291
$32.00

B89.1.6M-1984 (R1997)
Measurement of Qualified Plain Internal Diameters for Use as Master Rings and Ring Gages
Order No. L00048
$29.00

B89.1.9M-1984 (R1997)
Precision Gage Blocks for Length Measurement (Through 20 in. and 500 mm)
Order No. L00044
$29.00

B89.1.10M-1987 (R1995)
Dial Indicators (for Linear Measurements)
Order No. L00057
$29.00

B89.1.12M-1990
Methods for Performance Evaluation of Coordinate Measuring Machines
Order No. L06590
$35.00

B89.3.1-1972(R1997)
Measurement of Out-of-Roundness
Order No. L00020
$29.00

B89.3.4M-1985 (R1992)
Axes of Rotation—Methods for Specifying and Testing
Order No. L00066
$29.00

B89.4.1-1997
Methods for Performance Evaluation of Coordinate Measuring Machines
Order No. L06597
$45.00

B89.6.2-1973(R1995)
Temperature and Humidity Environment for Dimensional Measurement
Order No. L00047
$29.00

MC88.1-1972(R1995)
Guide for Dynamic Calibration of Pressure Transducers
Order No. L00042
$29.00

MC88.2-1974(R1995)
Redesignation of B88.2-1974(R1981) Procedures for Bench Calibration of Tank Level Gaging Tapes and Sounding Rules
Order No. L00043
$29.00

METRIC SYSTEM
Guides for familiarization with the International System (SI) of Units.

ASME Guide SI-1
ASME Orientation and Guide for Use of SI (Metric) Units Ninth Edition
1982
Order No. E00058
$23.00

SI-2
ASME Text Booklet, SI Units in Strength of Materials Second Edition
1976 14 pp.
Order No. E00082
$23.00

SI-3
ASME Text Booklet, SI Units in Dynamics
1976 20 pp.
Order No. E00083
$23.00

SI-4
ASME Text Booklet, SI Units in Thermodynamics
1976 55 pp.
Order No. E00084
$23.00

SI-5
ASME Text Booklet, SI Units in Fluid Mechanics
1976 36 pp.
Order No. E00085
$23.00

SI-6
ASME Text Booklet, SI Units in Kinematics
1976 14 pp.
Order No. E00086
$23.00

SI-7
ASME Text Booklet, SI Units in Heat Transfer
1977 36 pp.
Order No. E00087
$23.00

SI-8
ASME Text Booklet, SI Units in Vibration
1976 13 pp.
Order No. E00088
$23.00

SI-9
ASME Guide for Metrication of Codes and Standards SI (Metric) Units
1980 33 pp.
Order No. E00089
$23.00

SI-10
Steam Charts, SI (Metric) and U.S. Customary Units
1976 128 pp.
Hardbound
Order No. E00090
$28.00

NUCLEAR
Codes and standards applicable to nuclear facilities and technology.

AG-1-1997
Code on Nuclear Air and Gas Treatment
A I 2000
Order No. A12297
$152.00

AG-1-1994
Code on Nuclear Air and Gas Treatment
A I 1997
Order No. A12294
$139.00

B16.41-1983(R1989)
Functional Qualification Requirements for Power Operated Active Valve Assemblies for Nuclear Power Plants
Order No. J00077
$32.00

N278.1-1975(R1992)
Self-Operated and Power-Operated Safety-Related Valves Functional Specification Standard
Order No. A00084
$32.00

N509-1989(R1996)
Nuclear Power Plant Air-Cleaning Units and Components
A I 1994
Order No. A08589
$49.00

N510-1989(1995)
Testing of Nuclear Air-Treatment Systems
I 1994
Order No. A08289
$37.00

N626-1990
Qualifications and Duties for Authorized Nuclear Inspection Agencies and Personnel
A I L 1994
Order No. J09590
$40.00

N626-1985
Qualifications and Duties for Authorized Nuclear Inspection Agencies and Personnel
A I 1990
Order No. J09585
$40.00

N626.0-1982
Qualifications and Duties for Authorized Inspection Agencies, Nuclear Inspectors and Nuclear Inspector Supervisors
Order No. J00074
$29.00

N626.1-1982
Qualifications and Duties for Authorized Inspection Agencies, Nuclear Inservice Inspectors and Nuclear Inservice Inspector Supervisors
Order No. J00075
$29.00

N626.2-1982
Qualifications and Duties for Authorized Inspection Agencies, Nuclear Inspectors (Concrete) and Nuclear Inspector Supervisors (Concrete)
Order No. J00076
$29.00

N626.3-1993
Qualifications and Duties of Specialized Professional Engineers
A I 1998
Order No. A10093
$32.00

NOG-1-1995
Rules for Construction of Overhead And Gantry Cranes (Top Running Bridge, Multiple Girder)
A I 2000
Order No. A11995
$125.00

NQA-1-1997
Quality Assurance Requirements for Nuclear Facility Applications
A I 2000
Order No. A10597
$128.00

NQA-1-1994
Quality Assurance Program Requirements for Nuclear Facilities
A I 1997
Order No. A10594
$125.00

NQA-1-1989
Quality Assurance Program Requirements for Nuclear Facilities
A I 1992
Order No. A10589
$83.00

Comparison of NQA-1 and ISO 9001
Order No. A13793
$30.00

NUM-1-1996
Rules for Construction of Cranes, Monorails, and Hoists (with Bridge or Trolley or Hoist of the Underhung Type)
A I 1999
Order No. J14396
$115.00

OM-1998
Code for the Operation and Maintenance of Nuclear Power Plants
A I
Order No. A12898
$185.00

OM-1995
Code for Operation and Maintenance of Nuclear Power Plants
A I 1998
Order No. A12894
$175.00

OM-1990
Code for Operation and Maintenance of Nuclear Power Plants
A I 1993
Order No. AX2890
$175.00

OM-S/G-1997
Standards and Guides for Operation and Maintenance of Nuclear Power Plants
A I 2000
Order No. A13197
$175.00

OM-S/G-1994
Standards and Guides for Operation and Maintenance of Nuclear Power Plants
A I 1998
Order No. A13194
$175.00

QME-1-1997
Qualification of Active Mechanical Equipment Used in Nuclear Power Plants
A I L 2000
Order No. A13697
$105.00

OFFSHORE
Safety standards for quality assurance programs used in the manufacturing, remanufacturing, and assembly of valves for offshore service.

SPPE-1-1994
Quality Assurance and Certification of Safety and Pollution Prevention Equipment Used in Offshore Oil and Gas Operations
A L
Order No. A09394
$43.00

SPPE-2-1994
Accreditation of Testing Laboratories For Safety and Pollution Prevention Equipment Used In Offshore Oil and Gas Operations
A L
Order No. A09494
$45.00

OPERATOR QUALIFICATION AND CERTIFICATION

QHO-1-1994
Standard for the Qualification and Certification of Hazardous Waste Incinerator Operators
A I L 1999
Order No. A13894
$41.00

QMO-1-1993
Standard for the Qualification and Certification of Medical Waste Incinerator Operators
A I L 1998
Order No. A13393
$47.00

QRO-1-1994
Standard for the Qualification and Certification of Resource Recovery Facility Operators
A I L 1997
Order No. A13094
$49.00

PALLETS
Standards for terminology, sizes, and testing of pallets.

MH1-1997
Pallets, Slip Sheets, and Other Bases for Unit Loads
A 2000
Order No. K14097
$85.00

MH1.2.2M-1989(R1996)
Pallet Sizes
Order No. K08289
$32.00

MH1.5M-1993
Slip Sheets
Order No. K10393
$32.00

MH1.8M-1996
Wood Pallets
Order No. K13496
$35.00

MH1.9-1993
Export Pallets
Order No. K13393
$29.00

PERFORMANCE TEST CODES
ASME Performance Test Codes are applicable to the determination of performance of specific mechanical equipment that is designed to meet specified criteria for performance and operability. The codes provide requirements for instrumentation, test procedure, test environment, and interpretation of test results.

PTC 1-1991
General Instructions
Order No. C01791
$37.00

PTC 2-1980(R1985)
Code on Definitions and Values
Order No. C00007
$35.00

PTC 3.1-1958(R1992)
Diesel and Burner Fuels
Order No. D00024
$37.00

PTC 3.2-1990
Coal and Coke
Order No. C02390
$63.00

PTC 3.3-1969(R1992)
Gaseous Fuels
Order No. D00038
$37.00

PTC 4.1-1964(R1991)
Steam-Generating Units (with 1968 and 1969 Addenda)
Order No. C00025
$42.00

Diagram for Testing of a Steam Generator, Figure 1 (Pad of 100)
Order No. C00038
$23.00

Heat Balance of a Steam Generator, Figure 2 (Pad of 100)
Order No. C00039
$23.00

PTC 4.1a-1964
ASME Test Form for Abbreviated Efficiency Test—Summary Sheet (Pad of 100)
Order No. C00036
$23.00

PTC 4.1b-1964(R1965)
ASME Test for Abbreviated Efficiency Test—Calculation Sheet (Pad of 100)
Order No. C00037
$23.00

PTC 4.2 1969(R1997)
Coal Pulverizers
Order No. C00006
$37.00

PTC 4.3-1968(R1991)
Air Heaters
Order No. D00034
$35.00

PTC 4.4-1981(R1992)
Gas Turbine Heat Recovery Steam Generators
Order No. D00043
$42.00

PTC 5-1949
Reciprocating Steam Engines
Order No. C00022
$32.00

PTC 6-1996
Steam Turbines
Order No. C02896
$95.00

PTC 6A-1982(R1995)
Appendix A to Test Code for Steam Turbines
Order No. C00029
$39.00

PTC 6 on Steam Turbines—Interpretations
1977–1983
Order No. C00051
$32.00

PTC 6 Report 1985 (R1997)
Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines
A I 1991
Order No. D04186
$52.00

PTC 6S-1988(R1995)
Procedures for Routine Performance Test of Steam Turbines
I 1992
Order No. D04288
$74.00

PERFORMANCE TEST CODES
cont'd

PTC 7.1-1962(R1969)
Displacement Pumps
Order No. D00026
$32.00

PTC 8.2-1990
Centrifugal Pumps
I 1994
Order No. C00590
$49.00

PTC 9-1970(R1992)
Displacement Compressors, Vacuum Pumps and Blowers (With 1972 Errata)
Order No. CO0009
$34.00

PTC 10-1965(R1992)
Compressors and Exhausters
Order No. C00004
$25.00

PTC 11-1984(R1995)
Fans
Order No. C05284
$54.00

PTC 12.1-1978 (R1987)
Closed Feedwater Heaters
Order No. C00013
$32.00

PTC 12.2-1998
Steam Surface Condensers
Order No. C02698
$150.00

PTC 12.3-1997
Deaerators
Order No. D02397
$49.00

PTC 12.4-1992(R1997)
Moisture Separator Reheaters
Order No. C05992
$59.00

PTC 17-1973(R1997)
Reciprocating Internal-Combustion Engines
Order No. C00020
$32.00

PTC 18-1992
Hydraulic Prime Movers
Order No. C01892
$44.00

PTC 18.1-1978 (R1984)
Pumping Mode of Pump/Turbines
Order No. C00044
$32.00

PTC 19.1-1985 (R1990)
Instruments and Apparatus: Part 1 Measurement Uncertainty
I 1991
Order No. D04585
$55.00

PTC 19.2-1987
Instruments and Apparatus: Part 2 Pressure Measurement
A I 1992
Order No. D02987
$74.00

PTC 19.3-1974 (R1986)
Temperature Measurement
Order No. C00035
$38.00

PTC 19.5-1972
Application, Part II of Fluid Meters: Interim Supplement on Instruments and Apparatus
Order No. G00018
$23.00

PTC 19.5.1-1964
Weighing Scales
Order No. D00028
$32.00

PTC 19.7-1980 (R1988)
Measurement of Shaft Power
Order No. D00009
$32.00

PTC 19.8-1970 (R1985)
Measurement of Indicated Power
Order No. D00008
$32.00

PTC 19.10-1981
Part 10 Flue and Exhaust Gas Analyses
Order No. C00031
$43.00

PTC 19.11-1997
Steam and Water Sampling, Conditioning, and Analysis in the Power Cycle
Order No. D01197
$75.00

PTC 19.14-1958
Linear Measurements
Order No. D00014
$32.00

PTC 19.22-1986
Digital Systems Techniques
I 1991
Order No. C05486
$50.00

PTC 19.23-1980 (R1985)
Part 23 Guidance Manual for Model Testing
Order No. C00047
$35.00

PTC 20.1-1977 (R1988)
Speed and Load-Governing Systems for Steam Turbine-Generator Units
Order No. C00019
$32.00

PTC 20.2-1965 (R1986)
Overspeed Trip Systems for Steam Turbine-Generator Units
Order No. C00041
$32.00

PTC 20.3-1970 (R1986)
Pressure Control Systems Used on Steam Turbine Generator Units
Order No. C00027
$32.00

PTC 21-1991
Particulate Matter Collection Equipment
I 1996
Order No. D02191
$45.00

PTC 22-1997
Performance Test Code on Gas Turbines
I 2002
Order No. C01597
$75.00

PTC 23-1986(R1997)
Atmospheric Water Cooling Equipment
A 1992
Order No. C05586
$50.00

PTC 24-1976(R1982)
Ejectors
Order No. C00010
$32.00

PTC 25-1994
Pressure Relief Devices
Order No. C06194
$54.00

PERFORMANCE TEST CODES
cont'd

PTC 26-1962
Speed Governing Systems for Internal Combustion Engine Generator Units
Order No. D00027
$32.00

PTC 28-1965(R1985)
Determining the Properties of Fine Particulate Matter
Order No. D00030
$32.00

PTC 29-1965(R1985)
Speed-Governing Systems for Hydraulic Turbine-Generator Units
Order No. C00040
$32.00

PTC 30-1991(R1998)
Air Cooled Heat Exchangers
A I 1996
Order No. CO0057
$95.00

PTC 31-1972(R1991)
Ion Exchange Equipment
Order No. C00016
$32.00

PTC 32.1-1969 (R1992)
Nuclear Steam Supply Systems
Order No. C00012
$32.00

PTC 32.2 Report—1979(R1992)
Methods of Measuring the Performance of Nuclear Reactor Fuel in Light Water Reactors
Order No. C00045
$23.00

PTC 33-1978(R1991)
Large Incinerators
Order No. CX0046
$46.00

PTC 33a-1980(R1991)
Appendix to PTC 33-1978
Order No. C0046A
$23.00

ASME Form for Abbreviated Incinerator Efficiency Test
{Form PTC 33a-1980(R1991)}
Order No. C0046F
$23.00

PTC 36-1985
Measurement of Industrial Sound
Order No. C00053
$32.00

PTC 38-1980(R1985)
Determining the Concentration of Particulate Matter in a Gas Stream
Order No. C00049
$35.00

PTC 39.1-1980 (R1991)
Condensate Removal Devices for Steam Systems
Order No. C00048
$32.00

PTC 40-1991
Flue Gas Desulfurization Units
I 1996
Order No. CO5891
$49.00

PTC 42-1988
Wind Turbines
I 1993
Order No. CO5688
$33.00

PTC 46-1997
Overall Plant Performance
Order No. C06496
$250.00

PTC PM-1993
Performance Monitoring Guidelines for Steam Power Plants
Order No. C06093
$73.00

PIPING
Standards for the identification, design, and welding of piping.

A13.1-1996
Scheme for the Identification of Piping Systems
Order No. L03396
$30.00

B31G-1991
Manual for Determining the Remaining Strength of Corroded Pipelines: A Supplement to B31, Code for Pressure Piping
Order No. A12191
$30.00

B31.1-1998
Power Piping
A I C 2001
Order No. A05898
$164.00

B31.1-1995
Power Piping
A I 1998
Order No. A05895
$140.00

B31.2-1968
Fuel Gas Piping
Order No. A00036
$29.00

B31.3-1996
Process Piping
A I 1999
Order No. A03796
$220.00

B31.3-1993
Chemical Plant and Refinery Piping, Plus Addenda
A I
Order No. A03793
$189.00

B31.4-1992
Pipeline Transportion Systems for Liquid Hydrocarbons and Other Liquids
A I C 1995
Order No. A03892
$72.00

B31.4-1989
Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids
A I C 1992
Order No. A03889
$66.00

B31.5-1992
Refrigeration Piping
A I C 1997
Order No. A04092
$61.00

B31.8-1995
Gas Transmission and Distribution Systems
A I 1998
Order No. A04595
$103.00

B31.9-1996
Building Services Piping
A I 1999
Order No. A11696
$72.00

B31.11-1989(R1998)
Slurry Transportation Piping Systems
A I C 1994
Order No. AI2489
$79.00

PLUMBING
Standards for plumbing equipment.

A112.1.2-1991(R1998)
Air Gaps in Plumbing Systems
Order No. J04191
$29.00

A112.3.1-1993
Performance Standard and Installation Procedures for Stainless Steel Drainage Systems, for Sanitary, Storm, and Chemical Applications, Above and Below Ground
Order No. J13693
$32.00

A112.4.1-1993
Water Heater Relief Valve Drain Tubes
Order No. J13793
$28.00

A112.6.1M-1997
Floor-Affixed Supports for Off-the-Floor Plumbing Fixtures for Public Use
Order No. J04397
$34.00

A112.14.1-1975 (R1998)
Backwater Valves
Order No. J00035
$29.00

A112.18.1M-1996
Plumbing Fixture Fittings
Order No. J02996
$33.00

A112.18.3M-1996
Performance Requirements for Backflow Devices and Systems in Plumbing Fixture Fittings
Order No. J14196
$31.00

A112.19.1M-1994
Enameled Cast Iron Plumbing Fixtures
Order No. J01194
$33.00

A112.19.2M-1995
Vitreous China Plumbing Fixtures
Order No. J01295
$42.00

A112.19.3M-1987 (R1996)
Stainless Steel Plumbing Fixtures (Designed for Residential Use)
Order No. J00050
$30.00

A112.19.4M-1994
Porcelain Enameled Formed Steel Plumbing Fixtures
Order No. J06494
$33.00

A112.19.5-1979 (R1998)
Trim for Water-Closet Bowls, Tanks and Urinals
Order No. J00058
$29.00

A112.19.6-1995
Hydraulic Requirements for Water Closets and Urinals
Order No. J13095
$32.00

A112.19.7M-1995
Whirlpool Bathtub Appliances
Order No. J11995
$28.00

A112.19.8M-1987 (1996)
Suction Fittings for Use in Swimming Pools, Wading Pools, Spas, Hot Tubs, and Whirlpool Bathtub Appliances
Order No. J00118
$29.00

A112.19.9M-1991 (R1998)
Non-Vitreous Ceramic Plumbing Fixtures
Order No. J13291
$29.00

A112.19.10-1994
Dual Flush Devices for Water Closets
Order No. J13894
$29.00

A112.21.1M-1991 (R1998)
Floor Drains
Order No. J04090
$30.00

A112.21.2M-1983
Roof Drains
Order No. J00002
$29.00

A112.21.3M-1985 (R1995)
Hydrants for Utility and Maintenance Use
Order No. J00051
$29.00

A112.36.2M-1991 (R1998)
Cleanouts
Order No. J04591
$31.00

POWERED PLATFORMS
Standards that establish safety requirements for powered platforms.

A120.1-96
Powered Platforms for Building Maintenance
A I 1999
Order No. A14296
$44.00

PRESSURE VESSELS

Boiler and Pressure Vessel Code

BPE-1997
Bioprocessing Equipment
A I 2000
Order No. A14397
$165.00

PVHO-1-1997
Safety Standard for Pressure Vessels for Human Occupancy
A I 2000
Order No. A09297
$125.00

PVHO-1-1993
Safety Standard for Pressure Vessels for Human Occupancy
Order No. A09293
$125.00

PUMPS
Standards on pumps for chemical process service.

B73.1M-1991
Specification for Horizontal End Suction Centrifugal Pumps for Chemical Process
Order No. J01991
$29.00

B73.2M-1991
Specification for Vertical In-Line Centrifugal Pumps for Chemical Process
Order No. J04991
$29.00

B73.3M-1997
Specification for Sealless Horizontal End Suction Centrifugal Pumps for Chemical Process
A I 2002
Order No. J14597
$45.00

B73.5M-1995
Thermoplastic and Thermoset Polymer Material Horizontal End Suction Centrifugal Pumps for Chemical Process
Order No. J13995
$35.00

REINFORCED THERMOSET PLASTIC CORROSION RESISTANT EQUIPMENT
Rules of safety governing the design, fabrication, and inspection during construction of reinforced thermoset plastic equipment.

RTP-1-1995
Reinforced Thermoset Plastic Corrosion Resistant Equipment
A I L 2000
Order No. J12795
$139.00

RTP-1-1992
Reinforced Thermoset Plastic Corrosion Resistant Equipment
Order No. J12792
$139.00

SCREW THREADS
Standards that cover dimensional data.

B1.1-1989
Unified Inch Screw Threads (UN and UNR Thread Form)
Order No. M02889
$49.00

B1.2-1983 (R1991)
Gages and Gaging for Unified Inch Screw Threads
Order No. N00058
$43.00

B1.3M-1992
Screw Thread Gaging Systems for Dimensional Acceptability—Inch and Metric Screw Threads (UN, UNR, UNJ, M, and MJ)
Order No. N09692
$32.00

B1.5-1988(R1994)
Acme Screw Threads
Order No. N02788
$45.00

B1.7M-1984(R1992)
Nomenclature, Definitions, and Letter Symbols for Screw Threads
Order No. L00011
$32.00

B1.8-1988(R1994)
Stub Acme Screw Threads
Order No. N02888
$34.00

B1.9-1973(R1992)
Buttress Inch Screw Threads 7°/45° Form with 0.6 Pitch Basic Height of Thread Engagement
Order No. N00029
$32.00

B1.10M-1997
Unified Miniature Screw Threads
Order No. N02597
$32.00

B1.11-1958(R1994)
Microscope Objective Thread
Order No. N00020
$32.00

B1.12-1987(R1992)
Class 5 Interference-Fit Thread
Order No. N00032
$40.00

B1.13M-1995
Metric Screw Threads—M Profile
Order No. N04695
$37.00

B1.15-1995
Unified Inch Screw Threads (UNJ Thread Form)
Order No. N13195
$44.00

B1.16M-1984(R1992)
Gages and Gaging for Metric M Screw Threads
Order No. N00026
$44.00

B1.20.1-1983(R1992)
Pipe Threads, General Purpose (Inch)
Order No. N00103
$32.00

B1.20.3-1976(R1991)
Dryseal Pipe Threads (Inch)
Order No. N00076
$32.00

B1.20.5-1991
Gaging for Dryseal Pipe Threads (Inch)
Order No. N09290
$32.00

B1.20.7-1991
Hose Coupling Screw Threads (Inch)
Order No. K01491
$32.00

B1.21M-1997
Metric Screw Threads: MJ Profile
Order No. N09097
$39.00

B1.22M-1985(R1992)
Gages and Gaging for MJ Series Metric Screw Threads
Order No. N00086
$50.00

B1.30-1992
Screw Threads—Standard Practice for Calculating and Rounding Dimensions
Order No. N12692
$32.00

B1 Technical Report: Measurement Uncertainty for 60° Screw Thread Gage Element (Inch and Metric)
Order No. N12893
$29.00

STEEL STACKS
Standard that specifies design, construction, and recommendations for inspection.

STS-1-1992
Steel Stacks
Order No. L06992
$72.00

STORAGE TANKS
Standard for the design and fabrication of welded aluminum-alloy storage tanks.

B96.1-1993
Welded Aluminum-Alloy Storage Tanks
A I 1998
Order No. A12693
$69.00

SURFACE QUALITY
Standard for classification of surfaces.

B46.1-1995
Surface Texture (Surface Roughness, Waviness and Lay)
Order No. M01995
$69.00

TOOLS
Standards for machine tools, consumable tools, and hand tools.

Machine Tools

B5.1M-1985(R1992)
T-Slots, Their Bolts, Nuts and Tongues
Order No. M00024
$30.00

B5.5-1959(R1992)
Rotating Air Cylinders and Adapters
Order No. L00031
$30.00

B5.8-1972(R1994)
Chucks and Chuck Jaws
Order No. J00024
$30.00

B5.9-1967(R1992)
Spindle Noses for Tool Room Lathes, Engine Lathes, Turret Lathes and Automatic Lathes
Order No. M00010
$30.00

B5.10-1994
Machine Tapers
Order No. L00494
$35.00

B5.11-1964(R1994)
Spindle Noses and Adjustable Adaptors for Multiple Spindle Drilling Heads
Order No. M00008
$30.00

B5.16-1952(R1992)
Accuracy of Engine and Tool Room Lathes
Order No. J00004
$30.00

B5.18-1972(R1998)
Spindle Noses and Tool Shanks for Milling Machines
Order No. M00009
$30.00

B5.25-1978(R1992)
Punch and Die Sets
Order No. L00027
$30.00

B5.25M-1980(R1992)
Punch and Die Sets (Metric)
Order No. L00061
$30.00

B5.28-1971(R1994)
Mounting Dimensions of Lubricating and Coolant Pumps for Machine Tools
Order No. L00009
$30.00

B5.32-1977(R1994)
Grinding Machines, Surface, Reciprocating Table-Horizontal Spindle
Order No. J00028
$30.00

B5.32.1-1977(R1994)
Grinding Machines, Surface, Reciprocating Table-Vertical Spindle
Order No. J00053
$30.00

B5.33-1981(R1994)
External Cylindrical Grinding Machines—Pain
Order No. M00038
$30.00

B5.35-1983(R1994)
Machine Mounting Specifications for Abrasive Discs and Plate Mounted Wheels
Order No. M00035
$32.00

B5.37-1970(R1994)
External Cylindrical Grinding Machines—Centerless
Order No. M00039
$30.00

B5.40-1977(R1991)
Spindle Noses and Tool Shanks for Horizontal Boring Machines
Order No. M00048
$30.00

B5.42-1981(R1994)
External Cylindrical Grinding Machines Universal
Order No. M00040
$30.00

B5.43-1977(R1994)
Modular Machine Tool Standards
Order No. M00021
$30.00

B5.43M-1979(R1994)
Modular Machine Tool Standards (Metric)
Order No. M00112
$30.00

B5.44-1971(R1992)
Rotary Table Surface Grinding Machines
Order No. M00061
$30.00

B5.45-1972(R1991)
Milling Machines
Order No. M00067
$30.00

B5.46-1972(R1994)
Symbols for Machine Tool Indicator Plates
Order No. M00065
$30.00

B5.47-1972(R1991)
Milling Machine Arbor Assemblies
Order No. M00074
$30.00

B5.48-1977(R1994)
Ball Screws
Order No. M00089
$30.00

B5.49-1998
Glossary of Power Press Terms
Order No. M09098
$32.00

B5.50-1994
"V" Flange Tool Shanks for Machining Centers with Automatic Tool Changers
Order No. M09794
$29.00

B5.51M-1979(R1994)
Preferred SI Units for Machine Tools
Order No. M00099
$30.00

B5.52M-1980(R1994)
Mechanical Power Presses General Purpose Single Point, GAP Type (Metric)
Order No. M00120
$38.00

B5.53M-1982(R1994)
Cutter and Tool Grinding Machines
Order No. M00129
$30.00

B5.54M-1992
Methods for Performance Evaluation of Computer Numerically Controlled Machine Centers
Order No. M14492
$52.00

TOOLS
cont'd

B5.55M-1994
Specification and Performance Standard, Power Press Brakes
Order No. M15194
$30.00

B5.56M-1994
Specification and Performance Standard, Power Shears
Order No. M15294
$30.00

Consumable Tools
B94.2-1995
Reamers
Order No. L02895
$35.00

B94.3-1965(R1995)
Straight Cut-Off Blades for Lathes and Screw Machines
Order No. M00056
$29.00

B94.6-1984(R1995)
Knurling
Order No. K00022
$29.00

B94.7-1980(R1995)
Hobs
Order No. N00056
$29.00

B94.8-1967(R1987)
Inserted Blade Milling Cutter Bodies
Order No. K00044
$29.00

B94.9-1987
Taps-Cut and Ground Threads
Order No. K04387
$40.00

B94.11M-1993
Twist Drills
Order No. M02793
$35.00

B94.14-1968(R1995)
Punches—Basic Head Type
Order No. K00052
$29.00

B94.14.1-1977(R1995)
Punches—Basic Head Type (Metric)
Order No. M00092
$29.00

B94.16-1987(R1995)
Retainers—Basic Ball-Lock, Punch and Die Button, Light and Heavy Duty
Order No. K05487
$29.00

B94.16.1-1978(R1995)
Retainers—Basic Ball-Lock, Punch and Die Button, Light and Heavy Duty (Metric)
Order No. M00095
$29.00

B94.17-1987(R1995)
Gages—Functional, Ball-Lock Punches, Die Buttons, and Retainers
Order No. K05787
$29.00

B94.17.1-1977(R1995)
Gages—Functional, Ball-Lock Punches, Die Buttons and Retainers (Metric)
Order No. M00093
$29.00

B94.18-1987(R1995)
Punches—Basic Ball-Lock, Light and Heavy Duty
Order No. K05587
$29.00

B94.18.1-1977(R1995)
Punches—Basic Ball-Lock Light and Heavy Duty (Metric)
Order No. M00087
$29.00

B94.19-1997
Milling Cutters and End Mills
Order No. L00897
$44.00

B94.21-1968(R1995)
Gear Shaper Cutters
Order No. J00007
$29.00

B94.22-1968(R1995)
Punches—Variable Head Type
Order No. L00041
$29.00

B94.22.1-1977(R1995)
Punches—Variable Head Type (Metric)
Order No. L00050
$29.00

B94.23-1969(R1995)
Punch Guide Bushings—Variable, Press Fit
Order No. L00040
$29.00

B94.27-1970(R1995)
Die Buttons—Basic Taper Relief, Press Fit
Order No. M00041
$29.00

B94.27.1M-1983 (R1995)
Die Buttons—Basic Taper Relief, Press Fit (Metric)
Order No. M00132
$29.00

B94.28-1970(R1995)
Die Buttons—Basic Straight Relief, Press Fit
Order No. M00045
$29.00

B94.28.1M-1984 (R1995)
Die Buttons—Basic Straight Relief, Press Fit (Metric)
Order No. M00134
$29.00

B94.29-1970(R1995)
Die Buttons—Basic Ball Lock
Order No. M00059
$29.00

B94.29.1-1977(R1995)
Die Buttons—Basic Ball-Lock (Metric)
Order No. M00088
$29.00

B94.30-1970(R1995)
Die Buttons—Variable, Press Fit
Order No. M00060
$29.00

B94.31-1981(R1995)
Steel Rotary Slitting Knives and Steel Spacing Collars
Order No. K00061
$29.00

B94.33-1996
Jig Bushings
Order No. K02196
$49.00

B94.35-1972(R1995)
Drill Drivers, Split-Sleeve, Collet Type
Order No. J00030
$29.00

B94.37M-1979(R1985)
Roller Turner Type Cutting Tools Single Point
Order No. M00081
$29.00

B94.38-1972(R1995)
Punches—Variable, Angle Head Type and Related Quill Bushings
Order No. M00068
$29.00

B94.39-1972(R1995)
Punches—Basic, Combination Angle Head Type and Related Quill Bushings
Order No. M00069
$29.00

B94.40-1972(R1995)
Punches—Wire Type
Order No. M00070
$29.00

B94.41-1972(R1995)
Punches—Basic, Angle Head Type and Related Quill Bushings
Order No. M00071
$29.00

B94.43-1972(R1995)
Die Buttons—Variable, Press Fit, Headless and Head Type, Step Relief
Order No. M00075
$29.00

TOOLS
cont'd

B94.44-1972(R1995)
Punches—Basic, Cylindrical Head Type and Related Quill Bushings
Order No. M00076
$29.00

B94.49-1975(R1995)
Spade Drill Blades and Spade Drill Holders
Order No. L00035
$29.00

B94.50-1975(R1995)
Basic Nomenclature and Definitions for Single-Point Cutting Tools
Order No. M00084
$29.00

B94.51M-1994
Specifications for Band Saw Blades (Metal Cutting)
Order No. M08594
$28.00

B94.52M-1994
Specifications for Hacksaw Blades
Order No. M09194
$29.00

B94.53-1978(R1984)
Solid Steel Rectangular Metal Cutting Squaring Shear Knives-Dimensional Tolerances
Order No. M00096
$29.00

B94.54-1994
Specifications for Hole Saws, Hole Saw Arbors, and Hole Saw Accessories
Order No. M13594
$30.00

B94.55M-1985(R1995)
Tool Life Testing with Single-Point Turning Tools
Order No. M00137
$32.00

B94.56-1995
Gages—Functional, Ball-Lock Punches, Inch
Order No. M15595
$30.00

B94.56.1M-1995
Gages—Functional, Ball-Lock Punches, Metric
Order No. M15395
$30.00

Hand Tools
B107.1-1993
Socket Wrenches, Hand (Inch Series)
Order No. N08893
$32.00

B107.2-1995
Socket Wrenches, Extensions, Adaptors, and Universal Joints, Power Drive (Impact) (Inch Series)
Order No. N01395
$32.00

B107.4M-1995
Driving and Spindle Ends for Portable Hand, Impact, Air, and Electric Tools (Percussion Tools Excluded)
Order No. N02395
$36.00

B107.5M-1994
Socket Wrenches, Hand (Metric Series)
Order No. N08294
$32.00

B107.6-1994
Wrenches, Box, Angled, Open End, Combination, Flare Nut, and Tappet (Inch Series)
Order No. N08494
$29.00

B107.8M-1996
Adjustable Wrenches
Order No. N08396
$29.00

B107.9M-1994
Wrenches, Box, Angled, Open End, Combination, Flare Nut, and Tappet (Metric Series)
Order No. N08794
$32.00

B107.10M-1996
Handles and Attachments for Hand Socket Wrenches—Inch and Metric Series
Order No. N10096
$29.00

B107.11M-1993
Pliers, Diagonal Cutting, and Nippers, End Cutting
Order No. N10493
$32.00

B107.12-1997
Nut Drivers (Spin Type, Screwdriver Grip) (Inch Series)
Order No. N10797
$29.00

B107.13M-1996
Pliers—Long Nose, Long Reach
Order No. N11196
$29.00

B107.14M-1994
Hand Torque Tools
Order No. N10994
$29.00

B107.15-1993
Flat Tip and Phillips Screwdrivers
Order No. N11693
$32.00

B107.16M-1998
Shears (Metal Cutting, Hand)
Order No. N11498
$32.00

B107.17M-1997
Gages, Wrench Openings, Reference
Order No. N11297
$32.00

B107.18M-1996
Pliers (Wire Twister)
Order No. N11396
$29.00

B107.19-1993
Pliers, Retaining Ring
Order No. N11893
$32.00

B107.20M-1992
Pliers (Lineman's, Iron Worker's, Gas, Glass, and Fence)
Order No. N11792
$29.00

B107.21-1992
Wrench, Crowfoot Attachments
Order No. N12591
$29.00

B107.22M-1991
Electronic Cutters and Pliers
Order No. N12391
$29.00

B107.23M-1997
Pliers, Multiple Position, Adjustable
Order No. N12797
$32.00

B107.25M-1996
Pliers—Performance Test Methods
Order No. N13596
$29.00

B107.27-1996
Pliers, Multiple Position (Electrical Connector)
Order No. N13696
$32.00

B107.28M-1997
Electronic Torque Instruments
Order No. N14397
$32.00

B107.29M-1998
Electronic Tester, Hand Torque Tools
Order No. N14698
$32.00

TOOLS
cont'd

B107.31M-1997
Screwdrivers, Cross Tip Gaging
Order No. N13997
$35.00

B107.34M-1997
Socket Wrenches for Spark Plugs
Order No. N14297
$32.00

B107.35M-1997
Nut Drivers (Spin Type Screwdriver Grip) (Metric Series)
Order No. N14097
$32.00

B107.41M-1997
Nail Hammers—Safety Requirements
Order No. N13797
$29.00

B107.42M-1997
Hatchets: Safety Requirements
Order No. N14197
$28.00

B107.43M-1998
Wood-Slipping Wedges: Safety Requirements
Order No. N14497
$32.00

B107.44M-1998
Glaziers' Chisels and Wood Chisels—Safety Requirements
Order No. N15198
$32.00

B107.45M-1998
Ripping Chisels and Flooring/Electricians' Chisels: Safety Requirements
Order No. N14598
$32.00

B107.46M-1998
Stud, Screw, and Pipe Extractors: Safety Requirements
Order No. N14798
$32.00

B107.52M-1998
Nail-Puller Bars: Safety Requirements
Order No. N15298
$28.00

B107.53M-1998
Ball Peen Hammers: Safety Requirements
Order No. N15398
$28.00

B107.55M-1998
Axes: Safety Requirements
Order No. N15098
$28.00

TURBINES
Standards that cover considerations for procurement of turbines.

B133.1M-1983(R1997)
Gas Turbine Terminology
Order No. K00093
$29.00

B133.2-1977(R1997)
Basic Gas Turbines
Order No. K00090
$29.00

B133.3-1981(R1994)
Procurement Standard for Gas Turbine Auxiliary Equipment
Order No. K00107
$29.00

B133.4-1978(R1997)
Gas Turbine Control and Protection Systems
Order No. K00091
$29.00

B133.5-1978(R1997)
Procurement Standard for Gas Turbine Electrical Equipment
Order No. K00096
$29.00

B133.6-1978(R1994)
Procurement Standard for Gas Turbine Ratings and Performance
Order No. K00098
$29.00

B133.7M-1985(R1992)
Gas Turbine Fuels
Order No. K00086
$29.00

B133.8-1977(R1989)
Gas Turbine Installation Sound Emissions
Order No. K00087
$29.00

B133.9-1994
Measurement of Exhaust Emissions from Stationary Gas Turbine Engines
Order No. K09994
$36.00

B133.10-1981(R1994)
Procurement Standard for Gas Turbine Information to be Supplied by User and Manufacturer
Order No. K00109
$29.00

B133.11-1982(R1994)
Procurement Standard for Gas Turbine Preparation for Shipping and Installation
Order No. K00108
$29.00

B133.12-1981(R1994)
Procurement Standard for Gas Turbine Maintenance and Safety
Order No. K00106
$29.00

B133.16-1978(R1997)
Procurement Standard for Gas Turbine Marine Applications
Order No. K00095
$29.00

VALVES, FITTINGS, FLANGES, GASKETS
Standards for pressure-temperature ratings, materials, dimensions, tolerances, markings, and testing of valves, fittings, flanges and gaskets.

B16.1-1989
Cast Iron Pipe Flanges and Flanged Fittings
A I 1994
Order No. J01789
$48.00

B16.3-1992
Malleable Iron Threaded Fittings
A I 1997
Order No. J10392
$52.00

B16.4-1992
Gray Iron Threaded Fittings
A I 1997
Order No. J02292
$45.00

B16.5-1996
Pipe Flanges and Flanged Fittings
A I 2001
Order No. M01596
$95.00

VALVES, FITTINGS, FLANGES, GASKETS
Standards for pressure-temperature ratings, materials, dimensions, tolerances, markings, and testing of valves, fittings, flanges, and gaskets.

B16.9-1993
Factory-Made Wrought Steel Buttwelding Fittings
A I 1998
Order No. J10993
$55.00

B16.9-1986
Factory-Made Wrought Steel Buttwelding Fittings
A I 1991
Order No. J10986
$40.00

B16.10-1992
Face-to-Face and End-to-End Dimensions of Valves
A I 1997
Order No. J03192
$52.00

B16.11-1996
Forged Steel Fittings, Socket-Welding and Threaded
A I 2001
Order No. M01696
$39.00

B16.12-1991
Cast Iron Threaded Drainage Fittings
A I 1996
Order No. J02191
$32.00

B16.14-1991
Ferrous Pipe Plugs, Bushings and Locknuts with Pipe Threads
A I 1996
Order No. J03290
$32.00

B16.15-1985(R1994)
Cast Bronze Threaded Fittings
A I 1990
Order No. J01085
$40.00

B16.18-1984(R1994)
Cast Copper Alloy Solder Joint Pressure Fittings
Order No. J07184
$34.00

B16.20-1993
Metallic Gaskets for Pipe Flanges—Ring Joint, Spiral-Wound, and Jacketed
Order No. L03093
$41.00

B16.21-1992
Nonmetallic Flat Gaskets for Pipe Flanges
Order No. L01592
$35.00

B16.22-1995
Wrought Copper and Copper Alloy Solder Joint Pressure Fittings
A I 2000
Order No. M03095
$32.00

B16.23-1992
Cast Copper Alloy Solder Joint Drainage Fittings (DWV)
A I 1995
Order No. J13492
$49.00

B16.24-1991(R1998)
Cast Copper Alloy Pipe Flanges and Flanged Fittings: Class 150, 300, 400, 600, 900, 1500 and 2500
A I 1996
Order No. J00991
$35.00

B16.25-1997
Buttwelding Ends
A I 2002
Order No. J01397
$42.00

B16.26-1988
Cast Copper Alloy Fittings for Flared Copper Tubes
A I 1993
Order No. J00888
$32.00

B16.28-1994
Wrought Steel Buttwelding Short Radius Elbows and Returns
A I 1999
Order No. J11094
$34.00

B16.28-1986
Wrought Steel Buttwelding Short Radius Elbows and Returns
A I 1991
Order No. J11086
$29.00

B16.29-1994
Wrought Copper and Wrought Copper Alloy Solder Joint Drainage Fittings—DWV
A I 1999
Order No. N05794
$30.00

B16.33-1990
Manually Operated Metallic Gas Valves for Use in Gas Piping Systems up to 125 psig (Sizes 1/2 Through 2)
A I 1995
Order No. J00690
$32.00

B16.34-1996
Valves—Flanged, Threaded, and Welding End
A I 2001
Order No. J01896
$79.00

B16.36-1996
Orifice Flanges
A I 2001
Order No. J04696
$42.00

B16.38-1985(R1994)
Large Metallic Valves for Gas Distribution (Manually Operated, NPS-2 1/2 to 12, 125 psig Maximum)
A I 1988
Order No. J05585
$32.00

B16.39-1986(R1994)
Malleable Iron Threaded Pipe Unions
A I 1991
Order No. L04986
$34.00

B16.40-1985(R1994)
Manually Operated Thermoplastic Gas Shutoffs and Valves in Gas Distribution Systems
A I 1988
Order No. J05485
$32.00

B16.42-1987(R1997)
Ductile Iron Pipe Flanges and Flanged Fittings, Classes 150 and 300
A I 1992
Order No. J06687
$40.00

B16.44-1995
Manually Operated Metallic Gas Valves for Use in House Piping Systems
A I 2000
Order No. J14095
$35.00

B16.45-1987(R1997)
Cast Iron Fittings for Sovent Drainage Systems
A I 1992
Order No. J11187
$32.00

B16.47-1996
Large Diameter Steel Flanges: NPS 26 Through NPS 60
A I 2001
Order No. J12996
$52.00

B16.48-1997
Steel Line Blanks
A I 2002
Order No. J14497
$42.00

1998 ASME Boiler and Pressure Vessel Code: An International Code

The Boiler and Pressure Vessel Code establishes rules of safety governing the design, fabrication and inspection of boilers and pressure vessels, and nuclear power plant components during construction. The objective of the rules is to ensure reasonably certain protection of life and property and to provide a margin for deterioration in service. Advancements in design and material and the evidence of experience are constantly being added by addenda. Prices include update service through 2000.

3-year update service includes:

- Addenda issued once per year: July 1, 1999 and 2000.
- Supplements issued four times per year for each Code Cases book.
- Interpretations issued twice per year: with Addenda, in July 1998 and in December 1998, 1999, and 2000.

Note: Interpretations will not be available separately and are issued as part of the update service to the individual sections of the Code. Interpretations to Section III, Divisions 1 and 2 are issued as part of the update to Subsection NCA.

I Power Boilers
Order No. QX0010 *$210.00*

II Materials
Part A—Ferrous Material Specifications
Order No. QX002A *$385.00*
Part B—Nonferrous Material Specifications
Order No. QX002B *$350.00*
Part C—Specifications for Welding Rods, Electrodes, and Filler Metals
Order No. QX002C *$350.00*
Part D—Properties
Order No. QX002D *$315.00*

III Rules for Construction of Nuclear Power Plant Components
Subsection NCA
General Requirements for Division 1 and Division 2
Order No. QX003R *$240.00*
Division 1
Subsection NB
Class 1 Components
Order No. QX003B *$240.00*
Subsection NC
Class 2 Components
Order No. QX003C *$240.00*
Subsection ND
Class 3 Components
Order No. QX003D *$240.00*
Subsection NE
Class MC Components
Order No. QX003E *$240.00*
Subsection NF
Supports
Order No. QX003F *$185.00*
Subsection NG
Core Support Structures
Order No. QX003G *$185.00*
Subsection NH
Class 1 Components in Elevated Temperature Service
Order No. QX003H *$240.00*
Appendices
Order No. QX003A *$250.00*
Division 2
Code for Concrete Reactor Vessels and Containments
Order No. QX0032 *$295.00*
Division 3
Containment Systems and Transport Packagings for Spent Nuclear Fuel and High Level Radioactive Waste
Order No. QX0033 *$275.00*

IV Heating Boilers
Order No. QX0040 *$195.00*

V Nondestructive Examination
Order No. QX0050 *$215.00*

VI Recommended Rules for the Care and Operation of Heating Boilers
Order No. QX0060 *$125.00*

VII Recommended Guidelines for the Care of Power Boilers
Order No. QX0070 *$145.00*

VIII Pressure Vessels
Division 1
Order No. QX0081 *$385.00*
Division 2
Alternative Rules
Order No. QX0082 *$385.00*
Division 3
Alternative Rules for Construction of High Pressure Vessels
Order No. QX0083 *$295.00*

IX Welding and Brazing Qualifications
Order No. QX0090 *$215.00*

X Fiber-Reinforced Plastic Pressure Vessels
Order No. QX0100 *$185.00*

XI Rules for Inservice Inspection of Nuclear Power Plant Components
Order No. QX0011 *$370.00*

Code Cases: Boilers and Pressure Vessels
Order No. QX0120 *$275.00*

Code Cases: Nuclear Components
Order No. QX012N *$315.00*

Binder
Order No. QX0140 *$20.00*

Special Price for Complete Code
Order No. QX0230
$6,550.00

Complete Code Plus 30 Binders
Order No. QX0330
$7,150.00

To order: Call toll free: 800-843-2763. Fax: 973-882-1717. Write: ASME, 22 Law Drive, Box 2300, Fairfield, NJ 07007-2300. E-mail: infocentral@asme.org.

All orders require a valid purchase order. When ordering with your company purchase order, please make sure that your ASME account number, if available, is indicated. This account number can be found on ASME invoices, packing lists, and statements.

Prices subject to change without notice. Shipping/handling charges added to every order.

Accreditation Packages

Code books required for use with ASME Code Symbol Stamps

Note: For books other than the Boiler and Pressure Vessel Code (B31.1, PTC 25.3, NQA-1), the required edition as of July 1, 1998 is listed. The specific effective Addenda will be referenced in the applicable Boiler and Pressure Vessel Code section. Later editions of these referenced books will be required if and when referenced by the applicable Boiler and Pressure Vessel Code section.

CODE SYMBOL STAMPS  

PP PRESSURE PIPING
S POWER BOILERS

CODE BOOKS REQUIRED

SECTION I—POWER BOILERS
SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties

SECTION V—NONDESTRUCTIVE EXAMINATION
SECTION IX—WELDING & BRAZING QUALIFICATIONS
B31.1-1995—POWER PIPING

***ORDER NO. BPVCA**
$2,204.00

CODE SYMBOL STAMP 

E ELECTRIC BOILERS

CODE BOOKS REQUIRED

SECTION I—POWER BOILERS
SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part D Properties

B31.1-1995—POWER PIPING

***ORDER NO. BPVCE**
$1,429.00

CODE SYMBOL STAMP 

U2 PRESSURE VESSELS (DIV. 2)

CODE BOOKS REQUIRED

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties

SECTION V—NONDESTRUCTIVE EXAMINATION
SECTION VIII—PRESSURE VESSELS, DIVISION 2-ALTERNATIVE RULES
SECTION IX—WELDING & BRAZING QUALIFICATIONS

ORDER NO. BPVU1
$2,215.00

CODE SYMBOL STAMP 

U3 PRESSURE VESSELS (DIV. 3)

CODE BOOKS REQUIRED

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties

SECTION V—NONDESTRUCTIVE EXAMINATION
SECTION VIII—PRESSURE VESSELS, DIVISION 3-ALTERNATIVE RULES FOR CONSTRUCTION OF HIGH PRESSURE VESSELS
SECTION IX—WELDING & BRAZING QUALIFICATIONS

ORDER NO. BPVU3
$2,125.00

CODE SYMBOL STAMPS  

H HEATING BOILERS EXCEPT CAST IRON
HLW LINED POTABLE WATER HEATERS

CODE BOOKS REQUIRED

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals

SECTION IV—HEATING BOILERS
SECTION IX—WELDING & BRAZING QUALIFICATIONS

ORDER NO. BPHLW
$1,495.00

* Packages that include the 1998 edition of B31.1.

CODE SYMBOL STAMP 

A ASSEMBLY OF POWER BOILERS

CODE BOOKS REQUIRED

SECTION I—POWER BOILERS
SECTION II—MATERIALS
Part C Specifications for Welding Rods, Electrodes and Filler Metals
SECTION V—NONDESTRUCTIVE EXAMINATION
SECTION IX—WELDING & BRAZING QUALIFICATIONS
B31.1-1995—POWER PIPING

***ORDER NO. BPVCB**
$1,154.00

CODE SYMBOL STAMPS  

U PRESSURE VESSELS (DIV. 1)
UM MINIATURE PRESSURE VESSELS

CODE BOOKS REQUIRED

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties
SECTION V—NONDESTRUCTIVE EXAMINATION
SECTION VIII—PRESSURE VESSELS, DIVISION 1
SECTION IX—WELDING & BRAZING QUALIFICATIONS

ORDER NO. BPVUM
$2,215.00

CODE SYMBOL STAMP 

RP FIBER-REINFORCED PLASTIC VESSELS

CODE BOOKS REQUIRED

SECTION X—FIBER-REINFORCED PLASTIC VESSELS

ORDER NO. BPVRP
$185.00

CODE SYMBOL STAMP 

H CAST IRON SECTIONAL HEATING BOILERS

CODE BOOK REQUIRED

SECTION IV—HEATING BOILERS

ORDER NO. BPVCH
$195.00

CODE SYMBOL STAMP 

H FIELD ASSEMBLY OF BOILERS

CODE BOOKS REQUIRED

SECTION IV—HEATING BOILERS
SECTION IX—WELDING & BRAZING QUALIFICATIONS

ORDER NO. BPVCH
$410.00

CODE SYMBOL STAMP 

UD RUPTURE DISK DEVICES

CODE BOOKS REQUIRED†

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties
SECTION VIII—PRESSURE VESSELS DIVISION 1
SECTION IX—WELDING & BRAZING QUALIFICATIONS
PTC 25-1994—PRESSURE RELIEF DEVICES

ORDER NO. BPUD
$2,049.00

CODE SYMBOL STAMP 

UV PRESSURE VESSELS SAFETY VALVES

CODE BOOKS REQUIRED†

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties
SECTION VIII—PRESSURE VESSELS DIVISION 1 OR DIVISION 2
SECTION IX—WELDING & BRAZING QUALIFICATIONS
PTC 25-1994—PRESSURE RELIEF DEVICES

ORDER NO. BPUV1
(INCLUDES SECTION VIII, DIV. 1)
$2,049.00
ORDER NO. BPUV2
(INCLUDES SECTION VIII, DIV. 2)
$2,049.00

* Sections II and IX are not required for assemblers. Section II, Part C and Section IX are not required for manufacturers if welding and brazing are not within the scope of their work.

CODE SYMBOL STAMP 

UV3 PRESSURE VESSELS SAFETY VALVES

CODE BOOKS REQUIRED†

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties

SECTION VIII—PRESSURE VESSELS DIVISION 3
SECTION IX—WELDING & BRAZING QUALIFICATIONS
PTC 25-1994—PRESSURE RELIEF DEVICES

ORDER NO. BPUV3
$1,959.00

CODE SYMBOL STAMP 

M MINIATURE BOILERS

CODE BOOKS REQUIRED

SECTION I—POWER BOILERS
SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties

SECTION IX—WELDING & BRAZING QUALIFICATIONS
B31.1-1995—POWER PIPING

*__ORDER NO. BPVCM__
$1,989.00

CODE SYMBOL STAMP 

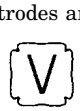
HV HEATING BOILER SAFETY VALVES

CODE BOOKS REQUIRED‡

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals

SECTION IV—HEATING BOILERS
SECTION IX—WELDING & BRAZING QUALIFICATIONS
PTC 25-1994—PRESSURE RELIEF DEVICES

ORDER NO. BPVHV
$1,544.00

CODE SYMBOL STAMP 

V BOILER SAFETY VALVES

CODE BOOKS REQUIRED†

SECTION I—POWER BOILERS
SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties

SECTION IX—WELDING & BRAZING QUALIFICATIONS
PTC 25-1994—PRESSURE RELIEF DEVICES

ORDER NO. BPVCV
$1,874.00

QUALITY SYSTEM CERTIFICATES
MATERIAL MANUFACTURERS, MATERIALS SUPPLIERS

CODE BOOKS REQUIRED

SECTION III—Rules for Construction of Nuclear Power Plant Components
Subsection NCA-General Requirements for Division 1 and Division 2
Appendices

SECTION II—Materials (Select applicable parts)
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties

ORDER NO. BPVMS

Applicants should note that they may need one or more of the following subsections of Section III depending on the scope of their work to appear on the certificate. In addition, these subsections contain material requirements on Section 2000 which reference Section V and Section IX.

Subsection NB-Class 1 Components
Subsection NC-Class 2 Components
Subsection ND-Class 3 Components
Subsection NE-Class MC Components
Subsection NF-Supports
Subsection NG-Core Support Structures
Subsection NH-Class 1 Components in Elevated Temperature Service
SECTION V—Nondestructive Examination
SECTION IX—Welding and Brazing Qualifications

* Section II, Part C and Section IX are not required for manufacturers if welding and brazing are not within the scope of their work.

CODE SYMBOL STAMPS







N TYPE CERTIFICATES OF AUTHORIZATION AND CERTIFICATES OF ACCREDITATION N, NA, NPT

CODE BOOKS REQUIRED

SECTION II—MATERIALS
Part D Properties (N and NA only)
SECTION III—RULES FOR CONSTRUCTION OF NUCLEAR POWER PLANT COMPONENTS
SUBSECTION NCA-GENERAL REQUIREMENTS FOR DIVISION 1 AND DIVISION 2 (NOT REQUIRED FOR SECTION III-3)
APPENDICES
SECTION V—NONDESTRUCTIVE EXAMINATION
SECTION IX—WELDING & BRAZING QUALIFICATIONS
NQA-1-1989—QUALITY ASSURANCE PROGRAM REQUIREMENTS FOR NUCLEAR FACILITIES

ORDER NO. BPNPT

Applicants should note that they may need one or more of the following Parts of Section II and Subsections of Section III depending on the scope of their work to appear on the certificate.

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties (NPT only)

Subsection NB-Class 1 Components
Subsection NC-Class 2 Components
Subsection ND-Class 3 Components
Subsection NE-Class MC Components
Subsection NF-Supports
Subsection NG-Core Support Structures
Subsection NH-Class 1 Components in Elevated Temperature Service
Division 2: Code for Concrete Reactor Vessels and Containments
Division 3: Containment Systems and Transport Packaging for Spent Nuclear Fuel and High Level Radioactive Waste

CODE SYMBOL STAMP



NV NUCLEAR SAFETY AND PRESSURE RELIEF VALVES

CODE BOOKS REQUIRED

SECTION III—RULES FOR CONSTRUCTION OF NUCLEAR POWER PLANT COMPONENTS
SUBSECTION NCA-GENERAL REQUIREMENTS FOR DIVISION 1 AND DIVISION 2
APPENDICES
SECTION II—MATERIALS (SELECT APPLICABLE PARTS)
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part D Properties

SECTION V—NONDESTRUCTIVE EXAMINATION
SECTION IX—WELDING & BRAZING QUALIFICATIONS
PTC 25-1994—PRESSURE RELIEF DEVICES
NQA-1-1989—QUALITY ASSURANCE PROGRAM REQUIREMENTS FOR NUCLEAR FACILITIES

ORDER NO. BPNVA (WITH SECTION II, PART A)
ORDER NO. BPNVB (WITH SECTION II, PART B)
ORDER NO. BPNVD (WITH SECTION II, PART D)

Applicants should note that they may need one or more of the following subsections of Section III depending on the scope of their work to appear on the certificate.

Subsection NB-Class 1 Components
Subsection NC-Class 2 Components
Subsection ND-Class 3 Components
Subsection NH-Class 1 Components in Elevated Temperature Service

ASME 9000

ASME International's ISO 9000 Registration Program

The American Society of Mechanical Engineers has been a world leader in codes, standards, accreditation, and certification for over a century. These programs have now been extended to include the registration (certification) of quality systems in conformance with the standards set by the International Organization of Standardization (ISO).

ASME's ISO 9000 Registration Program is accredited by the American National Accreditation Program for Registrars of Quality Systems (ANSI-RAB) and the Dutch Council for Accreditation (RvA).

ASME specializes in the registration of domestic and international suppliers of mechanical equipment and related materials, items, and services in the industries and sectors associated with the art, science, and practice of mechanical engineering. These industries include:

- Primary metals industries
- Fabricated metal products
- Industrial and commercial machinery and equipment
- Reinforced thermoset plastic tanks and vessels
- Engineering services

ASME uses quality system auditors experienced in the above-listed fields who have been certified to the international auditor qualification standard ISO 10011-2. Registration decision and policy making are conducted within the framework of the ASME codes, standards, accreditation, and certification committee structure. The use of volunteers from industry and government provides for a high quality program that is administered in a fair, impartial, and professional manner.

Over 4,000 companies from 57 countries are currently accredited by ASME in one or more of the following quality assurance based programs:

- Authorized Inspection Agency (AIA)
- Boiler and Pressure Vessel
- Fastener Manufacturers and Distributors (FAP)
- Nuclear Components
- Nuclear Materials (MM & MS)
- Qualification of Elevator Inspectors (QEI)
- Reinforced Thermoset Plastic Corrosion Resistant Vessel Fabricators (RTP)
- Safety and Pollution Prevention Equipment (SPPE)

Attention all ASME Boiler and Pressure Vessel Certificate Holders! Reduce assessment fees and achieve consistency by combining your upcoming shop review with an ISO 9000 assessment. ASME will utilize Auditors who are both qualified as BPV Team Leaders and ISO 9000 Assessors. Capitalize on this unique opportunity now—contact ASME.

For additional information write to: ASME Conformity Assessment, Three Park Avenue, New York, NY 10016-5990, or call Raj Manchanda at 212-591-8033. Fax: 212-591-8599. E-mail: accreditation@asme.org.

ASME Certification Programs

ASME administers four certification programs for technical personnel based on ASME standards that define the necessary qualifications of the individuals. One additional program is in development.

Operators of Resource Recovery Facilities Processing Municipal Solid Waste (MWCs)

- Applies to persons who hold, or wish to hold, the positions of Shift Supervisor or Chief Facility Operator
- Candidates are required to meet minimum education and experience requirements and pass a written multiple-choice examination for provisional certification
- Full certification is obtained through a site-specific oral examination

Operators of Medical Waste Incinerators (MWIs)

- Applies to persons who hold, or wish to hold, the positions of Operator or Operator Supervisor
- Candidates are required to demonstrate proficiency in the operation of the equipment and pass a written multiple-choice examination
- "Special Class" operator certification also available for persons in related fields (regulatory personnel, trainers, equipment manufacturers, etc.)

Operators of Hazardous Waste Incinerators

- Applies to Control Room Operators, Field Operators (waste receiving and handling, field equipment operation, and residual handling), and Supervisors
- Candidates are required to complete training in safety, combustion fundamentals, and basic operation, and pass a written multiple-choice examination for provisional certification
- Full certification is obtained through a site-specific oral examination in development

Geometric Dimensioning and Tolerancing Professionals Certification Exam

To certify your competence in using the Geometric Dimensioning and Tolerancing Y14.5 Standard, register now for the new certification exam sponsored by ASME. You can obtain certification on one of the following two levels:

- *Technologist GDTP*, which provides objective measure of your ability to understand drawings prepared using the Y14.5 Standard
- *Senior GDTP*, which provides additional objective measure of your ability to select geometric controls and apply them properly to drawings

Exams are given in various cities. Please contact ASME for the exam location nearest you. To obtain an exam application and full details: Phone: Raj Manchanda at 212-591-8033. Fax: 212-591-8599. E-mail: accreditation@asme.org. Or visit the Codes & Standards area of the ASME Web site: http://www.asme.org.

- Candidates will be required to pass a written multiple-choice examination based on the ASME Y14.5 Standard

Operators of High Capacity Fossil Fuel Fired Plants (in development)

- Applies to operators of oil, gas, and coal fired boilers with heat input greater than 10,000,000 Btu/h
- Certification criteria will include evaluation of qualifications in both safety and air emissions Geometric Dimensioning and Tolerancing Professionals
- Applies to designers, drafters, manufacturing engineers, CAD/CAM users, and others who need to understand the meaning of symbols, modifiers, and relationships of GD&T as applied to engineering drawings and related documents

For examination schedules and additional information, contact: ASME Conformity Assessment, Three Park Avenue, New York, NY 10016-5990. Tel: 212-591-8591. Fax: 212-591-8599. E-mail: accreditation@asme.org.

ASME A17 CD-ROM for Elevators and Escalators

The ASME A17 CD-ROM for Elevators and Escalators features access to the current A17 collection from your desktop:

- ASME A17.1-1996 Safety Code for Elevators and Escalators
- ASME A17.2.1-1996 Inspectors' Manual for Electric Elevators
- ASME A17.2.2-1994 (through A17.2.2b-1996) Inspectors' Manual for Hydraulic Elevators
- ASME A17.2.3-1994 (through A17.2.3a-1996) Inspectors' Manual for Escalators and Moving Walks
- ASME A17.3-1996 Safety Code for Existing Elevators and Escalators
- Interpretations for A17.1, A17.2.2., and A17.3
- Graphic images of all equations, figures, and tables assure fidelity of all visual information.

Sophisticated search capacities, navigational ease, custom annotation, customer support

The A17 CD-ROM gives you the freedom to move through the Code books unconstrained by pages (for your convenience, page numbers that match those of the print publications are provided). The clean, clear visual design inherits its colors from your own Windows choices. The combination of browsing and focused searching capabilities allows you to thoroughly and quickly investigate the *ASME Elevator and Escalator Code*.

Full Test Browser delivers compact, easily accessible electronic information, built from the same data that created the printed ASME A17 series books. It's simple to move through the collection, scanning the table of contents at any level of detail—from book titles to chapter titles to designated paragraphs. View the full table of contents or list tables or figures in the Table of Contents view. Jump to selected items with a click. The full text of the publications can also be read start to finish, more like a traditional book.

Order No.	Description	Price
A14401	Stand Alone License	$495
A14405	Network License, 2–5 users	$1,100
A14410	Network License, 6–10 users	$1,500
(NA)	Network License, more than 10 users	*

*For more than 10 users, please call ASME Information Central (800-843-2763) for a price quote.

Prices include an automatic subscription to the next two updated A17 CD-ROM.

PC requirements: System requirements for installation on a stand-alone personal computer are: 386DX processor minimum, 486 or higher preferred; 8 MB RAM minimum, with 500 KB free conventional memory; hard drive with approximately 7 MB of free space for software installation and 1 MB of additional free space for temporary files during program operation; mouse; VGA or SVGA monitor (SVGA preferred) with compatible video card (high-performance card preferred); CD-ROM drive compatible with Microsoft Compact Disc Extensions (MSCDEX) and ISO 9660; Microsoft Compact Disc Extensions (MSCDEX), version 2.2 or higher; DOS version 5.0 or higher; Microsoft Windows, 3.1 or higher, running in 366 Enhanced mode; Windows 95.

The Boiler and Pressure Vessel Code on CD-ROM†

The *Code* on CD-ROM provides pinpoint searching accuracy. The power of the computer allows you to search the entire *Code* for the occurrence of a single word or a complex combination of terms. You may browse the database by Title or Table of Contents, or by a specially developed thesaurus of subject classifications.

Electronic delivery allows you to quickly and easily investigate the *Code*. An especially convenient and time-saving feature called Hot Link allows you to skip from a particular point in the *Code* to referenced sections, tables, graphs, or equations and then back to the original text.

The Worldwide Standards Service—ASME from IHS

Make your research time more productive by having ASME documents available at your desk whenever you need them. With the American Society of Mechanical Engineers (ASME) documents available on CD-ROM you can view complete, full-text documents—exact images of the hardcopy originals—right at your own personal computer. The service contains the full text of all ASME Standards and Performance Test Codes. Included in every subscription is the comprehensive Worldwide Standards Service Index, which helps you locate standards by document number or by keyword. In addition to ASME documents, the Worldwide Standards Service Index contains document listings from approximately 400 of the world's leading standards developing bodies. The service is updated every 60 days to keep you abreast of all current revision activity.

For more information on the Boiler Code on CD-ROM or Worldwide Standards Service and free demo diskettes, call toll-free: 800-241-7824.

Information Handling Services®
15 Inverness Way East, P.O. Box 1154
Englewood, Colorado USA 80150
303-790-0600, ext. 59

† For the purpose of ASME accreditation, electronic versions of the Boiler and Pressure Vessel Code do not replace the need for a single hard copy version.

Electronic Stress Tables

Software based on ASME Boiler and Pressure Vessel Code Section II, Part D

ASME has good news for engineers who use the traditional methods of searching through stress tables. The new Electronic Stress Tables based on the *ASME Boiler and Pressure Vessel Code Section II, Part D (Materials Properties)* makes your search easier and faster. Designed to give you quick and accurate access to any or all BPVC Section II, Part D stress tables, the program provides a time-saving alternative to wading through eye-straining pages of spreadsheets. You'll find the software not only speeds your search, but also presents the information in a highly visual format, complete with notes and printing capability.

Search options

Fifteen different fields allow you to narrow your search criteria (filter), or reorder the data displayed (sort). The software allows you to create multiple line criteria for logical groupings and to add additional fields to create a compound sort.

Create Excel spreadsheets

Import any or all of the tables into an Excel spreadsheet in a numeric format. You must have a copy of Excel installed on your PC to run this addition to the application.

Highly compatible

To get started, all you need is an IBM or 100% compatible CPU running Windows 3.1, Windows NT or Windows 95, 12 megabytes of disk space, 4 megabytes of RAM, SVGA display, and 3.5″ floppy drive. Order No. DBOIID $265

RA-search On-Line Service

RA-search is a computer-based text retrieval program developed by Reedy Engineering. It contains full text searching and viewing capabilities of the more than 6,000 ASME Interpretations published that relate to the Codes and Standards listed below.

RA-search's database includes all of the published Interpretations to the following ASME Codes and Standards:

- All published Boiler and Pressure Vessel Code Interpretations (Sections I, II, III-1 and 2, IV, V, VI, VII, VIII-1 and 2, IX, X and XI)
- All published NQA-1 and NQA-2 Interpretations
- All published B16 (B16.3, B16.5, B16.15 and B16.34) and B31 (B31.1, B31.3, B31.4, B31.5, B31.8 and B31.9) Interpretations
- A39, A90, A112, B1, B2, B15, B18, B20, B30, B36, B56, B73, B94, CSD, MH14, N45.2, N509, N510, PTC's

RA-search costs $500 per year plus $3.00 per minute of on-line access. The annual rate is invoiced at the start of your subscription and the access charges are billed monthly. To start your subscription to RA-search Interpretations, or if you have any questions, contact: Rick Swayne, Reedy Engineering, 3425 South Bascomb Avenue, Campbell, CA 95008; or call 408-558-0330.

ACCREDITATION AND CERTIFICATION

Board on Accreditation, Registration, and Certification (BARC)
Alan Bagner (212) 591-8580

Boiler and Pressure Vessel Accreditation
Ronald Berberich (212) 591-8461

AIA Accreditation
Bibi Rahim (212) 591-8465

Fastener Accreditation
Bibi Rahim (212) 591-8465

ISO 9000 Registration
Christine Bujal (212) 591-8592

Nuclear Accreditation
Bibi Rahim (212) 591-8465

QAI Authorized Inspection
Ken Baron (212) 591-7019

QEI Accreditation (Elevator Inspectors)
Bibi Rahim (212) 591-8465

QHO Certification (Hazardous Waste Incinerator)
Sandra Bridgers (212) 591-8465

QMO Certification (Medical Waste Incinerator Operators)
Sandra Bridgers (212) 591-8465

QRO Certification (Resource Recovery Facility Operators)
Sandra Bridgers (212) 591-8465

RTP Accreditation (Reinforced Thermoset Plastic Corrosion Resistant Equipment)
Bibi Rahim (212) 591-8465

SPPE Accreditation (Offshore Equipment)
Bibi Rahim (212) 591-8465

NUCLEAR CODES AND STANDARDS

Board on Nuclear Codes and Standards (BNCS)
Gerry Eisenberg (212) 591-8510

Joint ASME/NRC Symposium on Valve and Pump Testing
Gerry Eisenberg (212) 591-8510

BPV Subcommittee on Nuclear Power (III)
Chris Sanna (212) 591-8513

BPV Joint ACI/ASME Committee on Concrete Components for Nuclear Service (III-2)
Chris Sanna (212) 591-8513

BPV Subcommittee on Nuclear Inservice Inspection (XI)
Joseph Saltarelli (212) 591-7005

Interpretations—Section III
Joseph Saltarelli (212) 591-7005

CONAGT Main Committee on Nuclear Air and Gas Treatment Equipment
Joseph Saltarelli (212) 591-7005

Committee on Cranes for Nuclear Facilities
Jess Moon (212) 591-8514

NQA Committee on Nuclear Quality Assurance
Chris Sanna (212) 591-8513

O&M Committee on Operation and Maintenance of Nuclear Power Plants
Jess Moon (212) 591-8514

QME Committee on Qualification of Mechanical Equipment Used in Nuclear Power Plants
Gerry Eisenberg (212) 591-8510

Committee on Nuclear Risk Management
Gerry Eisenberg (212) 591-8510

PERFORMANCE TEST CODES

Board on Performance Test Codes (BPTC)
William Hays (212) 591-8550

PTC Performance Monitoring
George Osolsobe (212) 591-8554

PTC 1 General Instructions
Jack H. Karian (212) 591-8552

PTC 2 Definitions and Values
George Osolsobe (212) 591-8554

PTC 3 Fuels
Jack H. Karian (212) 591-8552

PTC 4.1 Steam Generating Apparatus
Jack H. Karian (212) 591-8552

PTC 4.2 Coal Pulverizers
Jack H. Karian (212) 591-8552

PTC 4.3 Air Heaters
Jack H. Karian (212) 591-8552

PTC 4.4 Gas Turbine Heat Recovery Steam Generators
Jack H. Karian (212) 591-8552

PTC 5 Reciprocating Steam Driven Engines
Jack H. Karian (212) 591-8552

PTC 6 Steam Turbines
Jack H. Karian (212) 591-8552

PTC 7 Reciprocating Steam Driven Displacement Pumps
George Osolsobe (212) 591-8554

PTC 7.1 Displacement Pumps
George Osolsobe (212) 591-8554

PTC 8.2 Centrifugal Pumps
George Osolsobe (212) 591-8554

PTC 9 Displacement Compressors
Jack H. Karian (212) 591-8552

PTC 10 Compressors and Exhausters
Jack H. Karian (212) 591-8552

PTC 11 Fans
Jack H. Karian (212) 591-8552

PTC 12.1 Feedwater Heaters
George Osolsobe (212) 591-8554

PTC 12.2 Surface Steam Condensers
Jack H. Karian (212) 591-8552

PTC 12.3 Deaerators
Jack H. Karian (212) 591-8552

PTC 12.4 Moisture Separator Reheaters
Jack H. Karian (212) 591-8552

PTC 12.5 Single Phase Heat Exchangers
George Osolsobe (212) 591-8554

PTC 14 Evaporating Apparatus
Jack H. Karian (212) 591-8552

PTC 16 Gas Producers & Continuous Gas Generators
Jack H. Karian (212) 591-8552

PTC 17 Reciprocating Internal Combustion Engines
William Hays (212) 591-8550

PTC 18 Hydraulic Prime Movers
George Osolsobe (212) 591-8554

PTC 19.1 Measurement Uncertainty
George Osolsobe (212) 591-8554

PTC 19.2 Pressure Measurement
George Osolsobe (212) 591-8554

PTC 19.3 Temperature Measurement
Oliver Martinez (212) 591-8553

PTC 19.5 Application, Part II of Fluid Meters
Oliver Martinez (212) 591-8553

PTC 19.22 Digital Systems Techniques
Oliver Martinez (212) 591-8553

PTC 20.1 Speed & Load Governing Systems for Steam Generator Units
George Osolsobe (212) 591-8554

PTC 20.2 Overspeed Trip Systems for Steam Generator Units
George Osolsobe (212) 591-8554

PTC 20.3 Pressure Control Systems Used on Steam Turbine Generator Units
George Osolsobe (212) 591-8554

PTC 21 Particulate Matter Collection Equipment
William Hays (212) 591-8550

PTC 22 Gas Turbines
George Osolsobe (212) 591-8554

PTC 23 Atmospheric Water Cooling Equipment
Jack H. Karian (212) 591-8552

PTC 23.1 Spray Cooling Systems
Jack H. Karian (212) 591-8552

PTC 24 Ejectors
William Hays (212) 591-8550

PTC 25 Pressure Relief Devices
Umberto D'Urso (212) 591-8553

PTC 26 Speed Governing Systems for Internal Combustion Engine Generator Units
William Hays (212) 591-8550

PTC 28 Determining the Properties of Fine Particulate Matter
William Hays (212) 591-8550

PTC 29 Speed Governing Systems for Hydraulic Turbine-Generator Units
Oliver Martinez (212) 591-8553

PERFORMANCE TEST CODES (cont'd)

PTC 30 Air Cooled Heat Exchangers
Jack H. Karian (212) 591-8552

PTC 31 Ion Exchange Equipment
William Hays (212) 591-8550

PTC 32.1 Nuclear Steam Systems
William Hays (212) 591-8550

PTC 33 Large Incinerators
Jack H. Karian (212) 591-8552

PTC 34 Waste Combustors with Energy Recovery
Jack H. Karian (212) 591-8552

PTC 36 Measurement of Industrial Sound
George Osolsobe (212) 591-8554

PTC 38 Determining the Concentration of Particulate Matter in a Gas Steam
William Hays (212) 591-8550

PTC 39 Steam Traps
Oliver Martinez (212) 591-8553

PTC 40 Flue Gas Desulfurization Units
William Hays (212) 591-8550

PTC 42 Wind Turbines
William Hays (212) 591-8550

PTC 45 Emission Testing of Municipal Waste Combustors
Oliver Martinez (212) 591-8553

PTC 46 Overall Plant Performance Test
Jack H. Karian (212) 591-8552

PTC 47 Integrated Gasification Combined Cycle
Jack H. Karian (212) 591-8552

PTC 50 Fuel Cell Power Systems
Jack H. Karian (212) 591-8552

PRESSURE TECHNOLOGY CODES AND STANDARDS

Board on Pressure Technology Codes & Standards (BPTCS)
Mark Sheehan (212) 591-8530

BPE Bioprocessing Equipment Committee
Paul Stumpf (212) 591-8536

PCI Post Construction Issues Committee
June Yarmush (212) 591-8539

Boiler and Pressure Vessel Code Committee (Exec. & MC)
Joe Brzuszkiewicz (212) 591-8533

BPV SC on Power Boilers (SC I) (Incl. Sect. VII)
Paul Stumpf (212) 591-8536

BPV SC on Materials (SC II) (Parts A, B & D)
Faron Lyons (212) 591-8540

BPV SC on Transport Tanks (SC XIII)
Paul Stumpf (212) 591-8536

BPV SC on Heating Boilers (SC IV) (Incl. Sect. VI)
Brian Feder (212) 591-8534

BPV SC on Nondestructive Examination (SC V)
F. Shawn Fitzgerald (212) 591-8056

BPV SC on Safety Valve Requirements (SC SVR)
Umberto D'Urso (212) 591-8535

BPV SC on Pressure Vessels (SC VIII)
Alan Roby (212) 591-8538

BPV SC on Welding (SC IX) (Incl. Sect. II, Part C)
Jon Labrador (212) 591-8532

BPV SC on Fiber-Reinforced Plastic Pressure Vessels (SC X)
Alan Roby (212) 591-8538

B16 Main Committee—Valves, Flanges, Fittings and Gaskets
Patricia Reddington (212) 591-8537

B31 Code for Pressure Piping (Main Committee)
June Yarmush (212) 591-8539

B31.1 Power Piping Committee
Paul Stumpf (212) 591-8536

B31.3 Chemical Plant and Petroleum Piping Committee
Jon Labrador (212) 591-8532

B31.4/11 Liquid and Slurry Piping Transportation Systems Committee
Brian Feder (212) 591-8534

B31.5 Refrigeration Piping Committee
Faron Lyons (212) 591-8540

B31.8 Gas Transmission and Distribution Piping Systems
F. Shawn Fitzgerald (212) 591-8056

B31.9 Building Services Piping Section Committee
Faron Lyons (212) 591-8540

B96 Welded Aluminum Alloy Storage Tanks Committee
Jon Labrador (212) 591-8532

HPS High Pressure Systems Committee
Umberto D'Urso (212) 591-8535

PVHO Pressure Vessels for Human Occupancy Committee
Patricia Reddington (212) 591-8537

RTP Committee on Reinforced Thermoset Plastic Corrosion Resistant Equipment
Alan Roby (212) 591-8538

TDP Committee on Turbine Water Damage
Alan Roby (212) 591-8538

SAFETY CODES AND STANDARDS

Board on Safety Codes and Standards (BSCS)
Marcy Weinstock (212) 591-8526

Advisory Committee on Environmental Management (ACEM)
Geraldine Burdeshaw (212) 591-8523

A13 Committee on Scheme for theIdentification of Piping Systems
Melissa Aranzamendez (212) 591-8522

A17 Committee on Elevators and Escalators
Geraldine Burdeshaw (212) 591-8523

A18 Committee on Platform Lifts and Stairway Chairlifts
Melissa Aranzamendez (212) 591-8522

A90 Committee on Safety Standards for Manlifts
Joseph Wendler (212) 591-8524

A120 Committee on Safety Requirements for Powered Platforms for Building Maintenance
Melissa Aranzamendez (212) 591-8522

B15 Committee on Safety Standards for Mechanical Power Transmission Apparatus
Joseph Pang (212) 591-8525

B19 Committee on Safety Standards for Compressor Systems
Melissa Aranzamendez (212) 591-8522

B20 Committee on Safety Standards for Conveyors and Related Equipment
Joseph Wendler (212) 591-8524

B30 Committee on Safety Standards for Cableways, Cranes, Derricks, Hoists, Hooks, Jacks, and Slings
Joseph Pang (212) 591-8525

B56 Committee on Powered and Nonpowered Industrial Trucks
Umberto D'Urso (212) 591-8535

CSD Committee on Controls and Safety Devices for Automatically Fired Boilers
Melissa Aranzamendez (212) 591-8522

PALD Committee on Portable Automotive Lifting Devices
Joseph Wendler (212) 591-8524

QEI Committee on Qualification of Elevator Inspectors
Shari Griffin (212) 591-8586

QFO Committee on Qualification of High Capacity Fossil Fuel Fired Plant Operators
Joseph Wendler (212) 591-8524

QHO Committee on Qualification of Hazardous Waste Incinerator Operators
Shari Griffin (212) 591-8586

QMO Committee on Qualification of Medical Waste Incinerator Operators
John Millman (212) 591-8584

QRO Committee on Qualification of Resource Recovery Facility Operators
John Millman (212) 591-8584

SPPE Committee on Safety and Pollution Prevention Equipment Used in Offshore Oil and Gas Operations
Melissa Aranzamendez (212) 591-8522

STANDARDIZATION

Board on Standardization (BoS)
Steve Weinman (212) 591-7002

A112 Plumbing Materials and Equipment
Calvin Gomez (212) 591-7021

B1 Screw Threads
James Bird (212) 591-8018

B5 Machine Tools
James Bird (212) 591-8018

B18 Fasteners
Kate Padilla (212) 591-7004

B29 Chains and Sprockets
Kate Padilla (212) 591-7004

B32 Metal and Metal Alloy Wrought Mill Product Nominal Size
Steve Weinman (212) 591-7002

B40—Specifications for Pressure and Vacuum Gauges
James Bird (212) 591-8018

B46 Classification and Designation of Surface Qualities
Brian Biddinger (212) 591-8684

B47 Gage Blanks
Calvin Gomez (212) 591-7021

B73 Chemical Standard Pumps
Calvin Gomez (212) 591-7021

B89 Dimensional Metrology
Brian Biddinger (212) 591-8684

B94 Cutting Tools, Holders, Drivers, and Bushings
Brian Biddinger (212) 591-8684

B107 Hand Tools and Accessories
James Bird (212) 591-8018

B133 Gas Turbine Procurement
Kate Padilla (212) 591-7004

HST—Overhead Hoist
Calvin Gomez (212) 591-7021

MFC Measurement of Fluid in Closed Conduits
Kate Padilla (212) 591-7004

MH1 Pallets, Slip Sheets, and Other Bases for Unit Loads
Kate Padilla (212) 591-7004

SRB Slewing Ring Bearings
Oliver Martinez (212) 591-8553

STS Steel Stacks
Oliver Martinez (212) 591-8553

Y14 Engineering Drawing and Related Documentation Practices
Calvin Gomez (212) 591-7021

Data Report Forms

Boiler and Pressure Vessel Code Data Report Forms are available in pads of 100 sheets each at a cost of $22.00 per pad. Please order by the Order Number.

		Order No.
A-1	Manufacturer's Data Report for Pressure Vessels	E00112
A-2	Manufacturer's Partial Data Report (A Part of a Pressure Vessel Fabricated by One Manufacturer for Another Manufacturer)	E00114
A-3	Manufacturer's Data Report Supplementary Sheet	E00119
C-1	Certificate Holder's Data Report for Concrete Reactor Vessels and Containments	E00077
H-2	Manufacturers' Data Report for All Types of Boilers Except Watertube and Those Made of Cast Iron	E00101
H-3	Manufacturers' Data Report for Watertube Boilers	E00102
H-4	Manufacturers' Partial Data Report	E00103
H-5	Manufacturers' Master Data Report for Boilers Constructed from Cast Iron	E00041
HLW-6	Manufacturers' Data Report for Water Heaters or Storage Tanks	E00105
HLW-7	Manufacturers' Partial Data Report for Water Heaters and Storage Tanks	E00106
HLW-8	Manufacturers' Master Data Proof Test Report for Water Heaters or Storage Tanks	E00107
K-1	Manufacturers' Data Report for High Pressure Vessels	E00130
K-2	Manufacturers' Partial Data Report for High Pressure Vessels	E00131
K-3	Manufacturers' Data Report and Supplementary Sheet	E00132
N-1	Certificate Holder's Data Report for Nuclear Vessels	E00038
N-1A	Certificate Holder's Data Report for Nuclear Vessels (Alternate Form for Single Chamber Completely Shop-Fabricated Vessels Only)	E00039
N-2	Certificate Holder's Data Report for Identical Nuclear Parts and Appurtenances	E00040
N-3	Owner's Data Report for Nuclear Power Plant Components	E00024
N-5	Certificate Holder's Data Report for Installation or Shop Assembly of Nuclear Power Plant Components, Component Supports, and Appurtenances	E00025
N-6	Certificate Holder's Data Report for Storage Tanks	E00054
NCS-1	Certificate Holder's Data Report for Core Support Structures	E00065
ND-1	Certificate Holder's Data Report for Rupture Disk Devices	E00128
NF-1	Certificate Holder's Data Report for Components Supports	E00075
NIS-1	Owner's Report for Inservice Inspections	E00029
NIS-2	Owner's Report for Repairs or Replacements	E00030
NIS-2A	Repair/Replacement Certification Record	E00126
NM-1	Certificate Holder's Data Report for Tubular Products and Fittings Welded with Filler Metal	E00080
NPP-1	Certificate Holder's Data Report for Fabricated Nuclear Piping Subassemblies	E00062
NPV-1	Certificate Holder's Data Report for Nuclear Pumps or Valves	E00037
NV-1	Certificate Holder's Data Report for Pressure or Vacuum Relief Valves	E00042

OAR-1	Owner's Activity Report	E00127
P-2	Manufacturers' Data Report for All Types of Boilers Except Watertube and Electric	E00068
P-2A	Manufacturers' Data Report for All Types of Electric Boilers	E00120
P-3	Manufacturers' Data Report for Watertube Boilers, Superheaters, Waterwalls, and Economizers	E00069
P-3A	Engineering-Contractor Data Report for a Complete Boiler Unit	E00070
P-4	Manufacturer's Partial Data Report	E00071
P-4A	Manufacturer's Data Report for Fabricated Piping	E00072
P-5	Summary Data Report for Process Steam Generators	E00073
P-6	Manufacturers' Data Report Supplementary Sheet	E00074
P-7	Manufacturers' Data Report for Safety Valves	E00129
QB-482	Brazing Procedure Specification (BPS)	E00031
QB-484	Manufacturer's Record of Brazer or Brazing Operator Qualification Tests	E00033
QW-482	Suggested Format for Welding Procedure Specification (WPS)	E00006
QW-483	Suggested Format for Procedure Qualification Record (PQR)	E00007
QW-484	Suggested Format for Welder/Welding Operator Performance Qualifications (WPQ)	E00008
RP-1	Fabricator's Data Report for Fiber-Reinforced Plastic Pressure Vessels (Class I)	E00115
RP-2	Fabricator's Partial Data Report (Class I) (A Part of a Fiber-Reinforced Plastic Pressure Vessel Fabricated by One Manufacturer for Another Manufacturer)	E00116
RP-3	Fabricator's Data Report for Class II Vessels	E00124
RP-4	Fabricator's Partial Data Report for Class II Vessels (A Part of a Fiber-Reinforced Plastic Pressure Vessel Fabricated by One Fabricator for Another Fabricator)	E00125
U-1	Manufacturer's Data Report for Pressure Vessels	E00108
U-1A	Manufacturer's Data Report for Pressure Vessels (Alternative Form for Single Chamber, Completely Shop-Fabricated Vessels Only)	E00117
U-2	Manufacturer's Partial Data Report (A Part of a Pressure Vessel Fabricated by One Manufacturer for Another Manufacturer)	E00110
U-2A	Manufacturer's Partial Data Report (Alternative Form)	E00121
U-3	Manufacturer's Certificate of Compliance Covering Pressure Vessels to be Stamped with the UM Symbol. See U-1(j).	E00111
U-4	Manufacturer's Data Report Supplementary Sheet	E00118

Books of Interest to Boiler and Pressure Vessel Code Users

The Code: An Authorized History of the ASME Boiler and Pressure Vessel Code

by Wilbur Cross

Explore the fascinating history of the ASME Boiler and Pressure Vessel Code. From generations past and the events that led to the need for codes to the personalities and pioneers who created the first codes, this in-depth volume chronicles one of engineering's greatest achievements.

Contents: Recent Highlights and Milestones; Early History of Steam Power; Events Leading Up to the Need for Codes; People Who Created the First Codes; All Work and No Relief for the Committee; Progress and Challenges; Standards and Standardization; The Decade of Power; Pushing the Limits; The "Great Period"; Pressure Vessel Code Development; The Technology of Examination and Testing; Keeping Abreast of Technology; The Code Goes to War; The Nuclear Dawn; History Repeated; Achievements and Setbacks; A Century of Progress; Conclusion.

1989; 240 pp.; ISBN 0-7918-2024-6; Order No. 820246; $32.00.

Pressure Vessel and Piping Design Collected Papers: 1927–1959

Editorial committee: E. O. Bergman, J. J. Murphy, J. E. Soehrens, D. B. Wesstrom, F. S. G. Williams, and L. P. Zick.

This volume gives engineers a wider base of reference from which to make decisions on problems they encounter daily. It provides a valuable overview of the pressure retention discipline and will give readers an improved intuitive recognition of good design. Topics include: Openings; Bolted-Flange Joints; Heads; Shells;

Piping; Materials; Thermal Stress and Fatigue; Loads and Supports; External Pressure.

1960 (original printing); 720 pp.; hardcover; Order No. I00364; $95 (list)/$76 (ASME member).

Boiler and Pressure Vessel Certificates of Authorization for Use of Code Symbol Stamps and Testing Laboratories Holding Certificates of Acceptance for Capacity Certification of Pressure Relief Devices

The volume lists company names, addresses, and certificate expiration dates for companies holding Boiler and Pressure Vessel Certificates of Authorization for Sections I, IV, VIII Divisions 1 and 2, and X. Includes listings by Stamp (A, E, M, PP, S, V, H, HLW, HV, U, UM, UV, U2, and RP), and by U.S. states, Canadian provinces and other countries. Also includes a listing of companies holding certificates of acceptance for capacity certification of pressure relief devices.

Order No. E00052; $75.00.

Companies Holding Nuclear Certificates

Lists company names, addresses, scopes, and expiration dates for all nuclear certificates (N, NA, NPT, NV, QSC (MM), QSC (MO), QSC (MS), Interim Letters and Owner's Certificates). Also includes listing of accredited authorized nuclear inspection agencies.

Order No. E00061; $65.00.

Concensus Operating Practices for Control of Feedwater/Boiler Water Chemistry in Modern INDL Boilers (CRTD 34)

48 pp.; ISBN 0-7918-1204-9; Order No. I00367; $10 (list)/$8 (ASME member).

Key to Symbols

Key to Updating Service Symbols and Reaffirmation (R) Designation
Codes and Standards with updating service are indicated by the following symbols:

A = Addenda
Additions and revisions that are sent automatically to purchasers of code or standard.

I = Interpretations
Written replies to inquiries concerning interpretation of technical aspects of code or standard. Sent automatically to purchasers of code or standard.

C = Cases
Codes and standards committees meet regularly to consider proposed additions and revisions to code or standard and to formulate Cases to clarify the intent of existing requirements or provide, when the need is urgent, rules for materials or constructions not covered by existing rules. Sent automatically to purchasers of code or standard.

S = Supplement
Same as "A," Addenda

Year
(ex. A I 2000)
Next publication date of code or standard.
Updating service includes Addenda, Cases, Interpretations, or Supplements up to this date.

Reaffirmation
(ex. R1997)
Committee has determined that no changes are warranted in code or standard.

L = List
List of certificate holders

Alphanumeric Index

Codes & Standards	*Page*	*Codes & Standards*	*Page*
A13	15	**HST**	5
A17	6, 7	**MC88**	11
A90	10	**MFC**	9
A112	16	**MH1**	13
A120	16	**N278**	12
AG	12	**N509**	12
B1	17	**N510**	12
B4	5	**N626**	12
B5	18, 19	**NOG**	12
B15	10	**NQA**	12
B16	12, 22	**NUM**	12
B17	10	**OM**	12
B18	7–9	**PALD**	3
B19	4	**PTC**	13–15
B20	4	**PVHO**	16
B27	9	**QAI**	3
B29	3	**QEI**	7
B30	4, 5	**QHO**	13
B31	15	**QME**	12
B32	5	**QMO**	13
B36	5	**QRO**	13
B40	9	**RTP**	17
B46	18	**SI**	11
B47	9	**SPPE**	12
B56	10	**STS**	18
B73	16	**Y1**	5
B89	11	**Y14**	5, 6
B94	19, 20	**Y32**	6
B96	18		
B107	20, 21		
B133	21		
BPE	16		
CSD	4		
FAP	7		
HPS	10		

AUTHORIZED INSPECTION

QAI-1-1995
Qualifications for Authorized Inspection
A I L 1999
Order No. J09595
$38.00

AUTOMOTIVE LIFTING DEVICES
Safety standard for portable automotive lifting devices, covering requirements for safety, health, design, production, construction, maintenance, performance, or operation of equipment, or qualification of personnel.

PALD-1997
Safety Standard for Portable Automotive Lifting Devices
A I 2002
Order No. J13597
$89.00

BOILERS

Boiler and Pressure Vessel Code (see page 23)

CHAINS
Standards for the dimensions of transmission and conveyor chains.

B29.1M-1993
Precision Power Transmission Roller Chains, Attachments and Sprockets
Order No. K12793
$35.00

B29.2M-1982(R1994)
Inverted Tooth (Silent) Chains and Sprockets
Order No. K00111
$29.00

B29.3M-1994
Double-Pitch Power Transmission Roller Chains, and Sprockets
Order No. K07594
$30.00

B29.4M-1994
Double-Pitch Conveyor Roller Chains, Attachments and Sprockets
Order No. M07294
$32.00

B29.6M-1993
Steel Detachable Link Chains, Attachments, and Sprockets
Order No. M06693
$39.00

B29.8M-1993
Leaf Chains, Clevises and Sheaves
Order No. M00393
$29.00

B29.10M-1997
Heavy Duty Offset Sidebar Power Transmission Roller Chains and Sprocket Teeth
Order No. M07897
$33.00

B29.11M-1994
Combination Chains, Attachments, and Sprocket Teeth
Order No. K08194
$29.00

B29.12M-1997
Steel Bushed Rollerless Chains, Attachments, and Sprocket Teeth
Order No. K03097
$32.00

B29.14M-1996
"H" Type Mill Chains, Attachments, and Sprocket Teeth
Order No. M06296
$29.00

B29.15M-1997
Steel Roller Type Conveyor Chains, Attachments, and Sprocket Teeth
Order No. M07997
$34.00

B29.16M-1995
Welded Steel Type Mill Chains, Attachments, and Sprocket Teeth
Order No. K07995
$29.00

B29.17M-1983(R1995)
Hinge Type Flat Top Conveyor Chains and Sprocket Teeth
Order No. K00078
$29.00

B29.18M-1994
Welded Steel-Type Drag Chains, Attachments, and Sprocket Teeth
Order No. K08093
$29.00

B29.19M-1996
Agricultural Roller Chains, Attachments, and Sprockets
Order No. K08596
$29.00

B29.21M-1996
700 Class Welded Steel and Cast Chains, Attachments, and Sprockets for Water and Sewage Treatment Plants
Order No. K10596
$29.00

B29.22M-1995
Drop Forged Rivetless Chains, Sprocket Teeth Drive Chain/Drive Dogs
Order No. K10495
$27.00

B29.23M-1985(R1995)
Flexible Chain Couplings
Order No. K00114
$29.00

B29.24M-1995
Roller Load Chains for Overhead Hoists
Order No. K12695
$29.00

B29.25M-1994
Open Barrel Steel Pintle Type Conveyor Chains, Attachments, and Sprockets
Order No. K12494
$28.00

B29.26M-1996
Fatigue Testing Power Transmission Roller Chain
Order No. M16096
$32.00

COMPRESSORS
Safety standards for the construction, installation, operation, and maintenance of air and gas compression equipment.

B19.1-1995
Safety Standard for Air Compressor Systems
A I 2000
Order No. J09795
$48.00

B19.3-1991
Safety Standard for Compressors for Process Industries
A I 1996
Order No. JX7091
$37.00

CONTROLS
Safety requirements for the assembly, maintenance, and operation of controls and safety devices installed on automatically operated boilers directly fired with gas, oil, gas/oil, or electricity.

CSD-1-1998
Controls and Safety Devices for Automatically Fired Boilers
A I 2001
Order No. A09198
$52.00

Form CSD-1
Manufacturer's/ Installing Contractor's Report for ASME CSD-1
Order No. A091F1
$18.00

CONVEYORS
Safety standard for the design, construction, installation, maintenance, inspection, and operation of conveyors and conveying systems in relation to accident hazards.

B20.1-1996
Safety Standard for Conveyors and Related Equipment
A I 1999
Order No. J08897
$58.00

CRANES AND HOISTS
Safety standards for the construction, installation, operation, inspection, and maintenance of jacks, cranes, monorails, derricks, hoists, hooks, slings, and cableways; and specification standards for hoists.

B30.1-1998
Jacks
A I 2003
Order No. J06998
$57.00

B30.2-1996
Overhead and Gantry Cranes (Top Running Bridge, Single or Multiple Girder, Top Running Trolley Hoist)
A I 2001
Order No. J08996
$52.00

B30.3-1996
Construction Tower Cranes
A I 2001
Order No. J03396
$52.00

B30.4-1996
Portal, Tower, and Pedestal Cranes
A I 2001
Order No. J00196
$45.00

B30.5-1994
Mobile and Locomotive Cranes
A I 1999
Order No. J08394
$58.00

B30.6-1995
Derricks
A I 2000
Order No. J09395
$45.00

B30.7-1994
Base Mounted Drum Hoists
A I 1994
Order No. JX9094
$54.00

B30.8-1993
Floating Cranes and Floating Derricks
AI 1998
Order No. J12493
$52.00

B30.9-1996
Slings
A I 2001
Order No. J09296
$75.00

B30.10-1993
Hooks
A I 1998
Order No. J04893
$45.00

B30.11-1998
Monorails and Underhung Cranes
A I 2003
Order No. J12098
$52.00

B30.12-1992
Handling Loads Suspended From Rotorcraft
A I 1997
Order No. J04792
$47.00

B30.13-1996
Storage/Retrieval (S/R) Machines and Associated Equipment
A I 2001
Order No. J05296
$48.00

B30.14-1996
Side Room Tractors
A I 2001
Order No. J05996
$42.00

B30.16-1993
Overhead Hoists (Underhung)
A I 1998
Order No. J11693
$52.00

B30.17-1992
Overhead and Gantry Cranes (Top Running Bridge, Single Girder, Underhung Hoist)
A I 1997
Order No. JX6092
$40.00

B30.18-1993
Stacker Cranes (Top or Under Running Bridge, Multiple Girder with Top or Under Running Trolley Hoist)
A I 1998
Order No. J11393
$45.00

B30.19-1993
Cableways
A I 1998
Order No. J10893
$47.00

CRANES AND HOISTS
cont'd

B30.20-1993
Below-the-Hook Lifting Devices
A I 1998
Order No. J10093
$53.00

B30.21-1994
Manually Lever Operated Hoists
A I 1999
Order No. J12694
$49.00

B30.22-1993
Articulating Boom Cranes
A I 1998
Order No. J11793
$53.00

B30.25-1998
Scrap and Material Handlers
A I 2003
Order No. J14698
$50.00

HST-1M-1989(R1995)
Performance Standard for Electric Chain Hoists
Order No. J07889
$29.00

HST-2M-1989(R1995)
Performance Standard for Hand Chain Manually Operated Chain Hoists
Order No. J08789
$29.00

HST-3M-1991(R1996)
Performance Standard for Manually Lever Operated Chain Hoists
Order No. J09891
$29.00

HST-4M-1991(R1996)
Performance Standard for Overhead Electric Wire Rope Hoists
Order No. J09991
$29.00

HST-5M-1991(R1996)
Performance Standard for Air Chain Hoists
Order No. J10291
$29.00

HST-6M-1986(R1995)
Performance Standard for Air Wire Rope Hoists
Order No. J00106
$29.00

DIMENSIONS
Standards for limits and fits, preferred sizes, and pipe dimensions.

B4.1-1967(R1994)
Preferred Limits and Fits for Cylindrical Parts
Order No. L00022
$30.00

B4.2-1978(R1994)
Preferred Metric Limits and Fits
Order No. L00054
$32.00

B4.3-1978(R1994)
General Tolerances for Metric Dimensioned Products
Order No. L00055
$30.00

B4.4M-1981(R1994)
Inspection of Workpieces
Order No. L00062
$30.00

B32.1-1952(R1994)
Preferred Thickness for Uncoated, Thin, Flat Metals (Under 0.250/in.)
Order No. L00025
$29.00

B32.2-1969(R1994)
Preferred Diameters for Round Wire—0.500 Inches and Under
Order No. M00012
$29.00

B32.3M-1984(R1994)
Preferred Metric Sizes for Flat Metal Products
Order No. L00045
$29.00

B32.4M-1980(R1994)
Preferred Metric Sizes for Round, Square, Rectangle and Hexagon Metal Products
Order No. L00046
$29.00

B32.5-1977(R1994)
Preferred Metric Sizes for Tubular Metal Products other than Pipe
Order No. L00051
$29.00

B32.6M-1984(R1994)
Preferred Metric Equivalents of Inch Sizes for Tubular Metal Products other than Pipe
Order No. L00052
$29.00

B36.10M-1996
Welded and Seamless Wrought Steel Pipe
Order No. M03196
$35.00

B36.19M-1985(R1994)
Stainless Steel Pipe
Order No. M00013
$29.00

DRAWINGS AND TERMINOLOGY
Standards for drafting, abbreviations, letter symbols, graphs, and charts.

Abbreviations
Y1.1-1989
Abbreviations—for Use on Drawings and in Text
Order No. J00389
$49.00

Drafting Standards
Y14.1-1995
Decimal Inch Drawing Sheet Size and Format
Order No. N00195
$32.00

Y14.1M-1995
Metric Drawing Sheet Size and Format
Order No. N12995
$32.00

Y14.2M-1992(R1998)
Line Conventions and Lettering
Order No. N00292
$32.00

Y14.3M-1994
Multi and Sectional View Drawings
Order No. N00394
$40.00

Y14.4M-1989(R1994)
Pictorial Drawing
Order No. N00489
$29.00

Y14.5M-1994
Dimensioning and Tolerancing
Order No. N00594
$93.00

Y14.5.1M-1994
Mathematical Definition of Dimensioning and Tolerancing
Order No. N13294
$49.00

Y14.5M-1994 PLUS Y14.5.1M-1994
Special Offer
Order No. YX1494
$120.00

Y14.5.2-1995
Certification of Geometric Dimensioning and Tolerancing Professionals
Order No. N13495
$32.00

Y14.6-1978(R1993)
Screw Thread Representation
Order No. N00006
$29.00

DRAWINGS AND TERMINOLOGY
cont'd

Y14.6M-1981(R1998)
Screw Thread Representation (Metric Supplement)
Order No. N00097
$29.00

Y14.7.1-1971(R1998)
Gear Drawing Standards—Part 1: For Spur, Helical, Double Helical and Rack
Order No. K00067
$29.00

Y14.7.2-1978(R1994)
Gear and Spline Drawing Standards—Part 2: Bevel and Hypoid Gears
Order No. N00081
$29.00

Y14.8M-1996
Castings and Forgings
Order No. N12096
$46.00

Y14.13M-1981(R1998)
Mechanical Spring Representation
Order No. N00098
$29.00

Y14.18M-1986(R1998)
Optical Parts
Order No. N00115
$29.00

Y14.24M-1989(R1996)
Types and Applications of Engineering Drawings
Order No. N00122
$62.00

Y14.32.1M-1994
Chassis Frames Passenger Car and Light Truck—Ground Vehicle Practices
Order No. N13394
$32.00

Y14.34M-1996
Associated Lists
Order No. N10296
$35.00

Y14.35M-1997
Revision of Engineering Drawings and Associated Documents
Order No. N13097
$35.00

Y14.36M-1996
Surface Texture Symbols
Order No. N08096
$44.00

Y14.100M-1998
Engineering Drawing Practices
Order No. N13898
$45.00

Y14
Technical Report 4-1989 A Structural Language Format for Basic Shape Description
Order No. N11989
$94.00

Graphic Symbols Standards
Y32.2.3-1949(R1994)
Redesignation of Z32.2.3-1949(R1953) Graphic Symbols for Pipe Fittings, Valves and Piping
Order No. K00006
$30.00

Y32.2.4-1949(R1998)
Graphic Symbols for Heating, Ventilating, and Air Conditioning
Order No. K00005
$29.00

Y32.2.6-1950(R1998)
Graphic Symbols for Heat-Power Apparatus
Order No. K00004
$29.00

Y32.4-1997(R1994)
Graphic Symbols for Plumbing Fixtures for Diagrams Used in Architecture and Building Construction
Order No. K00007
$29.00

Y32.7-1972(R1994)
Graphic Symbols for Railroad Maps and Profiles
Order No. K00008
$29.00

Y32.10-1967(R1994)
Graphic Symbols for Fluid Power Diagrams
Order No. N00022
$29.00

Y32.11-1961(R1998)
Graphic Symbols for Process Flow Diagrams in Petroleum and Chemical Industries
Order No. K00040
$29.00

Y32.18-1972(R1998)
Symbols for Mechanical and Acoustical Elements as Used in Schematic Diagrams
Order No. K00011
$29.00

ELEVATORS AND ESCALATORS
Safety codes and standards for the design, construction, installation, operation, inspection, testing, maintenance, alteration, and repair of elevators, dumbwaiters, escalators, moving walks, material lifts and dumbwaiters with automatic transfer devices, wheelchair lifts, and inclined-stairway chair lifts.

A17 CD-ROM for Elevators and Escalators
Order No. A00144
$495.00

A17.1-1996
Safety Code for Elevators and Escalators
A I 1999
Order No. A09696
$125.00

A17.1-1993
Safety Code for Elevators and Escalators
A I 1996
Order No. A09693
$114.00

A17.1-1990
Safety Code for Elevators and Escalators
Order No. AX9690
$103.00

Handbook on A17.1 Safety Code for Elevators and Escalators-1996 Edition
Order No. A11296
$105.00

A17.1 special Offer-A17.1-1996 Plus A17.1 Handbook (1996 Edition)
Order No. AX1796
$205.00

A17.2.1-1996
Inspector's Manual for Electric Elevators
A I 2000
Order No. A13496
$52.00

A17.2.2-1997
Inspector's Manual for Hydraulic Elevators
A I 2000
Order No. A13997
$54.00

A17.2.3-1994
Inspector's Manual for Escalators and Moving Walks
A I 1997
Order No. A14194
$42.00

A17.3-1996
Safety Code for Existing Elevators and Escalators
A I 2001
Order No. A12396
$55.00

A17.3-1990
Safety Code for Existing Elevators and Escalators
Order No. A12390
$47.00

A17.4-1991
Guide for Emergency Evacuation of Passengers from Elevators
Order No. A12591
$25.00

A17.5-1996
Elevator and Escalator Electrical Equipment
A I 2000
Order No. A13296
$69.00

ELEVATORS AND ESCALATORS
cont'd

Interpretations: A17 Documents 1972–1979
Order No. A00108
$25.00

A17 Interpretations Nos. 2–13 1989
Order No. A96187
$39.00

QEI-1-1997
Standard for the Qualification of Elevator Inspectors
A I L 2000
Order No. A12097
$49.00

Checklist for Inspection and Test of Elevators Used for Construction*
Order No. A49C10
$41.00

Checklist for Inspection of Electric Elevators*
Order No. A134C1
$39.00

Checklist for Inspection of Hydraulic Elevators*
Order No. A050C7
$28.00

Checklist for Inspection of Escalators and Moving Walks*
Order No. A141C1
$29.00

**In pad form (100 per pad)*

FASTENERS
Standards for bolts, nuts, rivets, washers, keys, eyebolts, pins, and retaining rings.

FAP-1-1995
Quality Assurance Program Requirements for Fastener Manufacturers and Distributors and Testing Laboratories
A I L 1999
Order No. M13995
$46.00

Metric Fasteners
B18.1.3M-1983 (R1995)
Metric Small Solid Rivets
Order No. M00131
$29.00

B18.2.3.1M-1979 (R1995)
Metric Hex Cap Screws
Order No. M00100
$29.00

B18.2.3.2M-1979 (R1995)
Metric Formed Hex Screws
Order No. M00101
$29.00

B18.2.3.3M-1979 (R1995)
Metric Heavy Hex Screws
Order No. M00102
$29.00

B18.2.3.4M-1984 (R1995)
Metric Hex Flange Screws
Order No. M00103
$29.00

B18.2.3.5M-1979 (R1995)
Metric Hex Bolts
Order No. M00104
$29.00

B18.2.3.6M-1979 (R1995)
Metric Heavy Hex Bolts
Order No. M00105
$29.00

B18.2.3.7M-1979 (R1995)
Metric Heavy Hex Structural Bolts
Order No. M00106
$29.00

B18.2.3.8M-1981 (R1991)
Metric Hex Lag Screws
Order No. M00107
$29.00

B18.2.3.9M-1984 (R1995)
Metric Heavy Hex Flange Screws
Order No. M00133
$29.00

B18.2.3.10M-1996
Square Head Bolts (Metric Series)
Order No. M15896
$28.00

B18.2.4.1M-1979 (R1995)
Metric Hex Nuts, Style 1
Order No. M00114
$29.00

B18.2.4.2M-1979 (R1995)
Metric Hex Nuts, Style 2
Order No. M00117
$29.00

B18.2.4.3M-1979 (R1995)
Metric Slotted Hex Nuts
Order No. M00119
$29.00

B18.2.4.4M-1982 (R1993)
Metric Hex Flange Nuts
Order No. M00122
$29.00

B18.2.4.5M-1979 (R1998)
Metric Hex Jam Nuts
Order No. M00115
$29.00

B18.2.4.6M-1979 (R1998)
Metric Heavy Hex Nuts
Order No. M00116
$29.00

B18.3.1M-1986 (R1993)
Socket Head Cap Screws
Order No. M00094
$29.00

B18.3.2M-1979 (R1998)
Metric Series Hexagon Keys and Bits
Order No. M00108
$29.00

B18.3.3M-1986 (R1993)
Hexagon Socket Head Shoulder Screws
Order No. M00109
$29.00

B18.3.4M-1986 (R1993)
Hexagon Socket Button Head Cap Screws
Order No. M00110
$29.00

B18.3.5M-1986 (R1993)
Hexagon Socket Flat Countersunk Head Cap Screws
Order No. M00130
$29.00

B18.3.6M-1986 (R1993)
Metric Series Socket Set Screws
Order No. M00111
$29.00

B18.5.2.1M-1996
Metric Round Head Short Square Neck Bolts
Order No. L06496
$32.00

B18.5.2.2M-1982 (R1993)
Metric Round Head Square Neck Bolts
Order No. M00124
$29.00

B18.5.2.3M-1990 (R1998)
Round Head Square Neck Bolts with Large Head
Order No. M00141
$29.00

B18.6.5M-1986 (R1993)
Metric Thread Forming and Thread Cutting Tapping Screws
Order No. N00110
$30.00

B18.6.7M-1985 (R1993)
Metric Machine Screws
Order No. M00136
$29.00

FASTENERS
cont'd

B18.7.1M-1984
(R1992)
Metric General Purpose Semi-Tubular Rivets
Order No. N00106
$29.00

B18.8.3M-1995
Spring Pins—Coiled Type
Order No. M14295
$34.00

B18.8.4M-1994
Spring Pins—Slotted
Order No. M14794
$32.00

B18.8.5M-1994
Machine Dowel Pins—Hardened Ground
Order No. M14894
$32.00

B18.8.6M-1995
Cotter Pins (Metric Series)
Order No. M14695
$32.00

B18.8.7M-1994
Headless Clevis Pins
Order No. M14994
$32.00

B18.8.8M-1994
Headed Clevis Pins
Order No. M15094
$32.00

B18.8.9M-1998
Grooved Pins (Metric Series)
Order No. M16598
$34.00

B18.12-1962(R1991)
Glossary of Terms for Mechanical Fasteners
Order No. M00046
$29.00

B18.13.1M-1991
Screw and Washer Assemblies—Sems
A I 1996
Order No. N00124
$42.00

B18.16.1M-1979
(R1995)
Mechanical and Performance Requirements for Prevailing Torque-Type Steel Metric Hex Nuts and Hex Flange Nuts
Order No. M00118
$29.00

B18.16.2M-1979
(R1995)
Torque-Tension Test Requirements for Prevailing-Torque Type Steel Metric Hex Nuts and Hex Flange Nuts
Order No. M00113
$29.00

B18.16.3M-1982
(R1993)
Dimensional Requirements for Prevailing-Torque Type Steel Metric Hex Nuts and Hex Flange Nuts
Order No. M00123
$29.00

B18.18.1M-1987
(R1993)
Inspection and Quality Assurance for General Purpose Fasteners
Order No. M00125
$29.00

B18.18.2M-1987
(R1993)
Inspection and Quality Assurance for High-Volume Machine Assembly Fasteners
Order No. M00126
$29.00

Supplement to B18.18.2
Order No. M0126S
$29.00

B18.18.3M-1987
(R1993)
Inspection and Quality Assurance for Special Purpose Fasteners
Order No. M00127
$29.00

B18.18.4M-1987
(R1993)
Inspection and Quality Assurance for Fasteners for Highly Specialized Engineered Applications
Order No. M00128
$29.00

B18.18.5M-1998
Inspection and Quality Assurance Plan Requiring In-Process Inspection and Control
Order No. M16298
$32.00

B18.18.6M-1998
Quality Assurance Plan for Fasteners Produced in a Third Party Accreditation System
Order No. M16398
$28.00

B18.18.7M-1998
Quality Assurance Plan for Fasteners Produced in a Customer Approved Control Plan
Order No. M16498
$29.00

B18.21.2M-1994
Lock Washers
Order No. M14094
$32.00

B18.22M-1981(R1990)
Metric Plain Washers
Order No. L00063
$29.00

B18.24.1-1996
Part Identifying Number (PIN) Code System Standard for B18 Externally Threaded Products
A I 2002
Order No. M15996
$55.00

B18.25.1M-1996
Square and Rectangular Keys and Keyways
Order No. M15696
$32.00

B18.25.2M-1996
Woodruff Keys and Keyways
Order No. M15796
$32.00

B27.7-1977(R1993)
General Purpose Tapered and Reduced Cross Section Retaining Rings (Metric)
Order No. L00053
$29.00

B27.8M-1978(R1993)
General Purpose Metric Tapered and Reduced Cross Section Retaining Rings-Type 3DM1-Heavy Duty External Rings, Type 3EM1-Reinforced E″ Rings, Type 3FM1-8C Type Rings
Order No. L00060
$29.00

Fasteners in Customary Units
B18.1.1-1972(R1995)
Small Solid Rivets
Order No. M00006
$29.00

B18.1.2-1972(R1995)
Large Rivets
Order No. K00023
$29.00

B18.2.1-996
Square and Hex Bolts and Screws (Inch Series)
A I 1999
Order No. M04496
$47.00

B18.2.2-1987(R1993)
Square and Hex Nuts (Inch Series)
Order No. M00043
$29.00

B18.2.6-1996
Fasteners for Use in Structural Applications
A I 1999
Order No. M16196
$34.00

FASTENERS
cont'd

B18.3-1998
Socket Cap, Shoulder, and Set Screws, Hex and Spline Keys (Inch Series)
Order No. M00798
$39.00

B18.5-1990(R1998)
Round Head Bolts (Inch Series)
Order No. L03290
$30.00

B18.6.1-1981(R1997)
Wood Screws (Inch Series)
Order No. M0005
$29.00

B18.6.2-1972(R1993)
Slotted Head Cap Screws, Square Head Set Screws and Slotted Headless Set Screws
Order No. K00012
$29.00

B18.6.3-1972(R1997)
Machine Screws and Machine Screw Nuts
Order No. M00004
$29.00

B18.6.4-1981(R1997)
Thread Forming and Thread Cutting Tapping Screws and Metallic Drive Screws (Inch Series)
Order No. N00021
$30.00

B18.7-1972(R1992)
General Purpose Semi-Tubular Rivets, Full Tubular Rivets, Split Rivets and Rivet Caps
Order No. N00059
$29.00

B18.8.1-1994
Clevis Pins and Cotter Pins
Order No. L02494
$32.00

B18.8.2-1995
Taper Pins, Dowel Pins, Straight Pins, Grooved Pins and Spring Pins (Inch Series)
Order No. L05695
$39.00

B18.9-1996
Plow Bolts (Inch Series)
Order No. L02196
$32.00

B18.10-1982(R1992)
Track Bolts and Nuts
Order No. M00025
$29.00

B18.11-1961(R1992)
Miniature Screws
Order No. K00037
$29.00

B18.13-1996
Screw and Washer Assemblies—Sems (Inch Series)
Order No. N04396
$34.00

B18.15-1985(R1995)
Forged Eyebolts
Order No. M00037
$29.00

B18.21.1-1994
Lock Washers (Inch Series)
Order No. L00294
$32.00

B18.22.1-1965(R1998)
Plain Washers (Redesignation of B27.2-1965)
Order No. L00019
$29.00

B18.29.1-1993
Helical Coil Screw Thread Inserts—Free Running and Screw Locking (Inch Series)
Order No. M14392
$32.00

B27.6-1972(R1993)
General Purpose Uniform Cross Section Spiral Retaining Rings
Order No. L00037
$29.00

FLOW MEASUREMENT
Standards for fluid flow volume measurements in pipes.

MFC-1M-1991
Glossary of Terms Used in the Measurement of Fluid Flow in Pipes
Order No. J06591
$29.00

MFC-2M-1983(R1988)
Measurement Uncertainty for Fluid Flow in Closed Conduits
Order No. K00112
$33.00

MFC-3M-1989(R1995)
Measurement of Fluid Flow in Pipes Using Orifice, Nozzle and Venturi (Not an American National Standard)
Order No. K11389
$40.00

MFC-4M-1986(R1997)
Measurement of Gas Flow by Turbine Meters
Order No. K00118
$29.00

MFC-5M-1985(R1994)
Measurement of Liquid Flow in Closed Conduits Using Transit-Time Ultrasonic Flowmeters
Order No. K00115
$29.00

MFC-6M-1998
Measurement of Fluid Flow in Pipes Using Vortex Flow Meters
Order No. K11798
$32.00

MFC-7M-1987(R1992)
Measurement of Gas Flow by Means of Critical Flow Venturi Nozzles
Order No. K00119
$29.00

MFC-8M-1988
Fluid Flow in Closed Conduits—Connections for Pressure Signal Transmissions Between Primary and Secondary Devices
Order No. K12188
$29.00

MFC-9M-1988
Measurement of Liquid Flow in Closed Conduits by Weighing Method
Order No. K12588
$29.00

MFC-10M-1994
Method for Establishing Installation Effects on Flowmeters
Order No. K12394
$29.00

MFC-11M-1989 (R1994)
Measurement of Fluid Flow by Means of Coriolis Mass Flowmeters
Order No. K12989
$29.00

MFC-14M-1995
Measurement of Fluid Flow Using Small Bore Precision Orifice Meters
Order No. K13795
$30.00

MFC-16M-1995
Measurement of Fluid Flow in Closed Conduits by Means of Electromagnetic Flowmeters
Order No. K13895
$30.00

GAGE BLANKS
Standard for dimensions of standard blanks and components.

B47.1-1988(R1994)
Gage Blanks
Order No. M08288
$70.00

GAUGES
Standardizes principal features of gauges.

B40.1-1991
Gauges—Pressure Indicating Dial Type—Elastic Element
Order No. K01591
$28.00

B40.2-1991
Diaphragm Seals
Order No. K13191
$29.00

B40.3-1997
Bimetallic Actuated Thermometers
Order No. K13097
$32.00

B40.5-1995
Snubbers
Order No. K13695
$29.00

B40.6-1994
Pressure Limiter Valves
Order No. K13594
$29.00

HIGH PRESSURE SYSTEMS

HPS-1994
High Pressure Systems
A I 1999
Order No. A14094
$75.00

INDUSTRIAL TRUCKS
Safety standards relating to the elements of design, operation, and maintenance; also the standardization relating to principal dimensions to facilitate interchangeability, test methods, and test procedures of industrial trucks.

B56.1-1993
Safety Standard for Low Lift and High Lift Trucks
A I 1998
Order No. J08593
$69.00

B56.5-1993
Safety Standard for Guided Industrial Vehicles and Automated Functions of Manned Industrial Vehicles
A I 1998
Order No. J05793
$33.00

B56.6-1992
Safety Standard for Rough Terrain Forklift Trucks
A I 1997
Order No. J05692
$55.00

B56.7-1987(R1992)
Safety Standard for Industrial Crane Trucks
A I 1997
Order No. J07387
$40.00

B56.8-1993
Safety Standard for Personnel and Burden Carriers
A I 1996
Order No. J06893
$38.00

B56.9-1992
Safety Standard for Operator Controlled Industrial Tow Tractors
A I 1997
Order No. J11492
$40.00

B56.10-1992
Safety Standard for Manually Propelled High Lift Industrial Trucks
A I 1997
Order No. K13292
$50.00

B56.11.1-1992
Double Race or Bi-Level Swivel and Rigid Industrial Casters
A I 1995
Order No. K05192
$35.00

B56.11.3-1992
Load Handling Symbols for Powered Industrial Trucks
A I 1995
Order No. K07392
$32.00

B56.11.4-1992
Hook-Type Forks and Fork Carriers for Powered Industrial Forklift Trucks
A I 1995
Order No. K07492
$34.00

B56.11.5-1992
Measurement of Sound Emitted by Low Lift, High Lift, and Rough Terrain Powered Industrial Trucks
A I 1995
Order No. K12892
$34.00

B56.11.6-1992
Evaluation of Visibility from Powered Industrial Trucks
A I 1995
Order No. K13392
$35.00

KEYS
Standards for keys and keyseats.

B17.1-1967(R1998)
Keys and Keyseats
Order No. J00038
$29.00

B17.2-1967(R1998)
Woodruff Keys and Keyseats
Order No. J00037
$29.00

MACHINE GUARDING
Safety standard applying to any source of hazard to personnel from the operation of mechanical power transmission apparatus on machines, equipment, or systems that are stationary in their use, other than the point of operation.

B15.1-1996
Safety Standard For Mechanical Power Transmission Apparatus
A I 1999
Order No. J09496
$32.00

MANLIFTS
Safety standard for the manufacture, installation, maintenance, inspection, and operation of endless belt manlifts.

A90.1-1997
Safety Standard for Belt Manlifts
A I 2002
Order No. J10197
$39.00

MEASUREMENT
Standards for dimensional metrology and calibration of instruments.

B89
Technical Paper 1990, Space Plate Test Recommendations for Coordinate Measuring Machines
Order No. L07190
$29.00

B89
Technical Report 1990, Parametric Calibration of Coordinate Measuring Machines
Order No. L00070
$29.00

B89.1.2M-1991
Calibration of Gage Blocks by Contact Comparison Methods (Through 20 in. and 500 mm)
Order No. L07291
$32.00

B89.1.6M-1984 (R1997)
Measurement of Qualified Plain Internal Diameters for Use as Master Rings and Ring Gages
Order No. L00048
$29.00

B89.1.9M-1984 (R1997)
Precision Gage Blocks for Length Measurement (Through 20 in. and 500 mm)
Order No. L00044
$29.00

B89.1.10M-1987 (R1995)
Dial Indicators (for Linear Measurements)
Order No. L00057
$29.00

B89.1.12M-1990
Methods for Performance Evaluation of Coordinate Measuring Machines
Order No. L06590
$35.00

B89.3.1-1972(R1997)
Measurement of Out-of-Roundness
Order No. L00020
$29.00

B89.3.4M-1985 (R1992)
Axes of Rotation—Methods for Specifying and Testing
Order No. L00066
$29.00

B89.4.1-1997
Methods for Performance Evaluation of Coordinate Measuring Machines
Order No. L06597
$45.00

B89.6.2-1973(R1995)
Temperature and Humidity Environment for Dimensional Measurement
Order No. L00047
$29.00

MC88.1-1972(R1995)
Guide for Dynamic Calibration of Pressure Transducers
Order No. L00042
$29.00

MC88.2-1974(R1995)
Redesignation of B88.2-1974(R1981) Procedures for Bench Calibration of Tank Level Gaging Tapes and Sounding Rules
Order No. L00043
$29.00

METRIC SYSTEM
Guides for familiarization with the International System (SI) of Units.

ASME Guide SI-1
ASME Orientation and Guide for Use of SI (Metric) Units Ninth Edition
1982
Order No. E00058
$23.00

SI-2
ASME Text Booklet, SI Units in Strength of Materials Second Edition
1976 14 pp.
Order No. E00082
$23.00

SI-3
ASME Text Booklet, SI Units in Dynamics
1976 20 pp.
Order No. E00083
$23.00

SI-4
ASME Text Booklet, SI Units in Thermodynamics
1976 55 pp.
Order No. E00084
$23.00

SI-5
ASME Text Booklet, SI Units in Fluid Mechanics
1976 36 pp.
Order No. E00085
$23.00

SI-6
ASME Text Booklet, SI Units in Kinematics
1976 14 pp.
Order No. E00086
$23.00

SI-7
ASME Text Booklet, SI Units in Heat Transfer
1977 36 pp.
Order No. E00087
$23.00

SI-8
ASME Text Booklet, SI Units in Vibration
1976 13 pp.
Order No. E00088
$23.00

SI-9
ASME Guide for Metrication of Codes and Standards SI (Metric) Units
1980 33 pp.
Order No. E00089
$23.00

SI-10
Steam Charts, SI (Metric) and U.S. Customary Units
1976 128 pp.
Hardbound
Order No. E00090
$28.00

NUCLEAR
Codes and standards applicable to nuclear facilities and technology.

AG-1-1997
Code on Nuclear Air and Gas Treatment
A I 2000
Order No. A12297
$152.00

AG-1-1994
Code on Nuclear Air and Gas Treatment
A I 1997
Order No. A12294
$139.00

B16.41-1983(R1989)
Functional Qualification Requirements for Power Operated Active Valve Assemblies for Nuclear Power Plants
Order No. J00077
$32.00

N278.1-1975(R1992)
Self-Operated and Power-Operated Safety-Related Valves Functional Specification Standard
Order No. A00084
$32.00

N509-1989(R1996)
Nuclear Power Plant Air-Cleaning Units and Components
A I 1994
Order No. A08589
$49.00

N510-1989(1995)
Testing of Nuclear Air-Treatment Systems
I 1994
Order No. A08289
$37.00

N626-1990
Qualifications and Duties for Authorized Nuclear Inspection Agencies and Personnel
A I L 1994
Order No. J09590
$40.00

N626-1985
Qualifications and Duties for Authorized Nuclear Inspection Agencies and Personnel
A I 1990
Order No. J09585
$40.00

N626.0-1982
Qualifications and Duties for Authorized Inspection Agencies, Nuclear Inspectors and Nuclear Inspector Supervisors
Order No. J00074
$29.00

N626.1-1982
Qualifications and Duties for Authorized Inspection Agencies, Nuclear Inservice Inspectors and Nuclear Inservice Inspector Supervisors
Order No. J00075
$29.00

N626.2-1982
Qualifications and Duties for Authorized Inspection Agencies, Nuclear Inspectors (Concrete) and Nuclear Inspector Supervisors (Concrete)
Order No. J00076
$29.00

N626.3-1993
Qualifications and Duties of Specialized Professional Engineers
A I 1998
Order No. A10093
$32.00

NOG-1-1995
Rules for Construction of Overhead And Gantry Cranes (Top Running Bridge, Multiple Girder)
A I 2000
Order No. A11995
$125.00

NQA-1-1997
Quality Assurance Requirements for Nuclear Facility Applications
A I 2000
Order No. A10597
$128.00

NQA-1-1994
Quality Assurance Program Requirements for Nuclear Facilities
A I 1997
Order No. A10594
$125.00

NQA-1-1989
Quality Assurance Program Requirements for Nuclear Facilities
A I 1992
Order No. A10589
$83.00

Comparison of NQA-1 and ISO 9001
Order No. A13793
$30.00

NUM-1-1996
Rules for Construction of Cranes, Monorails, and Hoists (with Bridge or Trolley or Hoist of the Underhung Type)
A I 1999
Order No. J14396
$115.00

OM-1998
Code for the Operation and Maintenance of Nuclear Power Plants
A I
Order No. A12898
$185.00

OM-1995
Code for Operation and Maintenance of Nuclear Power Plants
A I 1998
Order No. A12894
$175.00

OM-1990
Code for Operation and Maintenance of Nuclear Power Plants
A I 1993
Order No. AX2890
$175.00

OM-S/G-1997
Standards and Guides for Operation and Maintenance of Nuclear Power Plants
A I 2000
Order No. A13197
$175.00

OM-S/G-1994
Standards and Guides for Operation and Maintenance of Nuclear Power Plants
A I 1998
Order No. A13194
$175.00

QME-1-1997
Qualification of Active Mechanical Equipment Used in Nuclear Power Plants
A I L 2000
Order No. A13697
$105.00

OFFSHORE
Safety standards for quality assurance programs used in the manufacturing, remanufacturing, and assembly of valves for offshore service.

SPPE-1-1994
Quality Assurance and Certification of Safety and Pollution Prevention Equipment Used in Offshore Oil and Gas Operations
A L
Order No. A09394
$43.00

SPPE-2-1994
Accreditation of Testing Laboratories For Safety and Pollution Prevention Equipment Used In Offshore Oil and Gas Operations
A L
Order No. A09494
$45.00

OPERATOR QUALIFICATION AND CERTIFICATION

QHO-1-1994
Standard for the Qualification and Certification of Hazardous Waste Incinerator Operators
A I L 1999
Order No. A13894
$41.00

QMO-1-1993
Standard for the Qualification and Certification of Medical Waste Incinerator Operators
A I L 1998
Order No. A13393
$47.00

QRO-1-1994
Standard for the Qualification and Certification of Resource Recovery Facility Operators
A I L 1997
Order No. A13094
$49.00

PALLETS
Standards for terminology, sizes, and testing of pallets.

MH1-1997
Pallets, Slip Sheets, and Other Bases for Unit Loads
A 2000
Order No. K14097
$85.00

MH1.2.2M-1989(R1996)
Pallet Sizes
Order No. K08289
$32.00

MH1.5M-1993
Slip Sheets
Order No. K10393
$32.00

MH1.8M-1996
Wood Pallets
Order No. K13496
$35.00

MH1.9-1993
Export Pallets
Order No. K13393
$29.00

PERFORMANCE TEST CODES
ASME Performance Test Codes are applicable to the determination of performance of specific mechanical equipment that is designed to meet specified criteria for performance and operability. The codes provide requirements for instrumentation, test procedure, test environment, and interpretation of test results.

PTC 1-1991
General Instructions
Order No. C01791
$37.00

PTC 2-1980(R1985)
Code on Definitions and Values
Order No. C00007
$35.00

PTC 3.1-1958(R1992)
Diesel and Burner Fuels
Order No. D00024
$37.00

PTC 3.2-1990
Coal and Coke
Order No. C02390
$63.00

PTC 3.3-1969(R1992)
Gaseous Fuels
Order No. D00038
$37.00

PTC 4.1-1964(R1991)
Steam-Generating Units (with 1968 and 1969 Addenda)
Order No. C00025
$42.00

Diagram for Testing of a Steam Generator, Figure 1 (Pad of 100)
Order No. C00038
$23.00

Heat Balance of a Steam Generator, Figure 2 (Pad of 100)
Order No. C00039
$23.00

PTC 4.1a-1964
ASME Test Form for Abbreviated Efficiency Test—Summary Sheet (Pad of 100)
Order No. C00036
$23.00

PTC 4.1b-1964(R1965)
ASME Test for Abbreviated Efficiency Test—Calculation Sheet (Pad of 100)
Order No. C00037
$23.00

PTC 4.2 1969(R1997)
Coal Pulverizers
Order No. C00006
$37.00

PTC 4.3-1968(R1991)
Air Heaters
Order No. D00034
$35.00

PTC 4.4-1981(R1992)
Gas Turbine Heat Recovery Steam Generators
Order No. D00043
$42.00

PTC 5-1949
Reciprocating Steam Engines
Order No. C00022
$32.00

PTC 6-1996
Steam Turbines
Order No. C02896
$95.00

PTC 6A-1982(R1995)
Appendix A to Test Code for Steam Turbines
Order No. C00029
$39.00

PTC 6 on Steam Turbines—Interpretations
1977–1983
Order No. C00051
$32.00

PTC 6 Report 1985 (R1997)
Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines
A I 1991
Order No. D04186
$52.00

PTC 6S-1988(R1995)
Procedures for Routine Performance Test of Steam Turbines
I 1992
Order No. D04288
$74.00

PERFORMANCE TEST CODES
cont'd

PTC 7.1-1962(R1969)
Displacement Pumps
Order No. D00026
$32.00

PTC 8.2-1990
Centrifugal Pumps
I 1994
Order No. C00590
$49.00

PTC 9-1970(R1992)
Displacement Compressors, Vacuum Pumps and Blowers (With 1972 Errata)
Order No. CO0009
$34.00

PTC 10-1965(R1992)
Compressors and Exhausters
Order No. C00004
$25.00

PTC 11-1984 (R1995)
Fans
Order No. C05284
$54.00

PTC 12.1-1978 (R1987)
Closed Feedwater Heaters
Order No. C00013
$32.00

PTC 12.2-1998
Steam Surface Condensers
Order No. C02698
$150.00

PTC 12.3-1997
Deaerators
Order No. D02397
$49.00

PTC 12.4-1992(R1997)
Moisture Separator Reheaters
Order No. C05992
$59.00

PTC 17-1973(R1997)
Reciprocating Internal-Combustion Engines
Order No. C00020
$32.00

PTC 18-1992
Hydraulic Prime Movers
Order No. C01892
$44.00

PTC 18.1-1978 (R1984)
Pumping Mode of Pump/Turbines
Order No. C00044
$32.00

PTC 19.1-1985 (R1990)
Instruments and Apparatus: Part 1 Measurement Uncertainty
I 1991
Order No. D04585
$55.00

PTC 19.2-1987
Instruments and Apparatus: Part 2 Pressure Measurement
A I 1992
Order No. D02987
$74.00

PTC 19.3-1974 (R1986)
Temperature Measurement
Order No. C00035
$38.00

PTC 19.5-1972
Application, Part II of Fluid Meters: Interim Supplement on Instruments and Apparatus
Order No. G00018
$23.00

PTC 19.5.1-1964
Weighing Scales
Order No. D00028
$32.00

PTC 19.7-1980 (R1988)
Measurement of Shaft Power
Order No. D00009
$32.00

PTC 19.8-1970 (R1985)
Measurement of Indicated Power
Order No. D00008
$32.00

PTC 19.10-1981
Part 10 Flue and Exhaust Gas Analyses
Order No. C00031
$43.00

PTC 19.11-1997
Steam and Water Sampling, Conditioning, and Analysis in the Power Cycle
Order No. D01197
$75.00

PTC 19.14-1958
Linear Measurements
Order No. D00014
$32.00

PTC 19.22-1986
Digital Systems Techniques
I 1991
Order No. C05486
$50.00

PTC 19.23-1980 (R1985)
Part 23 Guidance Manual for Model Testing
Order No. C00047
$35.00

PTC 20.1-1977 (R1988)
Speed and Load-Governing Systems for Steam Turbine-Generator Units
Order No. C00019
$32.00

PTC 20.2-1965 (R1986)
Overspeed Trip Systems for Steam Turbine-Generator Units
Order No. C00041
$32.00

PTC 20.3-1970 (R1986)
Pressure Control Systems Used on Steam Turbine Generator Units
Order No. C00027
$32.00

PTC 21-1991
Particulate Matter Collection Equipment
I 1996
Order No. D02191
$45.00

PTC 22-1997
Performance Test Code on Gas Turbines
I 2002
Order No. C01597
$75.00

PTC 23-1986(R1997)
Atmospheric Water Cooling Equipment
A 1992
Order No. C05586
$50.00

PTC 24-1976(R1982)
Ejectors
Order No. C00010
$32.00

PTC 25-1994
Pressure Relief Devices
Order No. C06194
$54.00

PERFORMANCE TEST CODES
cont'd

PTC 26-1962
Speed Governing Systems for Internal Combustion Engine Generator Units
Order No. D00027
$32.00

PTC 28-1965(R1985)
Determining the Properties of Fine Particulate Matter
Order No. D00030
$32.00

PTC 29-1965(R1985)
Speed-Governing Systems for Hydraulic Turbine-Generator Units
Order No. C00040
$32.00

PTC 30-1991(R1998)
Air Cooled Heat Exchangers
A I 1996
Order No. CO0057
$95.00

PTC 31-1972(R1991)
Ion Exchange Equipment
Order No. C00016
$32.00

PTC 32.1-1969 (R1992)
Nuclear Steam Supply Systems
Order No. C00012
$32.00

PTC 32.2 Report—1979(R1992)
Methods of Measuring the Performance of Nuclear Reactor Fuel in Light Water Reactors
Order No. C00045
$23.00

PTC 33-1978(R1991)
Large Incinerators
Order No. CX0046
$46.00

PTC 33a-1980(R1991)
Appendix to PTC 33-1978
Order No. C0046A
$23.00

ASME Form for Abbreviated Incinerator Efficiency Test
{Form PTC 33a-1980(R1991)}
Order No. C0046F
$23.00

PTC 36-1985
Measurement of Industrial Sound
Order No. C00053
$32.00

PTC 38-1980(R1985)
Determining the Concentration of Particulate Matter in a Gas Stream
Order No. C00049
$35.00

PTC 39.1-1980 (R1991)
Condensate Removal Devices for Steam Systems
Order No. C00048
$32.00

PTC 40-1991
Flue Gas Desulfurization Units
I 1996
Order No. CO5891
$49.00

PTC 42-1988
Wind Turbines
I 1993
Order No. CO5688
$33.00

PTC 46-1997
Overall Plant Performance
Order No. C06496
$250.00

PTC PM-1993
Performance Monitoring Guidelines for Steam Power Plants
Order No. C06093
$73.00

PIPING
Standards for the identification, design, and welding of piping.

A13.1-1996
Scheme for the Identification of Piping Systems
Order No. L03396
$30.00

B31G-1991
Manual for Determining the Remaining Strength of Corroded Pipelines: A Supplement to B31, Code for Pressure Piping
Order No. A12191
$30.00

B31.1-1998
Power Piping
A I C 2001
Order No. A05898
$164.00

B31.1-1995
Power Piping
A I 1998
Order No. A05895
$140.00

B31.2-1968
Fuel Gas Piping
Order No. A00036
$29.00

B31.3-1996
Process Piping
A I 1999
Order No. A03796
$220.00

B31.3-1993
Chemical Plant and Refinery Piping, Plus Addenda
A I
Order No. A03793
$189.00

B31.4-1992
Pipeline Transportion Systems for Liquid Hydrocarbons and Other Liquids
A I C 1995
Order No. A03892
$72.00

B31.4-1989
Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids
A I C 1992
Order No. A03889
$66.00

B31.5-1992
Refrigeration Piping
A I C 1997
Order No. A04092
$61.00

B31.8-1995
Gas Transmission and Distribution Systems
A I 1998
Order No. A04595
$103.00

B31.9-1996
Building Services Piping
A I 1999
Order No. A11696
$72.00

B31.11-1989(R1998)
Slurry Transportation Piping Systems
A I C 1994
Order No. AI2489
$79.00

PLUMBING
Standards for plumbing equipment.

A112.1.2-1991(R1998)
Air Gaps in Plumbing Systems
Order No. J04191
$29.00

A112.3.1-1993
Performance Standard and Installation Procedures for Stainless Steel Drainage Systems, for Sanitary, Storm, and Chemical Applications, Above and Below Ground
Order No. J13693
$32.00

A112.4.1-1993
Water Heater Relief Valve Drain Tubes
Order No. J13793
$28.00

A112.6.1M-1997
Floor-Affixed Supports for Off-the-Floor Plumbing Fixtures for Public Use
Order No. J04397
$34.00

A112.14.1-1975 (R1998)
Backwater Valves
Order No. J00035
$29.00

A112.18.1M-1996
Plumbing Fixture Fittings
Order No. J02996
$33.00

A112.18.3M-1996
Performance Requirements for Backflow Devices and Systems in Plumbing Fixture Fittings
Order No. J14196
$31.00

A112.19.1M-1994
Enameled Cast Iron Plumbing Fixtures
Order No. J01194
$33.00

A112.19.2M-1995
Vitreous China Plumbing Fixtures
Order No. J01295
$42.00

A112.19.3M-1987 (R1996)
Stainless Steel Plumbing Fixtures (Designed for Residential Use)
Order No. J00050
$30.00

A112.19.4M-1994
Porcelain Enameled Formed Steel Plumbing Fixtures
Order No. J06494
$33.00

A112.19.5-1979 (R1998)
Trim for Water-Closet Bowls, Tanks and Urinals
Order No. J00058
$29.00

A112.19.6-1995
Hydraulic Requirements for Water Closets and Urinals
Order No. J13095
$32.00

A112.19.7M-1995
Whirlpool Bathtub Appliances
Order No. J11995
$28.00

A112.19.8M-1987 (1996)
Suction Fittings for Use in Swimming Pools, Wading Pools, Spas, Hot Tubs, and Whirlpool Bathtub Appliances
Order No. J00118
$29.00

A112.19.9M-1991 (R1998)
Non-Vitreous Ceramic Plumbing Fixtures
Order No. J13291
$29.00

A112.19.10-1994
Dual Flush Devices for Water Closets
Order No. J13894
$29.00

A112.21.1M-1991 (R1998)
Floor Drains
Order No. J04090
$30.00

A112.21.2M-1983
Roof Drains
Order No. J00002
$29.00

A112.21.3M-1985 (R1995)
Hydrants for Utility and Maintenance Use
Order No. J00051
$29.00

A112.36.2M-1991 (R1998)
Cleanouts
Order No. J04591
$31.00

POWERED PLATFORMS
Standards that establish safety requirements for powered platforms.

A120.1-96
Powered Platforms for Building Maintenance
A I 1999
Order No. A14296
$44.00

PRESSURE VESSELS

Boiler and Pressure Vessel Code
(see page 23)

BPE-1997
Bioprocessing Equipment
A I 2000
Order No. A14397
$165.00

PVHO-1-1997
Safety Standard for Pressure Vessels for Human Occupancy
A I 2000
Order No. A09297
$125.00

PVHO-1-1993
Safety Standard for Pressure Vessels for Human Occupancy
Order No. A09293
$125.00

PUMPS
Standards on pumps for chemical process service.

B73.1M-1991
Specification for Horizontal End Suction Centrifugal Pumps for Chemical Process
Order No. J01991
$29.00

B73.2M-1991
Specification for Vertical In-Line Centrifugal Pumps for Chemical Process
Order No. J04991
$29.00

B73.3M-1997
Specification for Sealless Horizontal End Suction Centrifugal Pumps for Chemical Process
A I 2002
Order No. J14597
$45.00

B73.5M-1995
Thermoplastic and Thermoset Polymer Material Horizontal End Suction Centrifugal Pumps for Chemical Process
Order No. J13995
$35.00

REINFORCED THERMOSET PLASTIC CORROSION RESISTANT EQUIPMENT
Rules of safety governing the design, fabrication, and inspection during construction of reinforced thermoset plastic equipment.

RTP-1-1995
Reinforced Thermoset Plastic Corrosion Resistant Equipment
A I L 2000
Order No. J12795
$139.00

RTP-1-1992
Reinforced Thermoset Plastic Corrosion Resistant Equipment
Order No. J12792
$139.00

SCREW THREADS
Standards that cover dimensional data.

B1.1-1989
Unified Inch Screw Threads (UN and UNR Thread Form)
Order No. M02889
$49.00

B1.2-1983 (R1991)
Gages and Gaging for Unified Inch Screw Threads
Order No. N00058
$43.00

B1.3M-1992
Screw Thread Gaging Systems for Dimensional Acceptability—Inch and Metric Screw Threads (UN, UNR, UNJ, M, and MJ)
Order No. N09692
$32.00

B1.5-1988(R1994)
Acme Screw Threads
Order No. N02788
$45.00

B1.7M-1984(R1992)
Nomenclature, Definitions, and Letter Symbols for Screw Threads
Order No. L00011
$32.00

B1.8-1988(R1994)
Stub Acme Screw Threads
Order No. N02888
$34.00

B1.9-1973(R1992)
Buttress Inch Screw Threads 7°/45° Form with 0.6 Pitch Basic Height of Thread Engagement
Order No. N00029
$32.00

B1.10M-1997
Unified Miniature Screw Threads
Order No. N02597
$32.00

B1.11-1958(R1994)
Microscope Objective Thread
Order No. N00020
$32.00

B1.12-1987(R1992)
Class 5 Interference-Fit Thread
Order No. N00032
$40.00

B1.13M-1995
Metric Screw Threads—M Profile
Order No. N04695
$37.00

B1.15-1995
Unified Inch Screw Threads (UNJ Thread Form)
Order No. N13195
$44.00

B1.16M-1984(R1992)
Gages and Gaging for Metric M Screw Threads
Order No. N00026
$44.00

B1.20.1-1983(R1992)
Pipe Threads, General Purpose (Inch)
Order No. N00103
$32.00

B1.20.3-1976(R1991)
Dryseal Pipe Threads (Inch)
Order No. N00076
$32.00

B1.20.5-1991
Gaging for Dryseal Pipe Threads (Inch)
Order No. N09290
$32.00

B1.20.7-1991
Hose Coupling Screw Threads (Inch)
Order No. K01491
$32.00

B1.21M-1997
Metric Screw Threads: MJ Profile
Order No. N09097
$39.00

B1.22M-1985(R1992)
Gages and Gaging for MJ Series Metric Screw Threads
Order No. N00086
$50.00

B1.30-1992
Screw Threads—Standard Practice for Calculating and Rounding Dimensions
Order No. N12692
$32.00

B1 Technical Report: Measurement Uncertainty for 60° Screw Thread Gage Element (Inch and Metric)
Order No. N12893
$29.00

STEEL STACKS
Standard that specifies design, construction, and recommendations for inspection.

STS-1-1992
Steel Stacks
Order No. L06992
$72.00

STORAGE TANKS
Standard for the design and fabrication of welded aluminum-alloy storage tanks.

B96.1-1993
Welded Aluminum-Alloy Storage Tanks
A I 1998
Order No. A12693
$69.00

SURFACE QUALITY
Standard for classification of surfaces.

B46.1-1995
Surface Texture (Surface Roughness, Waviness and Lay)
Order No. M01995
$69.00

TOOLS
Standards for machine tools, consumable tools, and hand tools.

Machine Tools

B5.1M-1985(R1992)
T-Slots, Their Bolts, Nuts and Tongues
Order No. M00024
$30.00

B5.5-1959(R1992)
Rotating Air Cylinders and Adapters
Order No. L00031
$30.00

B5.8-1972(R1994)
Chucks and Chuck Jaws
Order No. J00024
$30.00

B5.9-1967(R1992)
Spindle Noses for Tool Room Lathes, Engine Lathes, Turret Lathes and Automatic Lathes
Order No. M00010
$30.00

B5.10-1994
Machine Tapers
Order No. L00494
$35.00

B5.11-1964(R1994)
Spindle Noses and Adjustable Adaptors for Multiple Spindle Drilling Heads
Order No. M00008
$30.00

B5.16-1952(R1992)
Accuracy of Engine and Tool Room Lathes
Order No. J00004
$30.00

B5.18-1972(R1998)
Spindle Noses and Tool Shanks for Milling Machines
Order No. M00009
$30.00

B5.25-1978(R1992)
Punch and Die Sets
Order No. L00027
$30.00

B5.25M-1980(R1992)
Punch and Die Sets (Metric)
Order No. L00061
$30.00

B5.28-1971(R1994)
Mounting Dimensions of Lubricating and Coolant Pumps for Machine Tools
Order No. L00009
$30.00

B5.32-1977(R1994)
Grinding Machines, Surface, Reciprocating Table-Horizontal Spindle
Order No. J00028
$30.00

B5.32.1-1977(R1994)
Grinding Machines, Surface, Reciprocating Table-Vertical Spindle
Order No. J00053
$30.00

B5.33-1981(R1994)
External Cylindrical Grinding Machines—Pain
Order No. M00038
$30.00

B5.35-1983(R1994)
Machine Mounting Specifications for Abrasive Discs and Plate Mounted Wheels
Order No. M00035
$32.00

B5.37-1970(R1994)
External Cylindrical Grinding Machines—Centerless
Order No. M00039
$30.00

B5.40-1977(R1991)
Spindle Noses and Tool Shanks for Horizontal Boring Machines
Order No. M00048
$30.00

B5.42-1981(R1994)
External Cylindrical Grinding Machines Universal
Order No. M00040
$30.00

B5.43-1977(R1994)
Modular Machine Tool Standards
Order No. M00021
$30.00

B5.43M-1979(R1994)
Modular Machine Tool Standards (Metric)
Order No. M00112
$30.00

B5.44-1971(R1992)
Rotary Table Surface Grinding Machines
Order No. M00061
$30.00

B5.45-1972(R1991)
Milling Machines
Order No. M00067
$30.00

B5.46-1972(R1994)
Symbols for Machine Tool Indicator Plates
Order No. M00065
$30.00

B5.47-1972(R1991)
Milling Machine Arbor Assemblies
Order No. M00074
$30.00

B5.48-1977(R1994)
Ball Screws
Order No. M00089
$30.00

B5.49-1998
Glossary of Power Press Terms
Order No. M09098
$32.00

B5.50-1994
"V" Flange Tool Shanks for Machining Centers with Automatic Tool Changers
Order No. M09794
$29.00

B5.51M-1979(R1994)
Preferred SI Units for Machine Tools
Order No. M00099
$30.00

B5.52M-1980(R1994)
Mechanical Power Presses General Purpose Single Point, GAP Type (Metric)
Order No. M00120
$38.00

B5.53M-1982(R1994)
Cutter and Tool Grinding Machines
Order No. M00129
$30.00

B5.54M-1992
Methods for Performance Evaluation of Computer Numerically Controlled Machine Centers
Order No. M14492
$52.00

TOOLS
cont'd

B5.55M-1994
Specification and Performance Standard, Power Press Brakes
Order No. M15194
$30.00

B5.56M-1994
Specification and Performance Standard, Power Shears
Order No. M15294
$30.00

Consumable Tools
B94.2-1995
Reamers
Order No. L02895
$35.00

B94.3-1965(R1995)
Straight Cut-Off Blades for Lathes and Screw Machines
Order No. M00056
$29.00

B94.6-1984(R1995)
Knurling
Order No. K00022
$29.00

B94.7-1980(R1995)
Hobs
Order No. N00056
$29.00

B94.8-1967(R1987)
Inserted Blade Milling Cutter Bodies
Order No. K00044
$29.00

B94.9-1987
Taps-Cut and Ground Threads
Order No. K04387
$40.00

B94.11M-1993
Twist Drills
Order No. M02793
$35.00

B94.14-1968(R1995)
Punches—Basic Head Type
Order No. K00052
$29.00

B94.14.1-1977(R1995)
Punches—Basic Head Type (Metric)
Order No. M00092
$29.00

B94.16-1987(R1995)
Retainers—Basic Ball-Lock, Punch and Die Button, Light and Heavy Duty
Order No. K05487
$29.00

B94.16.1-1978(R1995)
Retainers—Basic Ball-Lock, Punch and Die Button, Light and Heavy Duty (Metric)
Order No. M00095
$29.00

B94.17-1987(R1995)
Gages—Functional, Ball-Lock Punches, Die Buttons, and Retainers
Order No. K05787
$29.00

B94.17.1-1977(R1995)
Gages—Functional, Ball-Lock Punches, Die Buttons and Retainers (Metric)
Order No. M00093
$29.00

B94.18-1987(R1995)
Punches—Basic Ball-Lock, Light and Heavy Duty
Order No. K05587
$29.00

B94.18.1-1977(R1995)
Punches—Basic Ball-Lock Light and Heavy Duty (Metric)
Order No. M00087
$29.00

B94.19-1997
Milling Cutters and End Mills
Order No. L00897
$44.00

B94.21-1968(R1995)
Gear Shaper Cutters
Order No. J00007
$29.00

B94.22-1968(R1995)
Punches—Variable Head Type
Order No. L00041
$29.00

B94.22.1-1977(R1995)
Punches—Variable Head Type (Metric)
Order No. L00050
$29.00

B94.23-1969(R1995)
Punch Guide Bushings—Variable, Press Fit
Order No. L00040
$29.00

B94.27-1970(R1995)
Die Buttons—Basic Taper Relief, Press Fit
Order No. M00041
$29.00

B94.27.1M-1983 (R1995)
Die Buttons—Basic Taper Relief, Press Fit (Metric)
Order No. M00132
$29.00

B94.28-1970(R1995)
Die Buttons—Basic Straight Relief, Press Fit
Order No. M00045
$29.00

B94.28.1M-1984 (R1995)
Die Buttons—Basic Straight Relief, Press Fit (Metric)
Order No. M00134
$29.00

B94.29-1970(R1995)
Die Buttons—Basic Ball Lock
Order No. M00059
$29.00

B94.29.1-1977(R1995)
Die Buttons—Basic Ball-Lock (Metric)
Order No. M00088
$29.00

B94.30-1970(R1995)
Die Buttons—Variable, Press Fit
Order No. M00060
$29.00

B94.31-1981(R1995)
Steel Rotary Slitting Knives and Steel Spacing Collars
Order No. K00061
$29.00

B94.33-1996
Jig Bushings
Order No. K02196
$49.00

B94.35-1972(R1995)
Drill Drivers, Split-Sleeve, Collet Type
Order No. J00030
$29.00

B94.37M-1979(R1985)
Roller Turner Type Cutting Tools Single Point
Order No. M00081
$29.00

B94.38-1972(R1995)
Punches—Variable, Angle Head Type and Related Quill Bushings
Order No. M00068
$29.00

B94.39-1972(R1995)
Punches—Basic, Combination Angle Head Type and Related Quill Bushings
Order No. M00069
$29.00

B94.40-1972(R1995)
Punches—Wire Type
Order No. M00070
$29.00

B94.41-1972(R1995)
Punches—Basic, Angle Head Type and Related Quill Bushings
Order No. M00071
$29.00

B94.43-1972(R1995)
Die Buttons—Variable, Press Fit, Headless and Head Type, Step Relief
Order No. M00075
$29.00

TOOLS
cont'd

B94.44-1972(R1995)
Punches—Basic, Cylindrical Head Type and Related Quill Bushings
Order No. M00076
$29.00

B94.49-1975(R1995)
Spade Drill Blades and Spade Drill Holders
Order No. L00035
$29.00

B94.50-1975(R1995)
Basic Nomenclature and Definitions for Single-Point Cutting Tools
Order No. M00084
$29.00

B94.51M-1994
Specifications for Band Saw Blades (Metal Cutting)
Order No. M08594
$28.00

B94.52M-1994
Specifications for Hacksaw Blades
Order No. M09194
$29.00

B94.53-1978(R1984)
Solid Steel Rectangular Metal Cutting Squaring Shear Knives-Dimensional Tolerances
Order No. M00096
$29.00

B94.54-1994
Specifications for Hole Saws, Hole Saw Arbors, and Hole Saw Accessories
Order No. M13594
$30.00

B94.55M-1985(R1995)
Tool Life Testing with Single-Point Turning Tools
Order No. M00137
$32.00

B94.56-1995
Gages—Functional, Ball-Lock Punches, Inch
Order No. M15595
$30.00

B94.56.1M-1995
Gages—Functional, Ball-Lock Punches, Metric
Order No. M15395
$30.00

Hand Tools
B107.1-1993
Socket Wrenches, Hand (Inch Series)
Order No. N08893
$32.00

B107.2-1995
Socket Wrenches, Extensions, Adaptors, and Universal Joints, Power Drive (Impact) (Inch Series)
Order No. N01395
$32.00

B107.4M-1995
Driving and Spindle Ends for Portable Hand, Impact, Air, and Electric Tools (Percussion Tools Excluded)
Order No. N02395
$36.00

B107.5M-1994
Socket Wrenches, Hand (Metric Series)
Order No. N08294
$32.00

B107.6-1994
Wrenches, Box, Angled, Open End, Combination, Flare Nut, and Tappet (Inch Series)
Order No. N08494
$29.00

B107.8M-1996
Adjustable Wrenches
Order No. N08396
$29.00

B107.9M-1994
Wrenches, Box, Angled, Open End, Combination, Flare Nut, and Tappet (Metric Series)
Order No. N08794
$32.00

B107.10M-1996
Handles and Attachments for Hand Socket Wrenches—Inch and Metric Series
Order No. N10096
$29.00

B107.11M-1993
Pliers, Diagonal Cutting, and Nippers, End Cutting
Order No. N10493
$32.00

B107.12-1997
Nut Drivers (Spin Type, Screwdriver Grip) (Inch Series)
Order No. N10797
$29.00

B107.13M-1996
Pliers—Long Nose, Long Reach
Order No. N11196
$29.00

B107.14M-1994
Hand Torque Tools
Order No. N10994
$29.00

B107.15-1993
Flat Tip and Phillips Screwdrivers
Order No. N11693
$32.00

B107.16M-1998
Shears (Metal Cutting, Hand)
Order No. N11498
$32.00

B107.17M-1997
Gages, Wrench Openings, Reference
Order No. N11297
$32.00

B107.18M-1996
Pliers (Wire Twister)
Order No. N11396
$29.00

B107.19-1993
Pliers, Retaining Ring
Order No. N11893
$32.00

B107.20M-1992
Pliers (Lineman's, Iron Worker's, Gas, Glass, and Fence)
Order No. N11792
$29.00

B107.21-1992
Wrench, Crowfoot Attachments
Order No. N12591
$29.00

B107.22M-1991
Electronic Cutters and Pliers
Order No. N12391
$29.00

B107.23M-1997
Pliers, Multiple Position, Adjustable
Order No. N12797
$32.00

B107.25M-1996
Pliers—Performance Test Methods
Order No. N13596
$29.00

B107.27-1996
Pliers, Multiple Position (Electrical Connector)
Order No. N13696
$32.00

B107.28M-1997
Electronic Torque Instruments
Order No. N14397
$32.00

B107.29M-1998
Electronic Tester, Hand Torque Tools
Order No. N14698
$32.00

TOOLS
cont'd

B107.31M-1997
Screwdrivers, Cross Tip Gaging
Order No. N13997
$35.00

B107.34M-1997
Socket Wrenches for Spark Plugs
Order No. N14297
$32.00

B107.35M-1997
Nut Drivers (Spin Type Screwdriver Grip) (Metric Series)
Order No. N14097
$32.00

B107.41M-1997
Nail Hammers—Safety Requirements
Order No. N13797
$29.00

B107.42M-1997
Hatchets: Safety Requirements
Order No. N14197
$28.00

B107.43M-1998
Wood-Slipping Wedges: Safety Requirements
Order No. N14497
$32.00

B107.44M-1998
Glaziers' Chisels and Wood Chisels—Safety Requirements
Order No. N15198
$32.00

B107.45M-1998
Ripping Chisels and Flooring/Electricians' Chisels: Safety Requirements
Order No. N14598
$32.00

B107.46M-1998
Stud, Screw, and Pipe Extractors: Safety Requirements
Order No. N14798
$32.00

B107.52M-1998
Nail-Puller Bars: Safety Requirements
Order No. N15298
$28.00

B107.53M-1998
Ball Peen Hammers: Safety Requirements
Order No. N15398
$28.00

B107.55M-1998
Axes: Safety Requirements
Order No. N15098
$28.00

TURBINES
Standards that cover considerations for procurement of turbines.

B133.1M-1983(R1997)
Gas Turbine Terminology
Order No. K00093
$29.00

B133.2-1977(R1997)
Basic Gas Turbines
Order No. K00090
$29.00

B133.3-1981(R1994)
Procurement Standard for Gas Turbine Auxiliary Equipment
Order No. K00107
$29.00

B133.4-1978(R1997)
Gas Turbine Control and Protection Systems
Order No. K00091
$29.00

B133.5-1978(R1997)
Procurement Standard for Gas Turbine Electrical Equipment
Order No. K00096
$29.00

B133.6-1978(R1994)
Procurement Standard for Gas Turbine Ratings and Performance
Order No. K00098
$29.00

B133.7M-1985(R1992)
Gas Turbine Fuels
Order No. K00086
$29.00

B133.8-1977(R1989)
Gas Turbine Installation Sound Emissions
Order No. K00087
$29.00

B133.9-1994
Measurement of Exhaust Emissions from Stationary Gas Turbine Engines
Order No. K09994
$36.00

B133.10-1981(R1994)
Procurement Standard for Gas Turbine Information to be Supplied by User and Manufacturer
Order No. K00109
$29.00

B133.11-1982(R1994)
Procurement Standard for Gas Turbine Preparation for Shipping and Installation
Order No. K00108
$29.00

B133.12-1981(R1994)
Procurement Standard for Gas Turbine Maintenance and Safety
Order No. K00106
$29.00

B133.16-1978(R1997)
Procurement Standard for Gas Turbine Marine Applications
Order No. K00095
$29.00

VALVES, FITTINGS, FLANGES, GASKETS
Standards for pressure-temperature ratings, materials, dimensions, tolerances, markings, and testing of valves, fittings, flanges and gaskets.

B16.1-1989
Cast Iron Pipe Flanges and Flanged Fittings
A I 1994
Order No. J01789
$48.00

B16.3-1992
Malleable Iron Threaded Fittings
A I 1997
Order No. J10392
$52.00

B16.4-1992
Gray Iron Threaded Fittings
A I 1997
Order No. J02292
$45.00

B16.5-1996
Pipe Flanges and Flanged Fittings
A I 2001
Order No. M01596
$95.00

VALVES, FITTINGS, FLANGES, GASKETS
Standards for pressure-temperature ratings, materials, dimensions, tolerances, markings, and testing of valves, fittings, flanges, and gaskets.

B16.9-1993
Factory-Made Wrought Steel Buttwelding Fittings
A I 1998
Order No. J10993
$55.00

B16.9-1986
Factory-Made Wrought Steel Buttwelding Fittings
A I 1991
Order No. J10986
$40.00

B16.10-1992
Face-to-Face and End-to-End Dimensions of Valves
A I 1997
Order No. J03192
$52.00

B16.11-1996
Forged Steel Fittings, Socket-Welding and Threaded
A I 2001
Order No. M01696
$39.00

B16.12-1991
Cast Iron Threaded Drainage Fittings
A I 1996
Order No. J02191
$32.00

B16.14-1991
Ferrous Pipe Plugs, Bushings and Locknuts with Pipe Threads
A I 1996
Order No. J03290
$32.00

B16.15-1985(R1994)
Cast Bronze Threaded Fittings
A I 1990
Order No. J01085
$40.00

B16.18-1984(R1994)
Cast Copper Alloy Solder Joint Pressure Fittings
Order No. J07184
$34.00

B16.20-1993
Metallic Gaskets for Pipe Flanges—Ring Joint, Spiral-Wound, and Jacketed
Order No. L03093
$41.00

B16.21-1992
Nonmetallic Flat Gaskets for Pipe Flanges
Order No. L01592
$35.00

B16.22-1995
Wrought Copper and Copper Alloy Solder Joint Pressure Fittings
A I 2000
Order No. M03095
$32.00

B16.23-1992
Cast Copper Alloy Solder Joint Drainage Fittings (DWV)
A I 1995
Order No. J13492
$49.00

B16.24-1991(R1998)
Cast Copper Alloy Pipe Flanges and Flanged Fittings: Class 150, 300, 400, 600, 900, 1500 and 2500
A I 1996
Order No. J00991
$35.00

B16.25-1997
Buttwelding Ends
A I 2002
Order No. J01397
$42.00

B16.26-1988
Cast Copper Alloy Fittings for Flared Copper Tubes
A I 1993
Order No. J00888
$32.00

B16.28-1994
Wrought Steel Buttwelding Short Radius Elbows and Returns
A I 1999
Order No. J11094
$34.00

B16.28-1986
Wrought Steel Buttwelding Short Radius Elbows and Returns
A I 1991
Order No. J11086
$29.00

B16.29-1994
Wrought Copper and Wrought Copper Alloy Solder Joint Drainage Fittings—DWV
A I 1999
Order No. N05794
$30.00

B16.33-1990
Manually Operated Metallic Gas Valves for Use in Gas Piping Systems up to 125 psig (Sizes 1/2 Through 2)
A I 1995
Order No. J00690
$32.00

B16.34-1996
Valves—Flanged, Threaded, and Welding End
A I 2001
Order No. J01896
$79.00

B16.36-1996
Orifice Flanges
A I 2001
Order No. J04696
$42.00

B16.38-1985(R1994)
Large Metallic Valves for Gas Distribution (Manually Operated, NPS-2 1/2 to 12, 125 psig Maximum)
A I 1988
Order No. J05585
$32.00

B16.39-1986(R1994)
Malleable Iron Threaded Pipe Unions
A I 1991
Order No. L04986
$34.00

B16.40-1985(R1994)
Manually Operated Thermoplastic Gas Shutoffs and Valves in Gas Distribution Systems
A I 1988
Order No. J05485
$32.00

B16.42-1987(R1997)
Ductile Iron Pipe Flanges and Flanged Fittings, Classes 150 and 300
A I 1992
Order No. J06687
$40.00

B16.44-1995
Manually Operated Metallic Gas Valves for Use in House Piping Systems
A I 2000
Order No. J14095
$35.00

B16.45-1987(R1997)
Cast Iron Fittings for Sovent Drainage Systems
A I 1992
Order No. J11187
$32.00

B16.47-1996
Large Diameter Steel Flanges: NPS 26 Through NPS 60
A I 2001
Order No. J12996
$52.00

B16.48-1997
Steel Line Blanks
A I 2002
Order No. J14497
$42.00

1998 ASME

Boiler & Pressure Vessel Code

An International Code

The Boiler and Pressure Vessel Code establishes rules of safety governing the design, fabrication and inspection of boilers and pressure vessels, and nuclear power plant components during construction. The objective of the rules is to ensure reasonably certain protection of life and property and to provide a margin for deterioration in service. Advancements in design and material and the evidence of experience are constantly being added by addenda. Prices include update service through 2000.

3-year update service includes:

- Addenda issued once per year: July 1, 1999 and 2000.
- Supplements issued four times per year for each Code Cases book.
- Interpretations issued twice per year: with Addenda, in July 1998 and in December 1998, 1999, and 2000.

Note: Interpretations will not be available separately and are issued as part of the update service to the individual sections of the Code. Interpretations to Section III, Divisions 1 and 2 are issued as part of the update to Subsection NCA.

I Power Boilers
Order No. QX0010 *$210.00*

II Materials
Part A—Ferrous Material Specifications
Order No. QX002A *$385.00*
Part B—Nonferrous Material Specifications
Order No. QX002B *$350.00*
Part C—Specifications for Welding Rods, Electrodes, and Filler Metals
Order No. QX002C *$350.00*
Part D—Properties
Order No. QX002D *$315.00*

III Rules for Construction of Nuclear Power Plant Components
Subsection NCA
General Requirements for Division 1 and Division 2
Order No. QX003R *$240.00*
Division 1
Subsection NB
Class 1 Components
Order No. QX003B *$240.00*
Subsection NC
Class 2 Components
Order No. QX003C *$240.00*
Subsection ND
Class 3 Components
Order No. QX003D *$240.00*
Subsection NE
Class MC Components
Order No. QX003E *$240.00*
Subsection NF
Supports
Order No. QX003F *$185.00*
Subsection NG
Core Support Structures
Order No. QX003G *$185.00*
Subsection NH
Class 1 Components in Elevated Temperature Service
Order No. QX003H *$240.00*
Appendices
Order No. QX003A *$250.00*
Division 2
Code for Concrete Reactor Vessels and Containments
Order No. QX0032 *$295.00*
Division 3
Containment Systems and Transport Packagings for Spent Nuclear Fuel and High Level Radioactive Waste
Order No. QX0033 *$275.00*

IV Heating Boilers
Order No. QX0040 *$195.00*

V Nondestructive Examination
Order No. QX0050 *$215.00*

VI Recommended Rules for the Care and Operation of Heating Boilers
Order No. QX0060 *$125.00*

VII Recommended Guidelines for the Care of Power Boilers
Order No. QX0070 *$145.00*

VIII Pressure Vessels
Division 1
Order No. QX0081 *$385.00*
Division 2
Alternative Rules
Order No. QX0082 *$385.00*
Division 3
Alternative Rules for Construction of High Pressure Vessels
Order No. QX0083 *$295.00*

IX Welding and Brazing Qualifications
Order No. QX0090 *$215.00*

X Fiber-Reinforced Plastic Pressure Vessels
Order No. QX0100 *$185.00*

XI Rules for Inservice Inspection of Nuclear Power Plant Components
Order No. QX0011 *$370.00*

Code Cases: Boilers and Pressure Vessels
Order No. QX0120 *$275.00*

Code Cases: Nuclear Components
Order No. QX012N *$315.00*

Binder
Order No. QX0140 *$20.00*

Special Price for Complete Code
Order No. QX0230
$6,550.00

Complete Code Plus 30 Binders
Order No. QX0330
$7,150.00

TO ORDER:
Call toll free: 800-843-2763
Fax: 973-882-1717
Write: ASME
22 Law Drive
Box 2300
Fairfield, NJ 07007-2300
Email: infocentral@asme.org

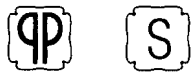
All orders require a valid purchase order. When ordering with your company purchase order, please make sure that your ASME account number, if available, is indicated. This account number can be found on ASME invoices, packing lists, and statements.

Prices subject to change without notice.
Shipping/handling charges added to every order.

Accreditation Packages

Code books required for use with ASME Code Symbol Stamps

Note: For books other than the Boiler and Pressure Vessel Code (B31.1, PTC 25.3, NQA-1), the required edition as of July 1, 1998 is listed. The specific effective Addenda will be referenced in the applicable Boiler and Pressure Vessel Code section. Later editions of these referenced books will be required if and when referenced by the applicable Boiler and Pressure Vessel Code section.

CODE SYMBOL STAMPS 

PP PRESSURE PIPING
S POWER BOILERS

CODE BOOKS REQUIRED

SECTION I—POWER BOILERS
SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties

SECTION V—NONDESTRUCTIVE EXAMINATION
SECTION IX—WELDING & BRAZING QUALIFICATIONS
B31.1-1995—POWER PIPING

****ORDER NO. BPVCA**
$2,204.00

CODE SYMBOL STAMP 

E ELECTRIC BOILERS

CODE BOOKS REQUIRED

SECTION I—POWER BOILERS
SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part D Properties

B31.1-1995—POWER PIPING

****ORDER NO. BPVCE**
$1,429.00

CODE SYMBOL STAMP 

U2 PRESSURE VESSELS (DIV. 2)

CODE BOOKS REQUIRED

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties

SECTION V—NONDESTRUCTIVE EXAMINATION
SECTION VIII—PRESSURE VESSELS, DIVISION 2-ALTERNATIVE RULES
SECTION IX—WELDING & BRAZING QUALIFICATIONS

ORDER NO. BPVU1
$2,215.00

CODE SYMBOL STAMP 

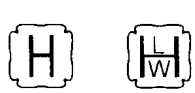
U3 PRESSURE VESSELS (DIV. 3)

CODE BOOKS REQUIRED

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties

SECTION V—NONDESTRUCTIVE EXAMINATION
SECTION VIII—PRESSURE VESSELS, DIVISION 3-ALTERNATIVE RULES FOR CONSTRUCTION OF HIGH PRESSURE VESSELS
SECTION IX—WELDING & BRAZING QUALIFICATIONS

ORDER NO. BPVU3
$2,125.00

CODE SYMBOL STAMPS 

H HEATING BOILERS EXCEPT CAST IRON
HLW LINED POTABLE WATER HEATERS

CODE BOOKS REQUIRED

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals

SECTION IV—HEATING BOILERS
SECTION IX—WELDING & BRAZING QUALIFICATIONS

ORDER NO. BPHLW
$1,495.00

CODE SYMBOL STAMP 

A ASSEMBLY OF POWER BOILERS

CODE BOOKS REQUIRED

SECTION I—POWER BOILERS
SECTION II—MATERIALS
Part C Specifications for Welding Rods, Electrodes and Filler Metals
SECTION V—NONDESTRUCTIVE EXAMINATION
SECTION IX—WELDING & BRAZING QUALIFICATIONS
B31.1-1995—POWER PIPING

****ORDER NO. BPVCB**
$1,154.00

CODE SYMBOL STAMPS  

U PRESSURE VESSELS (DIV. 1)
UM MINIATURE PRESSURE VESSELS

CODE BOOKS REQUIRED

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties
SECTION V—NONDESTRUCTIVE EXAMINATION
SECTION VIII—PRESSURE VESSELS, DIVISION 1
SECTION IX—WELDING & BRAZING QUALIFICATIONS

ORDER NO. BPVUM
$2,215.00

CODE SYMBOL STAMP 

RP FIBER-REINFORCED PLASTIC VESSELS

CODE BOOKS REQUIRED

SECTION X—FIBER-REINFORCED PLASTIC VESSELS

ORDER NO. BPVRP
$185.00

CODE SYMBOL STAMP 

H CAST IRON SECTIONAL HEATING BOILERS

CODE BOOK REQUIRED

SECTION IV—HEATING BOILERS

ORDER NO. BPVCH
$195.00

CODE SYMBOL STAMP 

H FIELD ASSEMBLY OF BOILERS

CODE BOOKS REQUIRED

SECTION IV—HEATING BOILERS
SECTION IX—WELDING & BRAZING QUALIFICATIONS

ORDER NO. BPVCH
$410.00

CODE SYMBOL STAMP 

UD RUPTURE DISK DEVICES

CODE BOOKS REQUIRED†

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties
SECTION VIII—PRESSURE VESSELS DIVISION 1
SECTION IX—WELDING & BRAZING QUALIFICATIONS
PTC 25-1994—PRESSURE RELIEF DEVICES

ORDER NO. BPUD
$2,049.00

CODE SYMBOL STAMP 

UV PRESSURE VESSELS SAFETY VALVES

CODE BOOKS REQUIRED†

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties
SECTION VIII—PRESSURE VESSELS DIVISION 1 OR DIVISION 2
SECTION IX—WELDING & BRAZING QUALIFICATIONS
PTC 25-1994—PRESSURE RELIEF DEVICES

ORDER NO. BPUV1
(INCLUDES SECTION VIII, DIV. 1)
$2,049.00
ORDER NO. BPUV2
(INCLUDES SECTION VIII, DIV. 2)
$2,049.00

CODE SYMBOL STAMP 

UV3 PRESSURE VESSELS SAFETY VALVES

CODE BOOKS REQUIRED†

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties

SECTION VIII—PRESSURE VESSELS DIVISION 3
SECTION IX—WELDING & BRAZING QUALIFICATIONS
PTC 25-1994—PRESSURE RELIEF DEVICES

ORDER NO. BPUV3
$1,959.00

CODE SYMBOL STAMP 

M MINIATURE BOILERS

CODE BOOKS REQUIRED

SECTION I—POWER BOILERS
SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties

SECTION IX—WELDING & BRAZING QUALIFICATIONS
B31.1-1995—POWER PIPING

****ORDER NO. BPVCM**
$1,989.00

CODE SYMBOL STAMP 

HV HEATING BOILER SAFETY VALVES

CODE BOOKS REQUIRED‡

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals

SECTION IV—HEATING BOILERS
SECTION IX—WELDING & BRAZING QUALIFICATIONS
PTC 25-1994—PRESSURE RELIEF DEVICES

ORDER NO. BPVHV
$1,544.00

CODE SYMBOL STAMP 

V BOILER SAFETY VALVES

CODE BOOKS REQUIRED†

SECTION I—POWER BOILERS
SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties

SECTION IX—WELDING & BRAZING QUALIFICATIONS
PTC 25-1994—PRESSURE RELIEF DEVICES

ORDER NO. BPVCV
$1,874.00

QUALITY SYSTEM CERTIFICATES
MATERIAL MANUFACTURERS, MATERIALS SUPPLIERS

CODE BOOKS REQUIRED

SECTION III—Rules for Construction of Nuclear Power Plant Components
Subsection NCA-General Requirements for Division 1 and Division 2
Appendices

SECTION II—Materials (Select applicable parts)
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties

ORDER NO. BPVMS (SEE PAGE 23 FOR PRICES)

Applicants should note that they may need one or more of the following subsections of Section III depending on the scope of their work to appear on the certificate. In addition, these subsections contain material requirements on Section 2000 which reference Section V and Section IX.

Subsection NB-Class 1 Components
Subsection NC-Class 2 Components
Subsection ND-Class 3 Components
Subsection NE-Class MC Components
Subsection NF-Supports
Subsection NG-Core Support Structures
Subsection NH-Class 1 Components in Elevated Temperature Service
SECTION V—Nondestructive Examination
SECTION IX—Welding and Brazing Qualifications

(SEE PAGE 23 FOR ORDER NUMBERS AND PRICES)

CODE SYMBOL STAMPS   

N TYPE CERTIFICATES OF AUTHORIZATION AND CERTIFICATES OF ACCREDITATION N, NA, NPT

CODE BOOKS REQUIRED

SECTION II—MATERIALS
Part D Properties (N and NA only)
SECTION III—RULES FOR CONSTRUCTION OF NUCLEAR POWER PLANT COMPONENTS
SUBSECTION NCA-GENERAL REQUIREMENTS FOR DIVISION 1 AND DIVISION 2 (NOT REQUIRED FOR SECTION III-3)
APPENDICES
SECTION V—NONDESTRUCTIVE EXAMINATION
SECTION IX—WELDING & BRAZING QUALIFICATIONS
NQA-1-1989—QUALITY ASSURANCE PROGRAM REQUIREMENTS FOR NUCLEAR FACILITIES

ORDER NO. BPNPT (SEE PAGE 23 FOR PRICES)

Applicants should note that they may need one or more of the following Parts of Section II and Subsections of Section III depending on the scope of their work to appear on the certificate.

SECTION II—MATERIALS
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part C Specifications for Welding Rods, Electrodes and Filler Metals
Part D Properties (NPT only)

Subsection NB-Class 1 Components
Subsection NC-Class 2 Components
Subsection ND-Class 3 Components
Subsection NE-Class MC Components
Subsection NF-Supports
Subsection NG-Core Support Structures
Subsection NH-Class 1 Components in Elevated Temperature Service
Division 2: Code for Concrete Reactor Vessels and Containments
Division 3: Containment Systems and Transport Packaging for Spent Nuclear Fuel and High Level Radioactive Waste

(SEE PAGE 23 FOR ORDER NUMBERS AND PRICES)

CODE SYMBOL STAMP 

NV NUCLEAR SAFETY AND PRESSURE RELIEF VALVES

CODE BOOKS REQUIRED

SECTION III—RULES FOR CONSTRUCTION OF NUCLEAR POWER PLANT COMPONENTS
SUBSECTION NCA-GENERAL REQUIREMENTS FOR DIVISION 1 AND DIVISION 2
APPENDICES
SECTION II—MATERIALS (SELECT APPLICABLE PARTS)
Part A Ferrous Material Specifications
Part B Nonferrous Material Specifications
Part D Properties

SECTION V—NONDESTRUCTIVE EXAMINATION
SECTION IX—WELDING & BRAZING QUALIFICATIONS
PTC 25-1994—PRESSURE RELIEF DEVICES
NQA-1-1989—QUALITY ASSURANCE PROGRAM REQUIREMENTS FOR NUCLEAR FACILITIES

ORDER NO. BPNVA (WITH SECTION II, PART A)
ORDER NO. BPNVB (WITH SECTION II, PART B)
ORDER NO. BPNVD (WITH SECTION II, PART D)
(SEE PAGE 23 FOR PRICES)

Applicants should note that they may need one or more of the following subsections of Section III depending on the scope of their work to appear on the certificate.

Subsection NB-Class 1 Components
Subsection NC-Class 2 Components
Subsection ND-Class 3 Components
Subsection NH-Class 1 Components in Elevated Temperature Service

(SEE PAGE 23 FOR ORDER NUMBERS AND PRICES)

** Packages marked with two asterisks include the 1998 edition of B31.1.
† Sections II and IX are not required for assemblers. Section II, Part C and Section IX are not required for manufacturers if welding and brazing are not within the scope of their work.
‡ Section II, Part C and Section IX are not required for manufacturers if welding and brazing are not within the scope of their work.

ASME 9000

ASME International's ISO 9000 Registration Program

The American Society of Mechanical Engineers has been a world leader in codes, standards, accreditation, and certification for over a century. These programs have now been extended to include the registration (certification) of quality systems in conformance with the standards set by the International Organization of Standardization (ISO).

ASME's ISO 9000 Registration Program is accredited by the American National Accreditation Program for Registrars of Quality Systems (ANSI-RAB) and the Dutch Council for Accreditation (RvA).

ASME specializes in the registration of domestic and international suppliers of mechanical equipment and related materials, items, and services in the industries and sectors associated with the art, science, and practice of mechanical engineering. These industries include:

- Primary metals industries
- Fabricated metal products
- Industrial and commercial machinery and equipment
- Reinforced thermoset plastic tanks and vessels
- Engineering services

ASME uses quality system auditors experienced in the above-listed fields who have been certified to the international auditor qualification standard ISO 10011-2. Registration decision and policy making are conducted within the framework of the ASME codes, standards, accreditation, and certification committee structure. The use of volunteers from industry and government provides for a high quality program that is administered in a fair, impartial, and professional manner.

Over 4,000 companies from 57 countries are currently accredited by ASME in one or more of the following quality assurance based programs:

- Authorized Inspection Agency (AIA)
- Boiler and Pressure Vessel
- Fastener Manufacturers and Distributors (FAP)
- Nuclear Components
- Nuclear Materials (MM & MS)
- Qualification of Elevator Inspectors (QEI)
- Reinforced Thermoset Plastic Corrosion Resistant Vessel Fabricators (RTP)
- Safety and Pollution Prevention Equipment (SPPE)

Attention all ASME Boiler & Pressure Vessel Certificate Holders!

Reduce assessment fees and achieve consistency by combining your upcoming shop review with an ISO 9000 assessment. ASME will utilize Auditors who are both qualified as BPV Team Leaders and ISO 9000 Assessors. Capitalize on this unique opportunity now—contact ASME.

For additional information write to:

ASME Conformity Assessment
Three Park Avneue
New York, NY 10016-5990
or call Raj Manchanda at 212-591-8033.
Fax 212-591-8599.
E-Mail: accreditation@asme.org

Asme Certification Programs

ASME administers four certification programs for technical personnel based on ASME standards that define the necessary qualifications of the individuals. One additional program is in development.

Operators of Resource Recovery Facilities Processing Municipal Solid Waste (MWCs)

- Applies to persons who hold, or wish to hold, the positions of Shift Supervisor or Chief Facility Operator
- Candidates are required to meet minimum education and experience requirements and pass a written multiple-choice examination for provisional certification
- Full certification is obtained through a site-specific oral examination

Operators of Medical Waste Incinerators (MWIs)

- Applies to persons who hold, or wish to hold, the positions of Operator or Operator Supervisor
- Candidates are required to demonstrate proficiency in the operation of the equipment and pass a written multiple-choice examination
- "Special Class" operator certification also available for persons in related fields (regulatory personnel, trainers, equipment manufacturers, etc.)

Operators of Hazardous Waste Incinerators

- Applies to Control Room Operators, Field Operators (waste receiving and handling, field equipment operation, and residual handling), and Supervisors
- Candidates are required to complete training in safety, combustion fundamentals, and basic operation, and pass a written multiple-choice examination for provisional certification
- Full certification is obtained through a site-specific oral examination in development

Geometric Dimensioning and Tolerancing Professionals Certification Exam

To certify your competence in using the Geometric Dimensioning and Tolerancing Y14.5 Standard, register now for the new certification exam sponsored by ASME. You can obtain certification on one of the following two levels:

- **Technologist GDTP**, which provides objective measure of your ability to understand drawings prepared using the Y14.5 Standard
- **Senior GDTP**, which provides additional objective measure of your ability to select geometric controls and apply them properly to drawings

Exams are given in various cities. Please contact ASME for the exam location nearest you. To obtain an exam application and full details, return this card, or
Phone: Raj Manchanda at 212-591-8033
Fax: 212-591-8599
E-mail: accreditation@asme.org
Or visit the Codes & Standards area of the ASME Web site: http://www.asme.org

- Candidates will be required to pass a written multiple-choice examination based on the ASME Y14.5 Standard

Operators of High Capacity Fossil Fuel Fired Plants (in development)

- Applies to operators of oil, gas, and coal fired boilers with heat input greater than 10,000,000 btu/h
- Certification criteria will include evaluation of qualifications in both safety and air emissions Geometric Dimensioning and Tolerancing Professionals
- Applies to designers, drafters, manufacturing engineers, CAD/CAM users, and others who need to understand the meaning of symbols, modifiers, and relationships of GD&T as applied to engineering drawings and related documents

For examination schedules and additional information, contact:
ASME Conformity Assessment
Three Park Avenue
New York, NY 10016-5990
Tel: 212-591-8591
Fax: 212-591-8599
E-Mail: accreditation@asme.org

Asme A17 CD-ROM for Elevators and Escalators

The ASME A17 CD-ROM for Elevators and Escalators features access to the current A17 collection from your desktop:

- ASME A17.1-1996 Safety Code for Elevators and Escalators
- ASME A17.2.1-1996 Inspectors' Manual for Electric Elevators
- ASME A17.2.2-1994 (through A17.2.2b-1996) Inspectors' Manual for Hydraulic Elevators
- ASME A17.2.3-1994 (through A17.2.3a-1996) Inspectors' Manual for Escalators and Moving Walks
- ASME A17.3-1996 Safety Code for Existing Elevators and Escalators
- INTERPRETATIONS for A17.1, A17.2.2., and A17.3
- Graphic images of all equations, figures, and tables assure fidelity of all visual information.

Sophisticated search capacities . . . navigational ease . . . custom annotation . . . customer support

The A17 CD-ROM gives you the freedom to move through the Code books unconstrained by pages (for your convenience, page numbers that match those of the print publications are provided). The clean, clear visual design inherits its colors from your own Windows choices. The combination of browsing and focused searching capabilities allows you to thoroughly and quickly investigate the *ASME Elevator and Escalator Code*.

Full Test Browser delivers compact, easily accessible electronic information, built from the same data that created the printed ASME A17 series books. It's simple to move through the collection, scanning the table of contents at any level of detail—from book titles to chapter titles to designated paragraphs. View the full table of contents or list tables or figures in the Table of Contents view. Jump to selected items with a click. The full text of the publications can also be read start to finish, more like a traditional book.

Order No.	Description	Price
A14401	Stand Alone License	$495
A14405	Network License, 2–5 users	$1,100
A14410	Network License, 6–10 users	$1,500
(NA)	Network License, more than 10 users	*

*For more than 10 users, please call ASME Information Central (800-843-2763) for a price quote.

Prices include an automatic subscription to the next two updated A17 CD-ROM.

PC Requirements: *System requirements for installation on a stand-alone personal computer are:*

*386DX processor minimum, 486 or higher preferred * 8 MB RAM minimum, with 500 KB free conventional memory * hard drive with approximately 7 MB of free space for software installation and 1 MB of additional free space for temporary files during program operation * mouse * VGA or SVGA monitor (SVGA preferred) with compatible video card (high-performance card preferred) * CD-ROM drive compatible with Microsoft Compact Disc Extensions (MSCDEX) and ISO 9660 * Microsoft Compact Disc Extensions (MSCDEX), version 2.2 or higher * DOS version 5.0 or higher * Microsoft Windows, 3.1 or higher, running in 366 Enhanced mode; Windows 95.

The Boiler and Pressure Vessel Code on CD-ROM*

The ***Code*** on CD-ROM provides pinpoint searching accuracy. The power of the computer allows you to search the entire ***Code*** for the occurrence of a single word or a complex combination of terms. You may browse the database by Title or Table of Contents, or by a specially developed thesaurus of subject classifications.

Electronic delivery allows you to quickly and easily investigate the ***Code***. An especially convenient and time-saving feature called Hot Link allows you to skip from a particular point in the ***Code*** to referenced sections, tables, graphs, or equations and then back to the original text.

*For the purpose of ASME accreditation, electronic versions of the Boiler & Pressure Vessel Code do not replace the need for a single hard copy version.

The Worldwide Standards Service—ASME from IHS

Make your research time more productive by having ASME documents available at your desk whenever you need them. With the American Society of Mechanical Engineers (ASME) documents available on CD-ROM you can view complete, full-text documents—exact images of the hardcopy originals—right at your own personal computer. The service contains the full text of all ASME Standards and Performance Test Codes. Included in every subscription is the comprehensive Worldwide Standards Service Index, which helps you locate standards by document number or by keyword. In addition to ASME documents, the Worldwide Standards Service Index contains document listings from approximately 400 of the world's leading standards developing bodies. The service is updated every 60 days to keep you abreast of all current revision activity.

For more information on the Boiler Code on CD-ROM or Worldwide Standards Service and free demo diskettes, call toll-free: 800-241-7824.

Information Handling Services®
15 Inverness Way East, P.O. Box 1154
Englewood, Colorado USA 80150
303-790-0600, ext. 59

Electronic Stress Tables

Software based on ASME boiler and pressure vessel code section II, Part D

ASME has good news for engineers who use the traditional methods of searching through stress tables. The new Electronic Stress Tables based on the *ASME Boiler and Pressure Vessel Code Section II, Part D (Materials Properties)* makes your search easier and faster. Designed to give you quick and accurate access to any or all BPVC Section II, Part D stress tables, the program provides a time-saving alternative to wading through eye-straining pages of spreadsheets. You'll find the software not only speeds your search, but also presents the information in a highly visual format, complete with notes and printing capability.

Search options

Fifteen different fields allow you to narrow your search criteria (filter), or reorder the data displayed (sort). The software allows you to create multiple line criteria for logical groupings and to add additional fields to create a compound sort.

Create Excel Spreadsheets

Import any or all of the tables into an Excel spreadsheet in a numeric format. You must have a copy of Excel installed on your PC to run this addition to the application.

Highly compatible

To get started, all you need is an IBM or 100% compatible CPU running Windows 3.1, Windows NT or Windows 95, 12 megabytes of disk space, 4 megabytes of RAM, SVGA display, and 3.5″ floppy drive. Order No. DBOIID $265

RA-search on-line service

RA-search is a computer-based text retrieval program developed by Reedy Engineering. It contains full text searching and viewing capabilities of the more than 6,000 ASME Interpretations published that relate to the Codes and Standards listed below.

RA-search's database includes all of the published Interpretations to the following ASME Codes and Standards:

- all published Boiler and Pressure Vessel Code Interpretations (Sections I, II, III-1 & 2, IV, V, VI, VII, VIII-1 & 2, IX, X and XI)
- all published NQA-1 and NQA-2 Interpretations
- all published B16 (B16.3, B16.5, B16.15 and B16.34) and B31 (B31.1, B31.3, B31.4, B31.5, B31.8 and B31.9) Interpretations
- A39, A90, A112, B1, B2, B15, B18, B20, B30, B36, B56, B73, B94, CSD, MH14, N45.2, N509, N510, PTC's

RA-search costs $500 per year plus $3.00 per minute of on-line access. The annual rate is invoiced at the start of your subscription and the access charges are billed monthly. To start your subscription to RA-search Interpretations, or if you have any questions, contact:

Rick Swayne, Reedy Engineering, 3425 South Bascomb Avenue, Campbell, CA 95008; or call 408-558-0330

ACCREDITATION AND CERTIFICATION

Board on Accreditation, Registration, and Certification (BARC)
Alan Bagner (212) 591-8580

Boiler and Pressure Vessel Accreditation
Ronald Berberich (212) 591-8461

AIA Accreditation
Bibi Rahim (212) 591-8465

Fastener Accreditation
Bibi Rahim (212) 591-8465

ISO 9000 Registration
Christine Bujal (212) 591-8592

Nuclear Accreditation
Bibi Rahim (212) 591-8465

QAI Authorized Inspection
Ken Baron (212) 591-7019

QEI Accreditation (Elevator Inspectors)
Bibi Rahim (212) 591-8465

QHO Certification (Hazardous Waste Incinerator)
Sandra Bridgers (212) 591-8465

QMO Certification (Medical Waste Incinerator Operators)
Sandra Bridgers (212) 591-8465

QRO Certification (Resource Recovery Facility Operators)
Sandra Bridgers (212) 591-8465

RTP Accreditation (Reinforced Thermoset Plastic Corrosion Resistant Equipment)
Bibi Rahim (212) 591-8465

SPPE Accreditation (Offshore Equipment)
Bibi Rahim (212) 591-8465

NUCLEAR CODES AND STANDARDS

Board on Nuclear Codes and Standards (BNCS)
Gerry Eisenberg (212) 591-8510

Joint ASME/NRC Symposium on Valve and Pump Testing
Gerry Eisenberg (212) 591-8510

BPV Subcommittee on Nuclear Power (III)
Chris Sanna (212) 591-8513

BPV Joint ACI/ASME Committee on Concrete Components for Nuclear Service (III-2)
Chris Sanna (212) 591-8513

BPV Subcommittee on Nuclear Inservice Inspection (XI)
Joseph Saltarelli (212) 591-7005

Interpretations—Section III
Joseph Saltarelli (212) 591-7005

CONAGT Main Committee on Nuclear Air and Gas Treatment Equipment
Joseph Saltarelli (212) 591-7005

Committee on Cranes for Nuclear Facilities
Jess Moon (212) 591-8514

NQA Committee on Nuclear Quality Assurance
Chris Sanna (212) 591-8513

O&M Committee on Operation and Maintenance of Nuclear Power Plants
Jess Moon (212) 591-8514

QME Committee on Qualification of Mechanical Equipment Used in Nuclear Power Plants
Gerry Eisenberg (212) 591-8510

Committee on Nuclear Risk Management
Gerry Eisenberg (212) 591-8510

PERFORMANCE TEST CODES

Board on Performance Test Codes (BPTC)
William Hays (212) 591-8550

PTC Performance Monitoring
George Osolsobe (212) 591-8554

PTC 1 General Instructions
Jack H. Karian (212) 591-8552

PTC 2 Definitions and Values
George Osolsobe (212) 591-8554

PTC 3 Fuels
Jack H. Karian (212) 591-8552

PTC 4.1 Steam Generating Apparatus
Jack H. Karian (212) 591-8552

PTC 4.2 Coal Pulverizers
Jack H. Karian (212) 591-8552

PTC 4.3 Air Heaters
Jack H. Karian (212) 591-8552

PTC 4.4 Gas Turbine Heat Recovery Steam Generators
Jack H. Karian (212) 591-8552

PTC 5 Reciprocating Steam Driven Engines
Jack H. Karian (212) 591-8552

PTC 6 Steam Turbines
Jack H. Karian (212) 591-8552

PTC 7 Reciprocating Steam Driven Displacement Pumps
George Osolsobe (212) 591-8554

PTC 7.1 Displacement Pumps
George Osolsobe (212) 591-8554

PTC 8.2 Centrifugal Pumps
George Osolsobe (212) 591-8554

PTC 9 Displacement Compressors
Jack H. Karian (212) 591-8552

PTC 10 Compressors and Exhausters
Jack H. Karian (212) 591-8552

PTC 11 Fans
Jack H. Karian (212) 591-8552

PTC 12.1 Feedwater Heaters
George Osolsobe (212) 591-8554

PTC 12.2 Surface Steam Condensers
Jack H. Karian (212) 591-8552

PTC 12.3 Deaerators
Jack H. Karian (212) 591-8552

PTC 12.4 Moisture Separator Reheaters
Jack H. Karian (212) 591-8552

PTC 12.5 Single Phase Heat Exchangers
George Osolsobe (212) 591-8554

PTC 14 Evaporating Apparatus
Jack H. Karian (212) 591-8552

PTC 16 Gas Producers & Continuous Gas Generators
Jack H. Karian (212) 591-8552

PTC 17 Reciprocating Internal Combustion Engines
William Hays (212) 591-8550

PTC 18 Hydraulic Prime Movers
George Osolsobe (212) 591-8554

PTC 19.1 Measurement Uncertainty
George Osolsobe (212) 591-8554

PTC 19.2 Pressure Measurement
George Osolsobe (212) 591-8554

PTC 19.3 Temperature Measurement
Oliver Martinez (212) 591-8553

PTC 19.5 Application, Part II of Fluid Meters
Oliver Martinez (212) 591-8553

PTC 19.22 Digital Systems Techniques
Oliver Martinez (212) 591-8553

PTC 20.1 Speed & Load Governing Systems for Steam Generator Units
George Osolsobe (212) 591-8554

PTC 20.2 Overspeed Trip Systems for Steam Generator Units
George Osolsobe (212) 591-8554

PTC 20.3 Pressure Control Systems Used on Steam Turbine Generator Units
George Osolsobe (212) 591-8554

PTC 21 Particulate Matter Collection Equipment
William Hays (212) 591-8550

PTC 22 Gas Turbines
George Osolsobe (212) 591-8554

PTC 23 Atmospheric Water Cooling Equipment
Jack H. Karian (212) 591-8552

PTC 23.1 Spray Cooling Systems
Jack H. Karian (212) 591-8552

PTC 24 Ejectors
William Hays (212) 591-8550

PTC 25 Pressure Relief Devices
Umberto D'Urso (212) 591-8553

PTC 26 Speed Governing Systems for Internal Combustion Engine Generator Units
William Hays (212) 591-8550

PTC 28 Determining the Properties of Fine Particulate Matter
William Hays (212) 591-8550

PTC 29 Speed Governing Systems for Hydraulic Turbine-Generator Units
Oliver Martinez (212) 591-8553

PERFORMANCE TEST CODES (cont'd)

PTC 30 Air Cooled Heat Exchangers
Jack H. Karian (212) 591-8552

PTC 31 Ion Exchange Equipment
William Hays (212) 591-8550

PTC 32.1 Nuclear Steam Systems
William Hays (212) 591-8550

PTC 33 Large Incinerators
Jack H. Karian (212) 591-8552

PTC 34 Waste Combustors with Energy Recovery
Jack H. Karian (212) 591-8552

PTC 36 Measurement of Industrial Sound
George Osolsobe (212) 591-8554

PTC 38 Determining the Concentration of Particulate Matter in a Gas Steam
William Hays (212) 591-8550

PTC 39 Steam Traps
Oliver Martinez (212) 591-8553

PTC 40 Flue Gas Desulfurization Units
William Hays (212) 591-8550

PTC 42 Wind Turbines
William Hays (212) 591-8550

PTC 45 Emission Testing of Municipal Waste Combustors
Oliver Martinez (212) 591-8553

PTC 46 Overall Plant Performance Test
Jack H. Karian (212) 591-8552

PTC 47 Integrated Gasification Combined Cycle
Jack H. Karian (212) 591-8552

PTC 50 Fuel Cell Power Systems
Jack H. Karian (212) 591-8552

PRESSURE TECHNOLOGY CODES AND STANDARDS

Board on Pressure Technology Codes & Standards (BPTCS)
Mark Sheehan (212) 591-8530

BPE Bioprocessing Equipment Committee
Paul Stumpf (212) 591-8536

PCI Post Construction Issues Committee
June Yarmush (212) 591-8539

Boiler and Pressure Vessel Code Committee (Exec. & MC)
Joe Brzuszkiewicz (212) 591-8533

BPV SC on Power Boilers (SC I) (Incl. Sect. VII)
Paul Stumpf (212) 591-8536

BPV SC on Materials (SC II) (Parts A, B & D)
Faron Lyons (212) 591-8540

BPV SC on Transport Tanks (SC XIII)
Paul Stumpf (212) 591-8536

BPV SC on Heating Boilers (SC IV) (Incl. Sect. VI)
Brian Feder (212) 591-8534

BPV SC on Nondestructive Examination (SC V)
F. Shawn Fitzgerald (212) 591-8056

BPV SC on Safety Valve Requirements (SC SVR)
Umberto D'Urso (212) 591-8535

BPV SC on Pressure Vessels (SC VIII)
Alan Roby (212) 591-8538

BPV SC on Welding (SC IX) (Incl. Sect. II, Part C)
Jon Labrador (212) 591-8532

BPV SC on Fiber-Reinforced Plastic Pressure Vessels (SC X)
Alan Roby (212) 591-8538

B16 Main Committee—Valves, Flanges, Fittings and Gaskets
Patricia Reddington (212) 591-8537

B31 Code for Pressure Piping (Main Committee)
June Yarmush (212) 591-8539

B31.1 Power Piping Committee
Paul Stumpf (212) 591-8536

B31.3 Chemical Plant and Petroleum Piping Committee
Jon Labrador (212) 591-8532

B31.4/11 Liquid and Slurry Piping Transportation Systems Committee
Brian Feder (212) 591-8534

B31.5 Refrigeration Piping Committee
Faron Lyons (212) 591-8540

B31.8 Gas Transmission and Distribution Piping Systems
F. Shawn Fitzgerald (212) 591-8056

B31.9 Building Services Piping Section Committee
Faron Lyons (212) 591-8540

B96 Welded Aluminum Alloy Storage Tanks Committee
Jon Labrador (212) 591-8532

HPS High Pressure Systems Committee
Umberto D'Urso (212) 591-8535

PVHO Pressure Vessels for Human Occupancy Committee
Patricia Reddington (212) 591-8537

RTP Committee on Reinforced Thermoset Plastic Corrosion Resistant Equipment
Alan Roby (212) 591-8538

TDP Committee on Turbine Water Damage
Alan Roby (212) 591-8538

SAFETY CODES AND STANDARDS

Board on Safety Codes and Standards (BSCS)
Marcy Weinstock (212) 591-8526

Advisory Committee on Environmental Management (ACEM)
Geraldine Burdeshaw (212) 591-8523

A13 Committee on Scheme for theIdentification of Piping Systems
Melissa Aranzamendez (212) 591-8522

A17 Committee on Elevators and Escalators
Geraldine Burdeshaw (212) 591-8523

A18 Committee on Platform Lifts and Stairway Chairlifts
Melissa Aranzamendez (212) 591-8522

A90 Committee on Safety Standards for Manlifts
Joseph Wendler (212) 591-8524

A120 Committee on Safety Requirements for Powered Platforms for Building Maintenance
Melissa Aranzamendez (212) 591-8522

B15 Committee on Safety Standards for Mechanical Power Transmission Apparatus
Joseph Pang (212) 591-8525

B19 Committee on Safety Standards for Compressor Systems
Melissa Aranzamendez (212) 591-8522

B20 Committee on Safety Standards for Conveyors and Related Equipment
Joseph Wendler (212) 591-8524

B30 Committee on Safety Standards for Cableways, Cranes, Derricks, Hoists, Hooks, Jacks, and Slings
Joseph Pang (212) 591-8525

B56 Committee on Powered and Nonpowered Industrial Trucks
Umberto D'Urso (212) 591-8535

CSD Committee on Controls and Safety Devices for Automatically Fired Boilers
Melissa Aranzamendez (212) 591-8522

PALD Committee on Portable Automotive Lifting Devices
Joseph Wendler (212) 591-8524

QEI Committee on Qualification of Elevator Inspectors
Shari Griffin (212) 591-8586

QFO Committee on Qualification of High Capacity Fossil Fuel Fired Plant Operators
Joseph Wendler (212) 591-8524

QHO Committee on Qualification of Hazardous Waste Incinerator Operators
Shari Griffin (212) 591-8586

QMO Committee on Qualification of Medical Waste Incinerator Operators
John Millman (212) 591-8584

QRO Committee on Qualification of Resource Recovery Facility Operators
John Millman (212) 591-8584

SPPE Committee on Safety and Pollution Prevention Equipment Used in Offshore Oil and Gas Operations
Melissa Aranzamendez (212) 591-8522

STANDARDIZATION

Board on Standardization (BoS)
Steve Weinman(212) 591-7002

A112 Plumbing Materials and Equipment
Calvin Gomez (212) 591-7021

B1 Screw Threads
James Bird (212) 591-8018

B5 Machine Tools
James Bird (212) 591-8018

B18 Fasteners
Kate Padilla (212) 591-7004

B29 Chains and Sprockets
Kate Padilla (212) 591-7004

B32 Metal and Metal Alloy Wrought Mill Product Nominal Size
Steve Weinman(212) 591-7002

B40—Specifications for Pressure and Vacuum Gauges
James Bird (212) 591-8018

B46 Classification and Designation of Surface Qualities
Brian Biddinger (212) 591-8684

B47 Gage Blanks
Calvin Gomez (212) 591-7021

B73 Chemical Standard Pumps
Calvin Gomez (212) 591-7021

B89 Dimensional Metrology
Brian Biddinger (212) 591-8684

B94 Cutting Tools, Holders, Drivers, and Bushings
Brian Biddinger (212) 591-8684

B107 Hand Tools and Accessories
James Bird (212) 591-8018

B133 Gas Turbine Procurement
Kate Padilla (212) 591-7004

HST—Overhead Hoist
Calvin Gomez (212) 591-7021

MFC Measurement of Fluid in Closed Conduits
Kate Padilla (212) 591-7004

MH1 Pallets, Slip Sheets, and Other Bases for Unit Loads
Kate Padilla (212) 591-7004

SRB Slewing Ring Bearings
Oliver Martinez (212) 591-8553

STS Steel Stacks
Oliver Martinez (212) 591-8553

Y14 Engineering Drawing and Related Documentation Practices
Calvin Gomez (212) 591-7021

Data Report Forms

Boiler & Pressure Vessel Code Data Report Forms are available in pads of 100 sheets each at a cost of $22.00 per pad. Please order by the Order Number.

		Order No.
A-1	Manufacturer's Data Report for Pressure Vessels	E00112
A-2	Manufacturer's Partial Data Report (A Part of a Pressure Vessel Fabricated by One Manufacturer for Another Manufacturer)	E00114
A-3	Manufacturer's Data Report Supplementary Sheet	E00119
C-1	Certificate Holder's Data Report for Concrete Reactor Vessels and Containments	E00077
H-2	Manufacturers' Data Report for All Types of Boilers Except Watertube and Those Made of Cast Iron	E00101
H-3	Manufacturers' Data Report for Watertube Boilers	E00102
H-4	Manufacturers' Partial Data Report	E00103
H-5	Manufacturers' Master Data Report for Boilers Constructed from Cast Iron	E00041
HLW-6	Manufacturers' Data Report for Water Heaters or Storage Tanks	E00105
HLW-7	Manufacturers' Partial Data Report for Water Heaters and Storage Tanks	E00106
HLW-8	Manufacturers' Master Data Proof Test Report for Water Heaters or Storage Tanks	E00107
K-1	Manufacturers' Data Report for High Pressure Vessels	E00130
K-2	Manufacturers' Partial Data Report for High Pressure Vessels	E00131
K-3	Manufacturers' Data Report and Supplementary Sheet	E00132
N-1	Certificate Holder's Data Report for Nuclear Vessels	E00038
N-1A	Certificate Holder's Data Report for Nuclear Vessels (Alternate Form for Single Chamber Completely Shop-Fabricated Vessels Only)	E00039
N-2	Certificate Holder's Data Report for Identical Nuclear Parts and Appurtenances	E00040
N-3	Owner's Data Report for Nuclear Power Plant Components	E00024
N-5	Certificate Holder's Data Report for Installation or Shop Assembly of Nuclear Power Plant Components, Component Supports, and Appurtenances	E00025
N-6	Certificate Holder's Data Report for Storage Tanks	E00054
NCS-1	Certificate Holder's Data Report for Core Support Structures	E00065
ND-1	Certificate Holder's Data Report for Rupture Disk Devices	E00128
NF-1	Certificate Holder's Data Report for Components Supports	E00075
NIS-1	Owner's Report for Inservice Inspections	E00029
NIS-2	Owner's Report for Repairs or Replacements	E00030
NIS-2A	Repair/Replacement Certification Record	E00126
NM-1	Certificate Holder's Data Report for Tubular Products and Fittings Welded with Filler Metal	E00080
NPP-1	Certificate Holder's Data Report for Fabricated Nuclear Piping Subassemblies	E00062
NPV-1	Certificate Holder's Data Report for Nuclear Pumps or Valves	E00037
NV-1	Certificate Holder's Data Report for Pressure or Vacuum Relief Valves	E00042

OAR-1	Owner's Activity Report	E00127
P-2	Manufacturers' Data Report for All Types of Boilers Except Watertube and Electric	E00068
P-2A	Manufacturers' Data Report for All Types of Electric Boilers	E00120
P-3	Manufacturers' Data Report for Watertube Boilers, Superheaters, Waterwalls, and Economizers	E00069
P-3A	Engineering-Contractor Data Report for a Complete Boiler Unit	E00070
P-4	Manufacturer's Partial Data Report	E00071
P-4A	Manufacturer's Data Report for Fabricated Piping	E00072
P-5	Summary Data Report for Process Steam Generators	E00073
P-6	Manufacturers' Data Report Supplementary Sheet	E00074
P-7	Manufacturers' Data Report for Safety Valves	E00129
QB-482	Brazing Procedure Specification (BPS)	E00031
QB-484	Manufacturer's Record of Brazer or Brazing Operator Qualification Tests	E00033
QW-482	Suggested Format for Welding Procedure Specification (WPS)	E00006
QW-483	Suggested Format for Procedure Qualification Record (PQR)	E00007
QW-484	Suggested Format for Welder/Welding Operator Performance Qualifications (WPQ)	E00008
RP-1	Fabricator's Data Report for Fiber-Reinforced Plastic Pressure Vessels (Class I)	E00115
RP-2	Fabricator's Partial Data Report (Class I) (A Part of a Fiber-Reinforced Plastic Pressure Vessel Fabricated by One Manufacturer for Another Manufacturer)	E00116
RP-3	Fabricator's Data Report for Class II Vessels	E00124
RP-4	Fabricator's Partial Data Report for Class II Vessels (A Part of a Fiber-Reinforced Plastic Pressure Vessel Fabricated by One Fabricator for Another Fabricator)	E00125
U-1	Manufacturer's Data Report for Pressure Vessels	E00108
U-1A	Manufacturer's Data Report for Pressure Vessels (Alternative Form for Single Chamber, Completely Shop-Fabricated Vessels Only)	E00117
U-2	Manufacturer's Partial Data Report (A Part of a Pressure Vessel Fabricated by One Manufacturer for Another Manufacturer)	E00110
U-2A	Manufacturer's Partial Data Report (Alternative Form)	E00121
U-3	Manufacturer's Certificate of Compliance Covering Pressure Vessels to be Stamped with the UM Symbol. See U-1(j).	E00111
U-4	Manufacturer's Data Report Supplementary Sheet	E00118

Books of Interest to Boiler & Pressure Vessel Code Users

The Code: An Authorized History of the ASME Boiler and Pressure Vessel Code

by Wilbur Cross

Explore the fascinating history of the ASME Boiler and Pressure Vessel Code. From generations past and the events that led to the need for codes to the personalities and pioneers who created the first codes, this in-depth volume chronicles one of engineering's greatest achievements.

Contents: Recent Highlights and Milestones; Early History of Steam Power; Events Leading Up to the Need for Codes; People Who Created the First Codes; All Work and No Relief for the Committee; Progress and Challenges; Standards and Standardization; The Decade of Power; Pushing the Limits; The "Great Period"; Pressure Vessel Code Development; The Technology of Examination and Testing; Keeping Abreast of Technology; The Code Goes to War; The Nuclear Dawn; History Repeated; Achievements and Setbacks; A Century of Progress; Conclusion.

1989 240 pp. ISBN: 0-7918-2024-6
Order No. 820246 $32.00

Pressure Vessel and Piping Design Collected Papers: 1927–1959

editorial committee: E.O. Bergman, J.J. Murphy, J.E. Soehrens, D.B. Wesstrom, F.S.G. Williams, and L.P. Zick

This volume gives engineers a wider base of reference from which to make decisions on problems they encounter daily. It provides a valuable overview of the pressure retention discipline and will give readers an improved intuitive recognition of good

design. Topics include: Openings; Bolted-Flange Joints; Heads; Shells; Piping; Materials; Thermal Stress and Fatigue; Loads and Supports; External Pressure

1960 (original printing) 720 pp. Hardcover
Order No. I00364 $95 (list)/$76 (ASME member)

Boiler and Pressure Vessel Certificates of Authorization for Use of Code Symbol Stamps and Testing Laboratories Holding Certificates of Acceptance for Capacity Certification of Pressure Relief Devices

The volume lists company names, addresses, and certificate expiration dates for companies holding Boiler and Pressure Vessel Certificates of Authorization for Sections I, IV, VIII Divisions 1 and 2, and X. Includes listings by Stamp (A, E, M, PP, S, V, H, HLW, HV, U, UM, UV, U2, and RP), and by U.S. states, Canadian provinces and other countries. Also includes a listing of companies holding certificates of acceptance for capacity certification of pressure relief devices.

Order No. E00052 $75.00

Companies Holding Nuclear Certificates

Lists company names, addresses, scopes, and expiration dates for all nuclear certificates (N, NA, NPT, NV, QSC (MM), QSC (MO), QSC (MS), Interim Letters and Owner's Certificates). Also includes listing of accredited authorized nuclear inspection agencies.

Order No. E00061 $65.00

Concensus Operating Practices for Control of Feedwater/Boiler Water Chemistry in Modern INDL Boilers (CRTD 34)

48 pp. ISBN: 0-7918-1204-9
Order No. I00367 $10 (list)/$8 (ASME member)

Index

A.O. Smith Engineered Storage Products Co., T-1 to T-8
ABB Lummus Crest, C-14
abrasives, A-1 to A-8
  bond of, A-5 to A-7
  characteristics of, A-4, 4
  computer numerical control (CNC) and, A-1 to A-3
  friability of, A-3 to A-4, A-6
  grade or hardness of, A-5 to A-7
  grain size of, A-6
  grinding wheels and, A-3 to A-7
  grit size of, A-5 to A-6
  material data safety sheet for, A-8
  packaging of, A-7 to A-8
  peening processes, A-7 to A-9
  physical characteristics of, A-2
  pumping fluids containing, P-274
  selection of, A-5 to A-7
  "stations" of, A-5
  structure or bulk density of, A-5
absolute pressure switches, C-383
absolute separators vertical, S-16 to S-19
absorbers
  in air pollution control and, A-83, A-96 to A-97
  of sound, A-11
absorptive linings in soundproofing panels, A-33 to A-37
absorptivity, M-10
Abtech Scientific Inc., P-6
acceleration
  braking properties and, C-347 to C-350
  in electric motors and, E-3 to E-5, E-12, E-24, E-29
  in fans, F-3
  vibrational analysis (VA) and, C-265, C-287, C-305, C-311
accelerometers, C-269, C-324
accident management, A-9
accreditation, ASME, Sp-25 to Sp-28, Sp-34
accuracy in degrees, thermometers, M-2
acetaldehyde, P-10 to P-32, P-40 to P-57, V-15
acetic acid, V-15
acetic anhydride, V-15
acetone, C-72, C-418, P-10 to P-31, P-40 to P-57, V-15
acetonitrile, P-10 to P-31, P-40 to P-57
acetylene, C-415
acid rain, A-9 to A-13, P-60 to P-62
acid treatment, R-5
acids, in lubricating oil, L-25
acoustic enclosures, turbine (*See* soundproofing and noise control), A-10 to A-37
acoustic performance, A-37
acoustics, in reciprocating (labyrinthine) compressors, C-116
acrolein, C-415
acrylamide, P-32
acrylic acid, P-10 to P-31, P-40 to P-57
acrylic baking enamel tank exterior coating, T-5
acrylic urethane tank exterior coating, T-5
acrylonitrile, P-10 to P-32, P-40 to P-57, V-15
activated carbon, R-6
activated diffusion healing (ADH), M-32
active vs. reactive sound fields, A-20
actual deposits, R-6
actuation values, switches, C-377
actuators, A-37 to A-47
  electrically controllable servo meter for, A-42
  electrohydraulic, A-37 to A-47, F-31
  in fuel system control, F-30 to F-34
  hydraulic amplification in, A-41
  hysteresis in, A-41
  magnetic bearings using, B-5 to B-9
  proportional and dynamic conversion of electrical control signals, A-40
  Turcon CTo electrohydraulic actuator, A-47
acylamide, P-10 to P-31, P-40 to P-57
AD CARB, carbon–graphite mix products, C-8 to C-13
additives, petroleum processing, R-5 to R-6
adhesives, A-1
adiabatic expansion factor in, M-25
adipate, P-10 to P-31, P-40 to P-57
adjustable frequency electric motors, E-10 to E-15
adsorption, R-6
Advanced Carbon Products, C-8 to C-13
advanced pattern groove (APG) seals, S-3 to S-4
aerfoils (*See* metallurgy; turbines)
aerobic bacteria, R-7
aeroderivative fuel regulator for gas turbine, C-354 to C-358
aerosols, chlorofluorocarbons (CFCs) in, O-17 to O-18
AGFA, P-3 to P-7
aggregate, A-1
agitators, A-38, D-4
agriculture and agricultural product processing (*See also* food industry), A-48 to A-49, C-34, Sp-1 to Sp-3
  coolers for dairies/dairy products, C-412 to C-413, C-420
  crystallizers (*See* crystallizers)
  ejectors, E-2 to E-3
  environmental accountability, E-40 to E-49
  feeding management (livestock), E-44
  irradiation in, I-1 to I-5
  ISO VILJA technology in, E-46, E-48
  pollutants of, P-12
air blowing, R-7
air conditioning (*See also* refrigerants)
  development of, R-14 to R-15
  glycol as coolant in, C-413
  heat pump and, H-4
  R-22 refrigerant in, R-11 to R-13
  refrigerants and for, R-11 to R-13
  sizing of, C-422 to C-423
air density, fans, F-1, F-3
air entrained cement, C-22 to C-23, C-28
air filtration, A-49 to A-83
  bleed extraction systems for, A-51 to A-52
  bypass door for, A-53, A-59 to A-60, A-76
  cartridge type filter in, A-54, A-58
  chimneys and, cleaning of, C-78
  coalescer panels for, A-52 to A-53
  compressor damage in, A-75
  corrosion in, A-55 to A-59, A-64 to A-69, A-71 to A-79
  design suitability of, A-55 to A-61
  drains and drain valve in, A-56, A-62 to A-63
  dust vs., A-51 to A-52, A-56, A-66, A-70, A-74 to A-76
  dynamic water eliminator (Hydra) filters in, A-81
  erosion in, A-75 to A-78
  filter designs for, A-74
  filter elements for, A-53
  filter materials in, A-61, A-79 to A-81
  filter specifications for, offshore environment, A-77 to A-79
  flare carbon vs., A-74 to A-75
  fouling in, A-55 to A-57, A-76 to A-78
  future developments in, A-81
  high efficiency (Hercules) filters in, A-81 to A-82
  high volume flow in, A-81
  high vs. low velocity systems, A-74
  housing for, A-53 to A-54, A-58 to A-61, A-70
  huff and puff design in, A-49 to A-50
  humidity vs., A-51 to A-53, A-57, A-74
  icing conditions vs., A-49 to A-50
  insect ingestion in, A-49, A-51, A-66
  low pressure loss in, A-81 to A-83
  maritime use of, A-61
  materials used in, A-58
  National Gas Turbine Establishment (NGTE), A-69
  new designs for, A-61 to A-66
  offshore or coastal environments vs., A-57, A-66 to A-84
  oil fume ingestion in, A-49
  operational experiences with, A-53 to A-61
  particle fusion in, A-78
  pollutant ingestion and build up in, A-49
  pulse jet–type, A-49
  rainfall vs., A-50 to A-63
  salt corrosion in, A-57, A-61 to A-64, A-69 to A-74
  sand ingestion in, A-49
  silencer outlet of, A-64

specifications for, A-83
spin tubes in, A-57
stainless steel housing in, A-58, A-60, A-62 to A-63, A-71 to A-72
tropical environments and, A-50 to A-74
turbine damage in, A-58
vane separators for, A-63, A-74
water eliminator for, A-82 to A-83
weather louvers for, A-56, A-61, A-74
weatherhoods for, A-55 to A-56, A-60
air horsepower (ahp), fans, F-5
air inlet manifolds, compressors and, paint and corrosion damage in, T-74 to T-75
air movement studies, in explosion hazard assessment, E-68, E-71
air plasma coatings, M-35
air pollution control, A-83 to A-101, E-52, P-57 to P-60
absorbers for, A-83, A-96 to A-97
accidents/incidents and, E-46
blocking of catalyst vs., A-96
carbon dioxide (CO_2) (*See also* carbon dioxide)
carbon monoxide and, C-62 to C-64
Catpac TCR system in, A-100 to A-101
cement kilns and, C-22, C-28 to C-34
chemisorption in, A-96, A-98
chimneys and, cleaning of, C-78
chlorofluorocarbons (CFCs) in, C-413
combined unit systems for, A-83, A-99 to A-101
condensers and, C-233
continuous emissions monitoring (CEM) in, C-34
DeNox systems, A-90, A-93, A-100
dense phase technology in, A-98
DEPAC system for, A-98 to A-99
dioxins and, A-96 to A-97
electrostatic precipitators and, A-83 to A-85
end product handling and, A-83, A-97 to A-98
environmental accountability, E-40 to E-49
fabric filters in, A-83, A-85 to A-86
Filpac TCR system in, A-100 to A-101
Filsorption system for, A-96 to A-97, A-100 to A-101
filter product ranges in, A-87 to A-88
filters in, A-96, A-98
flue gas desulfurization (FGD) in, A-83, A-86, A-89 to A-90
fly ash handling and, A-97 to A-98
heavy metals and, A-96 to A-97
IEA Greenhouse Gas Implementing Agreement, C-14
lime/limestone FGD in, A-90
nitrous oxide (NO_x) (*See* nitrogen oxides)
ozone, O-4 to O-26
poisoning of catalyst vs., A-96
power production and, P-72
seawater FGD in, A-90, A-92
selective catalytic reduction (SCR) in, A-83, A-90, A-93 to A-96
sintering of catalyst vs., A-96
sulfur dioxide (SO_x) in, A-83, P-60 to P-62
TCR, A-99 to A-100
wet/dry FGD in, A-89 to A-91, A-100
Wetpac TCR system in, A-100 to A-101
air purification, sterilization, A-101 to A-102
air separation unit (ASU), C-17
air vessel control, in hydrodynamic brakes, with capacitative heat storage, B-19
air washing, air purification using, A-102
airfoils (*See also* metallurgy), control of, C-336
AKZO Pharma, D-12
aldrin, V-15
alkalis, in lubricating oil, L-25
alkylation, in petroleum industry, R-4
AlliedSignal, P-4
alloys, M-39, M-42 to M-43
alphatron gauges, M-22
Alstom Corporation, A-95, C-14, C-78 to C-84, G-5 to G-20, L-8 to L-10, P-4, P-73
Altair Filters International Ltd., A-10 to A-37, A-50 to A-83, S-29 to S-38
altered refinery, R-7
alternative fuels, F-37
alternators, synchronous clutches and, P-107, P-112
aluminum and aluminum alloys, P-10 to P-31, P-40 to P-57, M-40, M-42
aluminum oxide, A-4, A-6, C-25, P-10 to P-31, P-40 to P-57
ambient air pressure (Po), gas turbines and, T-55 to T-58
ambient air pressure (Pt), gas turbines and, T-58
ambient air quality objective, ozone and, O-9 to O-10
American Gear Manufacturers Association (AGMA), P-75
American National Standards Institute (ANSI)
explosion hazard assessment, E-87
generators and turbogenerators design, G-1
American Petroleum Instate (API), R-8
explosion hazard assessment, E-86
specifications, Sp-3
American Public Health Association (APHA), R-8
American Society for Testing and Materials (ASTM), M-12
cement, C-22 to C-28
American Society of Heating, Refrigeration, Air Conditioning Engineers (ASHRAE), R-11
American Society of Mechanical Engineers (ASME), A-1, M-12
ASME (*See* American Society of Mechanical Engineers), A-1
accreditation packages, Sp-25 to Sp-28
boiler code on CD-ROM from, Sp-32
boiler/pressure vessel codes, Sp-24
certification programs of, Sp-30 to Sp-31
electronic stress tables from, Sp-33
elevator and escalator CD-ROM from, Sp-31 to Sp-32
Information Handling Services (IHS) of, Sp-32
ISO 9000 registration program, Sp-29
Power Test Codes, M-12
Power Test Codes for flow measurement, M-26 to M-29
Power Test Codes for pressure measurement, M-15
pressure vessel codes on CD-ROM from, Sp-32
RA search online service of, Sp-33
specifications, Sp-3
Worldwide Standard Service of, Sp-32
American Standards Association, A-3
amines, C-43, C-58
ammonia, C-33, C-418, P-10 to P-31, P-40 to P-57, V-15 to V-16
as refrigerant, R-13 to R-18
ammonia bisulfate formation (ammonia slip), C-33
ammonium nitrate, P-10 to P-31, P-40 to P-57
ammonium sulfate, P-10 to P-31, P-40 to P-57
ammonium thiosulfate, C-72
amplitude, in vibrational analysis (VA), C-263, C-268 to C-272, C-280 to C-282, C-302, C-315 to C-317
aniline, P-10 to P-31, P-40 to P-57
annulus, hydrodynamic power transmission and, P-123 to P-127
anodes, sacrificial anodes vs., C-247
anthracene, C-68, P-10 to P-31, P-40 to P-57
anthraquinone, C-68
anti-icing additives, R-8
antifreeze, C-408 to C-412
antifungals, freeze dry production of, D-17
anti-knock compounds, R-8
anti-nodes, vibrational analysis (VA), in plots of, C-276
antimony, P-10 to P-31, P-40 to P-57
antinutritive factors, E-43
antioxidants, R-8
antistatic coatings, P-3
antisurge control/valves, C-359 to C-368
in turbocompressors, C-332 to C-335
API 610 CAP 8 type process pumps, P-285 to P-292
API 610 CVA8 overhung vertical pumps, P-292 to P-296
API 610 double suction pumps, P-296 to P-299
API 610 rigid or flexible coupled pumps, P-296
apparent viscosity, P-272
aqueous corrosion, M-40
aquifers suitable for disposal of CO_2, C-14, C-21
ARCO Chemical, C-42 to C-62, D-7 to D-8
argon, C-95
ArLon 6100 interior tank coating, T-4
Armstrong Engineering Association, C-62 to C-77, H-6 to H-23
arsenic, P-10 to P-32, P-40 to P-57
as-low-as-reasonably-practicable (ALARP) risks and, E-80
ASARCO Inc., P-4
asbestos, P-10 to P-32, P-40 to P-57
ASC MC antisurge controller, C-361 to C-368
Asea Brown Boveri, P-73
asphaltum, in lubricating oil, L-25
AssiDomän forestry management case study in, F-13 to F-21, E-50 to E-56, P-204 to P-205
atmospheric conditions vs. power degradation, in gas turbines, T-65 to T-67
atmospheric distillation, in petroleum industry, R-2
attenuation of sound, A-33 to A-37
attractive electromagnetic suspension, in bearings, B-5
auditing, E-43
operational optimization, life cycle assessment and, L-19 to L-21
authorized deposits, R-8
authorized inspection, Sp-4
autoclaving, M-34
automotive lifting devices, Sp-4
autrients, C-78
Avon Aeroderivative gas generator, Rolls Royce, M-32
Avon Mk. 1535 turbine cooling air, M-34
axial balance, in pumps and, P-243 to P-247, P-275 to P-276
axial compressors (*See also* compressors; turbocompressors), C-136 to C-165
adjustable stator blade design (VARAX), C-140, C-337 to C-338
adjusting cylinder in, C-162
applications for, C-137, C-139
axial thrust bearings in, C-163 to C-164
bearing pedestals in, C-155 to C-160
bearings in, C-163 to C-164
blade carrier in, C-160
blades in, C-160 to C-162
casing inserts in, C-160
casings in, C-155 to C-160
control systems in, C-161
cross section view (VARAX), C-159
cross sectional view of, C-141
discharge temperature in, C-156
features of, C-156

fixed stator blade design (FIXAX), C-140, C-153, C-157
gas turbines and, T-47 to T-48
horizontally split casing design, C-142
humidity vs. molecular mass of wet gas in, C-156
intercooler in, C-340
journal bearings in, C-163
longitudinal view (VARAX), C-159
oil supply system for, C-339, C-341, C-343
operating range for, C-153 to C-155
performance data for, C-154 to C-158
remote setting of reference values in, C-161 to C-162
rotors in, C-160
seals in, C-342
servomotors in, C-161 to C-162
shaft seals in, C-162 to C-163
size vs. performance of, C-154 to C-158
solid coupling in, C-164 to C-165
stator blade settings/control, C-153 to C-155, C-162
suction volume of, C-140, C-142
type designations for, C-153
variable stator blade design (VARAX), C-153, C-157 to C-159
vertically split casing design, C-143 to C-144
axial flow, in gas turbines, T-51 to T-53
axial piston, in pumps, P-260 to P-264
axial thrust bearings (*See also* thrust bearings)
in axial compressors, C-163 to C-164
in centrifugal compressors, C-178 to C-181
in isotherm turbocompressors, C-200 to C-202
axial tilting pad bearings, in high speed gears, P-104 to P-107
axis of control, in magnetic bearings, B-6
azeotrope refrigerant mixtures, R-11

back-pressure steam turbines, T-96 to T-98
back-to-back high speed gear test bed, P-99 to P-101
background noise measurement, A-20 to A-21
backmixing, in chemical reactors, R-1
baffles, solids concentration in riser using, computational fluid dynamics, C-2
balancing (*See also* condition monitoring), B-1, C-288 to C-291
causes of imbalance and, C-288 to C-291
centrifugal pumps and, P-243 to P-247
damping and, C-316 to C-318
generators and turbogenerators, G-11
influence coefficients in, C-325 to C-327
misalignment and, C-290 to C-291
modal balancing in, C-327 to C-328
onspeed, for rotors, B-1
process of, C-289 to C-290
rotary pumps, P-275 to P-276
strobe lights for, C-278 to C-280
subsynchronrous frequency orders and, C-291 to C-294
vibrational analysis (VA) and, C-278 to C-280, C-288 to C-291, C-325 to C-328
ball bearings, B-1 to B-2
balling machines, B-28
barometers, C-329, M-18
barrier coatings, F-15, F-21 to F-23
BASF/Varta, P-4
batch reactors, R-1
batteries (*See* cells)
Bayer AG, P-3 to P-4
bayonet type electric heater, H-13
bayonet type vaporizers, V-7 to V-12
bearings (*See also* axial thrust bearings; journal bearings; lubrication; magnetic bearings), B-1 to B-11
attractive electromagnetic suspension in, B-5
in axial compressors, C-155 to C-160, C-163 to C-164
axial thrust, C-163 to C-164, C-178 to C-181, C-201 to C-202
ball, B-1 to B-2
carbon-graphite mix products and, C-10, C-13
in centrifugal compressors, C-176 to C-178
in electric motors, E-16
in generators and turbogenerators, G-7 to G-9, G-15
in high speed gears and, P-101 to P-107
in isotherm turbocompressors, C-211, C-214
journal type (*See* journal bearings)
magnetic type (*See* magnetic bearings)
needle, B-1 to B-2
noncontacting technology in, magnetic type bearings, B-5
radial magnetic type, B-6 to B-7
in reciprocating (labyrinthine) compressors, C-86, C-92
roller, B-1 to B-2
rolling contact type, B-1 to B-2
sliding contact type, B-2 to B-5
stator and rotor in, magnetic, B-5
in steam turbines, T-78
temperature monitoring for, C-261
three lobe journal type, B-3
thrust type (*See* thrust bearings)
tilting pad journal type, B-4, L-25
in worm gears, P-94
Belleville washer negative spring rate sensor in, C-386 to C-387
bellows gauges, M-21
bellows type sensors, C-386
belt drives, D-3
bending stiffness of panels vs. soundproofing quality, A-29 to A-31
benzene, C-72, C-418, P-10 to P-31, P-33, P-40 to P-57, V-15
benzene hexachloride, C-68
benzoic acid, C-68
benzoyl peroxide, P-10 to P-31, P-40 to P-57
benzyl chloride, P-10 to P-31, P-40 to P-57
Bernoulli's theorem, P-222
best practical treatment (BPT), R-8
betaine, E-43
beverage industries pollutants, P-15, P-29
bimetallic sensors, C-377
bimetallic thermometers, M-8 to M-9
bioabsorbable polymers, freeze dry production of, D-17
bioassay, R-9 to R-10
biochemical oxygen demand (BOD), P-69, R-8
bioconversion, E-43
biodegradation of plastics, P-2 to P-3
biodistegrable plastics, P-2
biofilter, E-43
biological diversity, E-50
biological engineering, A-49
biological oxygen demand (BOD), E-43
biomass fuels, P-205 to P-214
coal co-gasification, CO_2 removal and, C-17 to C-18
cost vs. energy efficiency for, C-18 to C-20
gasification and reformation of natural gas in series in, C-18
biomass gasifier (BIG), P-208 to P-210
Bioplastics Inc., P-2
bioremediation, A-49
biosensors, conductive polymers in, P-6
biotechnology, E-44
biotopes, F-13
biphenyl, P-10 to P-31, P-40 to P-57
bipolar junction transistors (BJT), E-29
Bis(2-ethylhexyl) adipate, P-10 to P-31, P-40 to P-57
Bis(2-ehtylhexyl) phthalate, P-10 to P-31, P-33, P-40 to P-57
bisphenol, C-68, V-15
black liquor gasifier (BLG), P-209
black liquor processing, E-51 to E-52, P-205 to P-214
blade carrier, in axial compressors, C-160
blade tip grinders, A-1
blades (*See also* fans; metallurgy)
in axial compressors, C-160 to C-162
cracks in, L-16
in fans, F-1 to F-3
in hydrodynamic power transmission, P-123 to P-127, P-138
in isotherm turbocompressors, C-211 to C-215
material selection for, M-41
root disc serration configuration in, L-16
in steam turbines and, T-76, T-88 to T-90
vibrational analysis (VA) and, C-294
Blasberg Oberflae-chentechnik, P-4
blast furnace gas, F-37
blending
gravity, in tanks, T-4 to T-8
in petroleum industry, R-2, R-5 to R-6
blocking of catalyst, in air pollution control, A-96
blowdown, R-8
blowers
rotary piston, C-232
rotating positive displacement compressors and, C-228 to C-232
synchronous clutches and, P-107
Bode plots, in vibrational analysis (VA), C-270, C-272 to C-280
boilers (*See also* pressure vessels; turbines, steam), B-11, S-38, Sp-4
ASME accreditation packages for, Sp-25 to Sp-28
ASME CD-ROM for, Sp-32
books of interest on, Sp-37 to Sp-38
centrifugal pumps and, P-242
dryness of steam produced by, T-111
hydrodynamic power transmission and, in pumps, P-152 to P-164
International Code for, Sp-24
ISO 9000 registration program, Sp-29
steam separators for, T-108 to T-112
Tomlinson, in pulp and paper mill, P-210
bond of abrasives, A-5 to A-7
booster, in ultrasonic cleaners, U-3
borescopes, B-11
Bourdon tube sensors, C-383, M-8, M-21, P-224
brakes, hydrodynamic, B-11 to B-23
advantages of, B-14
air vessel control in, with capacitative heat storage, B-19
applications for, B-13 to B-14, B-23 to B-28
braking power in, B-16, B-20 to B-21
braking torque in, B-16, B-20 to B-21
closed vs. open circuit, B-12 to B-13
components of, B-15 to B-16
control and energy dissipation in, B-12 to B-13
control characteristic curves for, B-23 to B-24
controls and control schematics for, B-16 to B-19
dimensions and technical data, single and double flow models, B-22 to B-23
heat exchanger control in, B-17
high speed and acceleration, in gas turbines, C-347 to C-350
models of, Voith Corp., B-14 to B-16
non-self-supporting, B-14 to B-16

operation of, B-11 to B-12
performance characteristics of, graphs for, B-16, B-20 to B-21
pump control in, B-17 to B-19
schematic of, B-25
self-supporting, B-15 to B-16
speed vs. braking torque/power, B-20 to B-21
through flow control in, B-18
VHBI, B-14 to B-16, B-25 to B-28
VHBK, B-27
VHBM, B-14 to B-16
brass, in condensers, C-245
brazing, M-31 to M-32
breakdown rpm region, in electric motors, E-24 to E-25
breather drains, in electric motors and, E-18
brick lined process equipment, B-23
Bridgestone Corp., P-4
briquette machine, A-48, B-28
Bristol Meyers Co., D-17
bromine, V-16
Brunei Shell petroleum Fairly 4 platform, A-73
brushes
carbon-graphite mix products for, C-13
generators and turbogenerators, G-16
brushless excitation, generators, G-12 to G-16
building materials industry, pollutants, P-30
bulb and capillary sensors, C-378 to C-379
Buna-N shaft seal, electric motors, E-17
burners, biomass fuel fed, from pulp and paper mill, P-210
business service industries pollutants, P-31
butadiene, C-415, P-10 to P-31, P-33, P-40 to P-57, V-15
butane cyclopropane, C-418
butane, F-41, V-15, V-17
butanol, V-16
butene, V-16
butterfly valves, C-392
butyl alcohols, P-10 to P-31, P-40 to P-57
butyl benzyl phthalate, P-10 to P-31, P-40 to P-57
butyl cresol, C-68
butylamine, V-16
butylene, V-15
butyric acid, C-68
bypass doors, in air filtration systems, A-53, A-59 to A-60, A-76

C.I acid green 3, P-10 to P-31, P-40 to P-57
cadmium, P-10 to P-31, P-34, P-40 to P-57
calcium carbonate, C-14
calcium cyanamide, P-10 to P-31, P-40 to P-57
calcium nitrate, C-68, C-72
calcium oxide (CaO), C-28
calcium sulfate, C-25
calibration
of pressure measurement devices, M-16
of temperature measuring instruments, M-11 to M-12
Cambridge Display Technology (CDT), P-5
Campbell diagrams, in vibrational analysis (VA), C-323 to C-326
Canadian Petroleum Products Insatiate (CPPI), R-8
Canadian studies in pollution, pollution control, P-8 to P-72
CAP 8 type process pumps (API 610), P-285 to P-292
capacitance method, in level sensing, C-388
capacity (cfm), in fans, F-1, F-3
capacity of flow, in flow measurement, M-24 to M-25
caprolactum, C-68
carbon dioxide (CO_2), C-13 to C-21, C-95, E-36, F-18, V-16
aquifers suitable for disposal of, C-14, C-21
biomass gasification and reformation of natural gas in series in, C-18
capture of, from gas turbine base power plants, C-14 to C-16
cement production and, C-29 to C-30
coal and biomass co-gasification in, C-17 to C-18
cost vs. energy efficiency in, C-18 to C-22
energy efficiencies vs. costs, C-18 to C-20
global warming, greenhouse gases and, C-13
ground injection project (Sweden) for, C-14 to C-21
IEA Greenhouse Gas Implementing Agreement, C-14
integrated gasification combined cycle (IGCC) and, C-14 to C-16
limiting emissions of, C-14
liquefaction of, for dry cleaning products, C-13
liquid phase methanol synthesis (LPMeOH) in, C-17 to C-18
methanol, power, heating, neutral co-production of, C-16 to C-17
monoethanol amine (MEA) solvent in, C-15
natural gas combined cycle (NGCC) and, C-14 to C-16
power production and, P-72
as refrigerant, R-13 to R-18
Selexol process for, C-15 to C-17
solidification of, C-14
syngas production in, C-17 to C-18
technology and cost options for, C-14 to C-21
transport and disposal of, C-20
carbon disulfide, P-10 to P-31, P-40 to P-57
carbon-graphite mix products, C-7 to C-13
applications for, example of, C-11 to C-12
bearing design for, C-10, C-13
chemical resistance of, C-9
close in tolerance of, C-10
coefficient of friction in, C-9
coefficient of thermal expansion in, C-9
design factors for, C-9 to C-11
electrical contacts and, C-13
environmental factors and, C-9
filters using, C-13
interference fit for, C-9
load bearing capacity of, C-9
material classification for, C-9
mating materials for, C-9
mechanical strength of, C-9
oxidization resistance in, C-9
products made from, C-13
properties of, C-8
running clearance for, C-10
seals using, C-12 to C-13
self-lubricating properties of, C-8
specialty products using, C-13
temperature parameters for, C-9
tolerances for, C-10
carbon monoxide (CO), C-15, E-38, P-62 to P-64
carbon tetrachloride, P-10 to P-31, P-34, P-40 to P-57, V-15
carcinogens, C-38, M-32
cardboard (*See* packaging; paper products and packaging)
cartons, paper, F-24
cartridge filters, F-6 to F-8
in air filtration systems, A-54, A-58
cascade diagrams, in vibrational analysis (VA), C-272 to C-273
casings
in axial compressors and, C-155 to C-160
in centrifugal compressors, C-170 to C-175
in isotherm turbocompressors, C-205 to C-206, C-210 to C-211
in rotary pumps, P-276
in steam turbines and, T-76
castings (*See also* metallurgy), M-43
catalyst, R-8
catalysts, fluidized beds, F-12
catalytic cracking, petroleum industry, P-2 to P-3
catalytic reactors, R-1
CATAPOL project, C-5, C-7
Catpac TCR system, air pollution control and, A-100 to A-101
caustic soda, C-43, C-58
cavitation, C-264, C-328, P-232
cells, chemical, C-22, P-4
Celsius temperature conversion, M-12
cement, Portland; and kilns, C-22 to C-34
additions to, C-24, C-28
air entrained, C-22 to C-23, C-28
American Society for Testing and Materials (ASTM) specs for, C-22 to C-28
carbon dioxide (CO_2) emission, C-34
CCME (Canada) National Emission Guidelines for kilns and, C-28 to C-34
certification of, by manufacturer, C-28
chemical requirements of, C-25 to C-26
clinkering process in, C-29 to C-30
combustion and operation modification (COM) to reduce emissions from, C-32
continuous emissions monitoring (CEM) in, C-34
definition of, C-23, C-29
energy consumption in production of, C-29 to C-30
hazardous wastes used in, C-33
inspection of, C-26
kilns for, environmental impact of, C-22, C-28 to C-34
low NO_x burners (LNBs) to reduce emission from, C-32 to C-33
manufacturer's statement for, C-27
measurement of emissions from, vs. costs, C-34
nitrogen oxide (NO_x) and, C-29 to C-31
ordering information for, C-23
packaging and package marking for, C-27 to C-28
physical properties of, C-24, C-29 to C-30
production of, C-28 to C-34
rejection of, C-26
sampling of, C-24
selective noncatalytic reduction (SNCR) to reduce emission from, C-33
setting time for, C-24
staged air combustion to reduce emission from, C-32
storage of, C-28
sulfur dioxide (SO_2) emission from, C-34
testing of, C-24
types and classification of, C-22 to C-23
centrifugal compressors (*See also* compressors; turbocompressors), C-81, C-136, C-165 to C-190
adaptability to application demands in, C-167 to C-170
applications for, C-182 to C-190
axial thrust bearings in, C-178 to C-181
axial thrust in, C-176 to C-178
barrel type design, C-171 to C-174
bearings in, C-176 to C-178
building block system in, C-170 to C-171
casing in, C-170 to C-175
drainage system in, C-181 to C-182
drive for, C-188
features of, C-170 to C-190
flow conditions and, C-167 to C-168
fluid injection devices in, C-182
horizontally split casing design in, C-166 to C-170, C-172
impellers in, C-172, C-174 to C-177, C-186
inlet guide vane assembly in, C-181
journal bearings in, C-178 to C-181

magnetic bearings and, C-185 to C-186, C-189 to C-190
materials used in, C-182 to C-183
motor for, C-187 to C-190
pressure range for, C-168 to C-170
process conditions for, C-168
retrofit design modification for gas pipeline station, C-182 to C-186
rotor in, C-189 to C-190
selection chart for, C-168
series designations for, C-169
shaft seal in, C-180
size vs. performance in, C-167 to C-170
solid coupling in, C-176 to C-178
testing of, C-188 to C-189
type range of, C-166 to C-170
vertically split casing design in, C-168 to C-170
centrifugal gas exhausters, E-56 to E-57
centrifugal pumps, P-218 to P-259
applications for, P-218 to P-219, P-241 to P-255
axial balance in, P-243 to P-247
Bernoulli's theorem and, P-222
boiler feed, P-242
Bourdon tube to measure flow in, P-224
cavitation in, P-232
characteristic curves of performance for, P-228
condensate, NPSH and, P-236 to P-237, P-243
continuity equation in, P-221 to P-224
double volute, P-247 to P-248
friction loss in, P-222
head in, P-222, P-224 to P-228
hot water, NPSH and, P-236
Hydraulic Institute Standards for, P-231
hydraulic pressure, P-242 to P-243
hydraulics of, P-221 to P-241
impellers in, P-219, P-226 to P-228
installation of, P-257 to P-258
kinetic head in, P-222
materials selection, by liquids to be pumped, P-241
minimum flow through, P-237 to P-240
nonclogging, P-243
operation of, P-257 to P-258
parallel operations for, P-228 to P-230
performance analysis of, P-226 to P-228
piping for, P-257
potential head in, P-222
power/horsepower requirement calculation for, P-225 to P-226
pressure head in, P-221 to P-224
priming of, P-257
recirculation connection for, P-240 to P-241
seals in, P-250 to P-254
series operations for, P-228 to P-230
single stage, double suction type, P-241 to P-242
specific speed of, P-219 to P-221
static head in, P-221 to P-224
stuffing boxes for, P-248 to P-254
suction head in, net positive (NPSH), P-232 to P-237, P-257
suction head in, P-222
suction lift in, P-222
temperature rise in, P-237 to P-240
temperature vs. efficiency in, P-254 to P-255
total discharge head in, P-223 to P-224
total head in, P-222
troubleshooting, P-258 to P-259
valves in, P-257
velocity head in, P-221 to P-224
vibration limits for, P-255 to P-258
viscous liquids and, P-230 to P-232
warmup procedures for, P-254
centrifuges, A-48, C-34 to C-38
ceramic abrasive, A-4
ceramics, A-2, C-38
CERCLA, propylene glycol and, C-412 to C-413
certification, Sp-14, Sp-30 to Sp-31, Sp-34
chains, Sp-4
characteristic curves of performance, pump, P-228
check valves, C-392
Chem Systems/Air Products, C-17 to C-18
chemical cleaning, C-38, T-67
chemical complex (petrochemicals), chemical plant, C-38 to C-41
chemical engineering, Sp-1 to Sp-3
Chemical Manufacturers Association (CMA) guidelines, C-61
chemical oxygen demand (COD), R-8
chemical reactors, R-1
chemical treating, R-5
chemicals (*See also* hazardous materials; pollutants; toxic chemicals), C-41 to C-42, Sp-1 to Sp-3
inorganic, C-64
pollutants and, P-25 to P-26
toxic, handling of, C-42 to C-62
transport of, T-40 to T-41
chemisorption, in air pollution control, A-96, A-98
CHEMTREC information service, telephone number, C-61
chillers
for agricultural and food products, A-48
crystallizers as (*See also* crystallizers), C-62
chimneys (*See also* stacks), C-78
chlordane, P-7
chlorine dioxide, P-10 to P-31, P-40 to P-57, V-15
chlorine, C-86, P-10 to P-31, P-40 to P-57, V-15 to V-16
chlormethane, P-11 to P-31, P-40 to P-57
chloroacetic acid, P-11 to P-31, P-40 to P-57
chlorobenezene, C-68, C-72, V-15
chloroethane, P-11 to P-31, P-40 to P-57
chlorofluorcarbons (CFCS) (*See also* ozone; refrigerants), C-13 to C-14, R-10 to R-11, R-14 to R-15
aerosols using, O-17 to O-18
coatings containing, O-14, O-16 to O-26
cost of replacing use of, O-16 to O-22
foams containing, O-14, O-16 to O-26
insulation containing, O-14, O-16 to O-26
ozone and depletion and, C-13 to C-14
packaging containing, O-14, O-16 to O-26
refrigerants containing, O-14, O-16 to O-26
solvents containing, O-14, O-16 to O-26
sterilants containing, O-14, O-16 to O-26
chloroform, P-11 to P-31, P-34, P-40 to P-57
chloromethanes, V-15
chromium, P-11 to P-31, P-34, P-40 to P-57
circular pitch, worm gears, P-84
civil engineering, Sp-1 to Sp-3
Clarkson Co. (knife gate valve), C-402 to C-403
Clean Air Act, propylene glycol and, C-412 to C-413
cleaning, in metallurgy and, M-32, M-34
cleanliness factor, C-243 to C-244
clutch type couplings, C-347
clutches, synchronous clutch couplings (*See also* power transmission), P-107 to P-115
coal
biomass co-gasification, CO_2 removal and, C-17 to C-18
cost vs. energy efficiency for, C-18 to C-20
coal gas, F-37
coal powered energy plants, integrated gasification combined cycle (IGCC), CO_2 removal and, C-14 to C-16
coalescer panels, in air filtration systems, A-52 to A-53
coating processes, computational fluid dynamics to model, C-7
coatings (*See also* metallurgy), M-31, M-34 to M-35
anticorrosion, C-442
chlorofluorocarbons in, O-14, O-16 to O-26
tank exteriors, T-5 to T-8
tank interior, for potable water, T-9 to T-13
tank interiors, T-4 to T-8
cobalt and cobalt alloys, M-41 to M-42, P-11 to P-31, P-40 to P-57
abrasives and, A-2
codes, commonly used, Sp-1 to Sp-3
codes, life cycle assessment (LCA), L-19
coding systems, color coding in, C-81
coefficient of discharge, flow measurement, M-24
coefficient of friction, worm gears and, P-86 to P-88
coefficient of performance (COP), refrigerants and, R-18
cogeneration, C-78 to C-84
life cycle assessment (LCA) and, L-17 to L-18
synchronous clutches and, P-107 to P-108, P-114 to P-115
coil type stab electric heater, H-14
"coincidence frequency," A-27, A-30
coke oven gas, F-37
coking processes, in petroleum industry, R-4
cold-to-cold expansion joints, E-58 to E-59
color coding, C-81
columns (*See also* towers and columns), T-40
combined cycle applications, C-78, P-107
combined cycle gas turbine (CCGT) plants, explosion hazard assessment for, E-67, E-70 to E-71, E-74, E-81 to E-85
combined flow coefficient, flow measurement, M-24
combined heat and power (CHP) plants, P-208, P-210
explosion hazard assessment for, E-67, E-70 to E-71, E-81 to E-85
combined unit systems, in air pollution control and, A-83, A-99 to A-101
combining processes, in petroleum industry, R-4
combustion, computational fluid dynamics to model, C-1 to C-7
combustion cans, material selection for, M-40 to M-41
combustion chamber, in gas turbines, T-49 to T-51
combustors (*See* turbines)
communication industries pollutants, P-29
commutators, carbon-graphite mix products and, C-13
comparators, optical, A-2
complex vibration, in vibrational analysis (VA), C-263
compounds, M-39
compressible fluid, in flow measurement, M-28 to M-29
compression, air purification using, A-102
compressors (*See also* axial compressors; centrifugal compressors; integrally geared compressors; isotherm compressors; reciprocating or laby compressors; rotating positive displacement compressors; turbocompressors), C-81 to C-232, C-264, Sp-5
air filtration systems and, A-75
air inlet manifolds, paint and corrosion damage in, T-74 to T-75
antisurge control in, C-359 to C-368
applications for, C-138

atmospheric conditions vs. power degradation in, T-65 to T-67
axial (*See* axial compressors)
axial inlet design, C-148
centrifugal (*See* centrifugal compressors)
chemical cleaners for, T-67
cleaning or washing of, T-59 to T-75
cleaning/washing equipment and procedures for, T-71 to T-75
control systems for, C-136
corrosion and corrosion inhibitors in, T-74 to T-75
debris in, A-59
detergent cleaning of, T-70
expanders (turboexpanders) using, C-151 to C-152
foaming cleaners for, T-67 to T-68
fouling of, T-63 to T-65, T-67
gas turbines and, T-47
horizontally split casing design in, C-142, C-166 to C-170
hot vs. cold water cleaning of, T-73 to T-74
integrally geared (*See* integrally geared compressors)
isotherm turbo (*See* isotherm turbocompressors)
offline cleaning of, T-67, T-70 to T-71, T-73 to T-74
online cleaning of, T-67, T-70 to T-73
reciprocating or labyrinthine (laby) (*See also* reciprocating or labyrinthine compressors)
rinsing after cleaning of, T-68
rotating positive displacement (*See* rotating positive displacement compressors)
rotors in, M-40
screw (*See* rotating positive displacement compressors)
single stage design, C-147 to C-148
stall in, T-48 to T-49
subsynchronrous frequency vibration, C-291 to C-294
surging in, T-48 to T-49
TURBAIR vacuum blowers using, C-149 to C-150
turbo- (*See* turbocompressors)
turbochargers and, T-112 to T-114
vertically split casing design, C-143 to C-144
water only cleaning of, online, T-69 to T-70
water- vs. solvent-based cleaners for, T-68 to T-69
computational fluid dynamics (CFD), C-1 to C-7
air flow testing using, E-72
applications of, C-1, C-3 to C-6
case studies in, C-7
CATAPOL project in, C-5, C-7
combustion and incineration modeling using, C-7
complex shape modeling using, C-5
cooling systems modeling using, C-7
die design using, C-4
electrostatic precipitator flow using, C-5
emissions modeling using, C-7
filter flow using, C-4, C-7
Fluent project in, C-6 to C-7
fluid handling and flow distribution modeling using, C-3
gasoline hydro treatment modeling (OLMES project) using, C-7
IcePak for, C-5, C-7
liquid natural gas studies using, L-21, L-23
materials processing modeling using, C-7
mixing of materials in tank using, C-3, C-6
OLMES project in, C-6 to C-7
polymerization temperature prediction using, C-5
reactor modeling using, C-3
separation of materials modeling using, C-7
slide valve and, flow through, C-2
solids concentration in riser using baffle plate, C-2
trajectory of exhaust plume using, C-3
ventilation modeling, C-6
windage heating (labyrinth seals) predictions using, C-6 to C-7
computer aided design/manufacturing (CAD/CAM), A-1, C-1
computer aided tomography (CAT) scan, M-37
computer numerical control (CNC), A-1 to A-3, C-1, N-1
condensate fuel, F-37
condensate pumps, NPSH and, P-236 to P-237, P-243
condensate receiving tank, in reciprocating (labyrinthine) compressors, C-109
condensers, C-233 to C-252
air cooler, C-235 to C-236
air inleakage in, C-247
backpressure and electrical output effects, C-233
cold end optimization in, C-244
corrosion in, C-245
cost-benefit analysis of, C-240
design and manufacture of, C-250
design layout of, C-234 to C-239
dome of, C-234 to C-235
environmental protection and, C-233
flashbox in, C-238 to C-239
fouling in, C-243 to C-244, C-248
foundation mounting and turbine connection, C-239
hotwell in, C-238
installation benefits in, C-240
layout philosophy in, C-244
makeup water in, C-240
material selection in, C-244 to C-246
oxygen content of condensate in, C-240
sacrificial anodes vs. corrosion, C-247
selection criteria, C-250 to C-252
shell of, C-236
sponge ball cleaning system for, C-248 to C-249
startup venting in, C-248
steam dump device (SDD) in, C-248 to C-250
steamside pressure losses in, C-242 to C-243
subcooling in, C-240
support plates in, C-236 to C-238
thermal performance of, C-239, C-242 to C-244
tube bundle, C-234, C-240 to C-242, C-244 to C-246
tubesheet materials in, C-246, C-251
type series of, C-244
venting for, C-246 to C-250
waterbox venting in, C-248
waterboxes in, C-238
condensing steam turbines and, multistage, T-94 to T-96
condition monitoring (CM) (*See also* measurement; vibrational analysis), C-253 to C-331
applications and benefits of, C-253, C-256 to C-257
basic components of, C-257
borescopes in, B-11
diagnosing problems using, C-299
diagnosis grids in, C-255 to C-256
gas path monitoring using, C-299
instrumentation requirements for, C-299
lifecycle assessment (LCA) and, C-253
nonintrusive wear monitoring in, C-330
packaged monitoring systems for, C-286 to C-287
packaged, selection criteria for, C-259 to C-260
parameters for, setting recommendations in, C-258 to C-259
performance analysis (PA) in, C-261, C-299
predictive strategies in, C-254
pressure measurement in, C-328 to C-330
preventive strategies in, C-254 to C-261
programmable logic controllers (PLCs) in, C-259
reactive strategies vs., C-254
retrofitting and, C-255
return on investment (ROI) for, C-299
risk management and, C-254
rules for, C-299 to C-301
scope of systems for, C-253 to C-261
selection criteria for, C-253 to C-261
spectrum analyzer for, C-298 to C-299
surge control in, C-328 to C-330
temperature monitoring in, C-260 to C-261
time between overhaul (TBO) and, C-256
troubleshooting philosophy using, C-298 to C-301
vibration analysis (VA) in (*See* vibrational analysis)
vibration monitoring (VM) in, C-257 to C-258
conductance method, in level sensing, C-388
conduction error, in temperature measurement, M-13
conductive polymers, P-4 to P-7
connectors, in soundproofing and noise control, A-24 to A-25
consolidated diaphragm comparator, M-22
constant fill couplings, C-347
construction industries pollutants, P-27 to P-28
Contex, P-4
continuity equation, pump, P-221 to P-224
continuous emissions monitoring (CEM), C-34
continuous reactors, R-1
control algorithms, for magnetic bearings, B-5, B-10
control panels, seals for, S-8
control regulators (*See* actuators)
control systems and controls (*See also* measurement; sensors; switches; valves), C-331 to C-353, Sp-1 to Sp-3, Sp-5
aeroderivative fuel regulator for gas turbine, retrofit of controls for, C-354 to C-358
airfoil movement and, C-336
antisurge control in, C-359 to C-368
current to pressure (I/P) converters for, C-392 to C-401
DCS in, C-355
dynamic conversion of electrical control signals, A-40
fault tolerance in, C-355
flow control in, C-333
fuel flow, C-403 to C-408, F-29 to F-37
in axial compressors, C-161
in electric heater, H-16 to H-20
in electric motors, E-10 to E-15
in generators and turbogenerators, G-18 to G-20
in hydrodynamic brakes, B-16 to B-19, B-23 to B-24
in magnetic bearings, B-5 to B-6, B-8 to B-11
integrated control system (Series 9500) for gas turbine generators, C-359 to C-361, C-374 to C-377
knife gate valves in, C-399, C-402 to C-403

liquid level/liquid flow control in, C-388 to E-18
load limit control in, C-334
lubrication systems, C-336
malsynchronization control, C-344
man machine interface (MMI) in, C-356
manifold flow (fuel) control in, C-403 to C-408
modular component units for, A-40
monitoring logic in, C-336
open loop, in magnetic bearings, B-10
overload/overspeed protection in, C-336, C-343 to C-350, C-368 to C-374
power limitation in, C-333, C-350 to C-354
pressure control in, C-333, C-368 to C-374, C-383 to C-388
process control in, C-333 to C-334
proportional conversion of electrical control signals, A-40
retrofit controls in, C-354 to C-408
safety systems and, C-332 to C-333
SCADA in, C-355
sensors for, C-377 to E-18
short circuit control, C-344
steam turbines and, T-78 to T-79
stepper motor–driven valves in, C-403 to C-405
switches for, C-377 to E-18
temperature limitation in, C-333, C-377 to C-382
torque control, C-344 to C-345
turbocompressors, C-331 to C-336
valves for, CE-18 to CE-392
vaporizers, V-25 to V-31
vibration limitation in, C-333
conversion factors, vaporizers, V-7
conversion of units, Sp-1 to Sp-3
conversion process, in petroleum industry, R-2 to R-4
conversion tables (*See* specifications list)
converters
current to pressure (I/P), C-392 to C-401
hydrodynamic power transmission and, P-138
ultrasonic cleaners and, U-2
conveyors (*See also* drives; power transmission), A-48, C-408, D-3, E-21 to E-22, Sp-5
coolant, engine, C-408 to C-412
cooler bearings, in electric motors, E-16
coolers, A-48
for agricultural and food products, A-48
condensers in, C-235 to C-236
cooling (*See also* air conditioning; refrigerants), C-413 to C-423
air conditioner sizing and, C-422 to C-423
capacities charts for air conditioners, C-416 to C-417
chlorofluorocarbons (CFCs) in, C-413
computational fluid dynamics modeling using, C-7
cooling towers in, C-423 to C-426
dairy coolers, C-412 to C-413, C-420
design specifications/standards for, C-413 to C-418
ejectors in, E-2 to E-3
electric heater controls, H-18
electric motors and, E-28 to E-29
flow through turbine, L-17
in generators and turbogenerators, G-1 to G-3, G-11 to G-12, G-16
hazardous gas environments, design specifications for, C-415, C-418
mean time between failure (MTBF) of products for, C-418
military standards/specifications (MIL-Std) in, C-414
NEC design specifications for, C-415, C-418
NEMA design specifications for, C-413 to C-414
Peltier Effect in, C-418
in reciprocating (labyrinthine) compressors, C-103
refrigerants in, C-413, R-10 to R-19
reliability of products for, C-418
solid state technology in, C-413
thermoelectric cooling process in, C-418 to C-423
UL/CSA design standards for, C-418
cooling towers, C-423 to C-426, R-8
evaporative, C-423 to C-426
fans, F-4
plume abatement design, C-424 to C-426
cooling valve, in reciprocating (labyrinthine) compressors, C-96
Cooper, C-81
coordinate measuring machine (CMM), A-2 to A-3
copper, P-11 to P-31, P-40 to P-57
conductivity of, vs. polymers, P-5
copper-nickel alloys, in condensers, C-245
copper resistance thermometer, M-4
corrosion, M-40
in air filtration systems and, A-55 to A-59, A-64 to A-71, A-76 to A-77
anticorrosion coatings, C-442
aqueous, M-40
in compressors, T-74 to T-75
in condensers, C-243 to C-245
electric motors and
hot, M-40
in isotherm turbocompressors vs., C-191 to C-193
ozone and, O-8 to O-9
sacrificial anodes vs., C-247
corrosion inhibitors, compressors and, T-68
corrosive gases, C-86
corrugated packaging (*See* packaging; paper products and packaging)
corrugated soundproofing panels, unlined, A-29 to A-33
couplings (*See also* power transmission)
in axial compressors, solid type, C-164 to C-165
brake properties at high speed and acceleration, C-347 to C-350
in centrifugal compressors, solid type, C-176 to C-178
clutch type, C-347
constant fill, C-347
hydrodynamic power transmission and, P-138 to P-147, P-155 to P-165
hydrodynamic principle in turbocouplings and, C-346 to C-347
in isotherm turbocompressors, solid, C-195 to C-201
resonant, C-317
Safeset overspeed/overload protective coupling, C-344 to C-350, P-184 to P-191
synchronous clutches and, P-107 to P-115, P-118
torque converters and, torque limiting, P-184 to P-191
turbo- (type T) in, P-191 to P-196
Type RK variable speed, P-147 to P-149
Type SVL variable speed, P-145 to P-147
type SVNL variable speed, P-142 to P-145
variable speed, C-347
covers, floating, F-12
cracking, of expansion joints, E-60 to E-61
cracking processes, in petroleum industry, P-2 to P-4
cranes and hoists, Sp-5 to Sp-6
crankcase, in reciprocating (labyrinthine) compressors, C-86, C-92, C-107
crankshaft, in reciprocating (labyrinthine) compressors, C-99, C-108
creep deformation, M-39 to M-41
cresol, P-11 to P-31, P-40 to P-57
critical damping, C-312 to C-313, C-315
crosshead fastening, in reciprocating (labyrinthine) compressors, C-86, C-93
crude oil (*See also* refineries, petroleum), F-37 to F-38, F-43 to F-45, R-1 to R-10
crushers, C-442
cryogenic separation, C-17
cryogenic technologies
using reciprocating (labyrinthine) compressors, in natural gas service, C-121 to C-126
turboexpanders and, T-114 to T-115
vaporizers in, V-5 to V-7, V-17 to V-19
crystalline nature of materials, M-39
crystallization
agitators in, A-38
toluene diisocyanate (TDI), C-43
crystallizers, C-62 to C-77
batch process for, C-64
benefits of, C-63 to C-66
capacities for, C-66
chlorobenezene separation in, C-68
drive system in, C-77
easily separable crystals and, scraped surface design for, C-76 to C-77
fatty acid separation in, C-68
fouling tendencies in
high boiling point rise vs., C-67
ideal vs. nonideal cases for, C-72
latent heat of, C-62
process of, C-62
purity of product in, C-63
scraped surface continuous crystallization process for, C-64 to C-66
solubility curves of chemicals in, C-66, C-71 to C-75
solvents and, C-70
temperatures for, C-66
vacuum crystallization process in, C-71
Van T'Hoff equation in, C-72 to C-75
vapor pressures and, C-67
viscous fluids and, C-67
waxes and, C-70
CSA, generators and turbogenerators design, G-1
Cultor Corporation and environmental accountability, E-42 to E-48
cumenes, P-11 to P-31, P-40 to P-57
current, electric motors requirements, E-10, E-12 to E-13, E-15, E-29
current-to-pressure (I/P) converters, C-392 to C-401
current transformers, in generators and turbogenerators, G-19
CVA8 type pumps (API 610) overhung vertical inline integral bearing frame flexibly coupled, P-292 to P-653
CVAR type rigid coupled pumps, P-296
cyanides, P-11 to P-31, P-40 to P-57
cyanoacetamide, C-68
cyclic usage factor, life cycle assessment, L-3 to L-6
cyclohexane, P-11 to P-31, P-40 to P-57
cyclone tube separators, S-12, S-17 to S-18
cylinder velocity recovery capability, fans, F-4
cylinders, in reciprocating (labyrinthine) compressors, C-103, C-126 to C-132
cylinders for holding toxic chemicals, C-52 to C-54

dairy coolers, C-412 to C-413, C-420
dampeners (*See also* pulsating dampeners)
in fuel gas conditioning system, F-27
pulsation type, P-214 to P-218
damping, A-28, A-33, C-264, C-307 to C-313
amplitude in, C-315 to C-317
critical damping in, C-312 to C-313, C-315

displacement and, C-314 to C-315
forced vibration in, C-313 to C-316
friction (coulomb) damping in, C-307
inertia in, C-314 to C-315
magnification factor (D/F/k) in, C-316
natural frequency in, C-316
overdamping and decay in, C-312
phase angle in, C-315 to C-316
solid or structural damping in, C-307
steady-state oscillations and, C-313 to C-314
underdamping and decay in, C-312 to C-313
vibrational analysis (VA), C-305 to C-306
viscous damping in, C-308 to C-311
DCS control systems, C-355
DDT, P-7
dead band, in governors, T-102
deadweight tester and gauges, M-18 to M-20
decabromodiphenyl oxide, P-11 to P-31, P-40 to P-57
deceleration, electric motors, E-29
deforestation effects, F-12
deformation, M-39, P-102, P-104
degree of freedom, in vibrational analysis (VA), C-301 to C-302, C-310
Demag Delaval, C-219 to C-228, C-233 to C-252, C-377 to E-18, L-25 to M-31, P-74 to P-98, P-218 to P-281, T-112 to T-115
demand charge, energy costs and, E-9 to E-10
DeNox systems, in air pollution control, A-90, A-93, A-100
dense phase technology, air pollution control and, A-98
density at reference temperature (DENS.REF), T-25 to T-27
DEPAC system, air pollution control and, A-98 to A-99
Department of Defense, cement specifications, C-22
derating of electric motors, E-29
desalination, D-1 to D-2
metallurgical content of seawater and, D-2
multistage flash (MSF) process in, D-1
reverse osmosis (RO) process in, D-1
desiccants, D-4
detergent cleaners, C-78, T-68, T-70
deterioration of gas turbines, L-10 to L-11
deterioration of steam turbines, L-11
deterministic vibration, in vibrational analysis (VA), C-263
diagnosis grids, in condition monitoring, C-255 to C-256
dialysis, D-2
diaphragm gauges, M-21
diaphragm valves, CE-18 to CE-392
diaphragms, in steam turbines and, T-77
dibutyl cresol, C-68
dibutyl phthalate, P-11 to P-31, P-40 to P-57
dicalcium silicate (C2S), C-25 to C-26
dichloro-2-butylene, V-15
dichlorobenzene, P-11 to P-31, P-40 to P-57
dichloroethane, P-11 to P-31, P-35, P-40 to P-57
dichloromethane, P-11 to P-31, P-35, P-40 to P-57
dieldrin, V-15
dies, computational fluid dynamics to analyze shapes in, C-4
diesel engines, E-38
diesel fuel, F-37
diethanolamine, P-11 to P-31, P-40 to P-57
diethyl ether, V-16
diethyl phthalate, P-11 to P-31, P-40 to P-57
differential pressure, C-389, M-14, M-18 to M-24
differential pressure-actuated switches, C-383
diffuser, in isotherm turbocompressors, cross section, C-213
diffusion rate, M-39
diglycerides, C-68
dilution ventilation, in preventing explosion, E-87
dimensioning, dimensions, Sp-6
ASME certification program for, Sp-30
dimerization, C-58, C-72
dimethyl ether, V-16
dimethyl hydantoin
dimethyl sulfate, P-11 to P-31, P-35, P-40 to P-57
dimethyl terephthalate, C-68, V-15
dimethylaminomethylene, freeze dry production of, D-17
dinitro-o-cresol, P-11 to P-31, P-40 to P-57
dinitrotoluene, P-11 to P-31, P-40 to P-57
di-n-octyl phthalate, P-11 to P-31, P-40 to P-57
dioxane, P-11 to P-31, P-35, P-40 to P-57
dioxin, A-96 to A-97, P-7
discharge head, in pumps, P-223 to P-224
discs, turbine, material selection for, M-40
dislocations, M-39
displacement, vibrational analysis (VA), C-287, C-314 to C-315
displacement probes, in vibrational analysis (VA), C-266 to C-270
displacement pumps, P-260, P-267 to P-268
disposal of toxic chemicals, C-62
distillate No. 2 (*See* diesel fuel)
distillation, C-62, C-67, D-2 to D-3
atmospheric, in petroleum processing, R-2
computation fluid dynamics in, C-1 to C-7
vacuum, in petroleum processing, R-2
distortion (physical), expansion joints vs., E-61
diversion systems, high voltage and lightning strikes, T-30 to T-31
diverters, D-3
doctor, in paper industry, D-3
dome, condenser, C-234 to C-235
Dorr-Oliver Inc., centrifuges, C-35 to C-38
double volute pumps, P-247 to P-248
Dowtherm heater, H-22
drain valve and drains, in air filtration systems, A-56, A-62 to A-63
drawings and terminology, Sp-6 to Sp-7
drilling, M-35
drive belts, vibrational analysis (VA) of, C-296
drives and drive systems (*See also* power transmission), D-3 to D-4
in centrifugal compressors, C-188
in crystallizer, C-77
hydrokinetic (*See* hydrodynamic power transmission)
in isotherm turbocompressors, C-194 to C-200
drug industries pollutants, P-29
drum (*See* separators)
drums for holding toxic chemicals, C-54
dry cleaning products, C-13
dry cooling towers, C-423 to C-426
dry scrubbers, S-10 to S-29
capacity of, S-11 to S-12
cyclone tube in, S-12, S-17 to S-18
design of, S-11 to S-12
filter for, S-13 to S-16
fuel gas conditioning system, F-28 to F-29
optional features of, S-12 to S-13
pressure loss in, S-12
removable tube section in, S-13
drying (*See also* freeze drying), D-4 to D-18
agitators for, D-4
computation fluid dynamics in, C-1 to C-7
dessicants for, D-4
ejectors in, E-2 to E-3
fans for, D-4
fluid bed dryers in, D-4
heat exchangers for, D-4
ovens for, D-4
vacuum drying, D-4, D-12 to D-13
DS F88/146, A-18
ductility, of ceramics, C-38
ducts and ducting (*See also* expansion joints), D-18
dust, vs. air filtration systems, A-51 to A-52, A-56, A-66, A-70, A-74 to A-76
duty cycle, electric motor, E-5
dynamic absorbers, vibration vs., C-276
dynamic conversion of electrical control signals, A-40
dynamic water eliminator (Hydra) filters, A-81

earth loop heat pump, H-1 to H-3
ecolabeling, E-54 to E-55
ecological industrial parks, A-48 to A-49, E-1
ecosystem management, F-12
ecosystems, E-1 to E-3, F-12
eddy current probes, C-267 to C-268
eddy current testing, M-36
efficiency
of electric motors, E-5 to E-6
of gas turbines, T-43 to T-44
of pumps, temperature vs., P-254 to P-255
of steam turbines, T-82, T-84, T-89 to T-90
of worm gears, P-85 to P-88
ejectors, E-2 to E-3, N-1
elastic deformation, M-39
elastic gauges, M-20 to M-22
elbow flowmeters, C-330
electric control signals (*See* control systems)
electric heaters (*See* heaters, electric)
electric hot pipe tank heaters, H-12 to H-15
electric motors and controls, A-48, E-3 to E-36
AC induction motors used for variable frequency control, E-18 to E-35
acceleration in, E-3 to E-5, E-12, E-24, E-29
accessory devices for, E-35
adjustable frequency motors as, E-10 to E-15
applications for, cost effective solutions, E-6
breakdown rpm region in, E-24 to E-25
breakdown torque point in, E-12 to E-13
breather drains in, E-18
carbon-graphite mix products for brushes in, C-13
cast iron construction, in high corrosion environment, E-15 to E-16
centrifugal compressors, C-187 to C-190
controllers for, motor/controller combinations in, E-10
conveyor powering application, E-21 to E-22
cooler bearings in, E-16
cooling in, E-28 to E-29
corrosion prevention in, E-18
cost, energy costs of, E-7 to E-10
cost, operating cost of, E-6 to E-7
current requirements of, E-10, E-12 to E-13, E-15, E-29
deceleration in, E-29
demand charge and, energy costs of, E-9 to E-10
design/parameter considerations in, E-23 to E-24
duty cycle requirements in, E-5
efficiency of, components affecting, E-5 to E-6
enclosures for, environment of operation and, E-15
energy use in, E-3 to E-5
fans and, E-17, E-28 to E-39
flexibility of mechanical design in, E-34 to E-35
flux level in, motor flux, E-32
frame design vs. performance in, E-34 to E-35

frequency requirements of, E-23 to E-24, E-29, E-35
friction in, E-5 to E-6
gaskets in, E-17
harmonics in, E-14, E-33 to E-34
heating, heat transfer (*See* insulation and temperature parameters, below)
high performance drives and, E-23
horsepower requirements of, E-23 to E-24
induction motors as, E-10 to E-15
insulation and temperature parameters for, E-14 to E-15, E-25 to E-28, E-31 to E-32
kilovolt-ampere (KVA) rating in, E-8 to E-9
load requirements in, E-5 to E-7
load torque requirements for, E-11 to E-15
lubrication in, E-15 to E-16
maintenance of, E-7
NEMA design C and D motors in, E-14
noise of, A-24, E-32 to E-34, E-28
oil mist lubrication in, E-16
overload torque in, E-24 to E-25
oversizing (derating) of, disadvantages of, E-29, E-34
parasitic forces and noise in, E-33 to E-34
peak current rating in, E-25 to E-26
positive lubrication system (PLS) in, E-15 to E-16
power factor in, E-7 to E-9
power transistor technology and, E-29 to E-32
pulse wave modulation (PWM) and, E-32
pumping applications for, E-22
reactive kilovolt-ampere (KVAR) rating in, E-8 to E-9
real power kw consumed by, E-7 to E-8
RMS current rating in, E-25 to E-26, E-28
rotor in, E-20
sizing considerations for, E-34
speed range of, E-15
speed torque curves for, E-3 to E-4, E-10 to E-15
speed vs. frequency in, induction type, E-13 to E-14
squirrel cage type, E-19
starting torque in, E-14, E-24
stator in, E-19
synchronous speed point in, E-12 to E-13
temperatures parameters for, E-14
testing of, E-18 to E-22
torque requirements in, E-23 to E-25
transition voltage (dV/dt) time, E-30
vibrational analysis in, E-18 to E-22, E-33
voltage requirements of, E-15, E-23 to E-24, E-29, E-32
voltage unbalance and operating cost in, E-6 to E-7
V-ring shaft slinger (Buna-N shaft seal) in, E-17
electric plastics, P-3 to P-7
Electric Power Research Institute (EPRI), C-351
electric transducers, M-22 to M-24
electric vehicles, conductive polymers in, P-6
electrical and electronics products industries pollutants, P-22 to P-23
electrical contacts, carbon-graphite mix products and, C-13
electrical discharge machining (EDM), M-35
electrical radiant furnace vaporizers, V-5 to V-6
electrical system, in reciprocating (labyrinthine) compressors, C-110
electrodialysis, D-2
Electrofin electric heaters, H-8 to H-12
electrohydraulic actuators, A-37
applications for, A-39
components of, A-41
control regulator function and main features of, A-42
conversion of electrical control signals in, components for, A-40
drive/control value options in, A-45
fuel system, F-31
power regulated electromagnet in, A-41
regulating valves using, A-43
regulator using, electrically controlled, A-44
sensors and, A-46
speed protection device using, A-47 to A-48
technical data for electronic component assemblies in, A-45
electrohydraulic governors, T-104 to T-106
electromagnetic bearings (*See* magnetic bearings)
electromagnets, electrohydraulic actuators using, A-41
electromechanical (displacement) method, in level sensing, C-389
electronic data loggers, for vibrational analysis (VA), C-270
electrostatic precipitators
in air pollution control, A-83 to A-85, A-102
computational fluid dynamics to analyze flow through, C-5
elevators and escalators, Sp-7 to Sp-8, Sp-31 to Sp-32
elutriating stream concept, centrifugation, C-35
emergency power generation (*See* engines, gas)
emissions (*See also* air pollution; carbon dioxide), E-36 to E-38, E-50 to E-52, F-20, P-57 to P-60
acid rain, Canada vs. U.S., A-10 to A-12
boiler, B-11
carbon dioxide (CO_2) (*See* carbon dioxide)
carbon monoxide (CO) as, E-38, P-62 to P-64
CCME (Canada) National Emission Guidelines for kilns, C-28 to C-34
cement kilns and, C-22, C-28 to C-34
computational fluid dynamics to model, C-1 to C-7
continuous emissions monitoring (CEM) in, C-34
crude/heavy fuel oil, F-44 to F-45
environmental accountability, E-40 to E-49
environmental economics of, E-50 to E-52, E-54
fossil fuel, E-36
fuel system control, F-30
gasoline powered engines, E-39
guidelines for, stationary combustion turbines, E-36 to E-38
liquefied natural gas (LNG), F-41
naphtha, F-43
nitrogen oxide (NO_x), F-43, F-45, E-36 to E-38, E-50
ozone, O-4 to O-26
particulate, E-36
power production and, P-72
refrigerants and, R-13
sulfur oxides (SO_x) (*See* sulfur oxides)
emissivity, M-10
emulsification, agitators in, A-38
enclosures, gas turbine (GT) explosion hazard assessment for, E-67, E-70 to E-71
end product handling, air pollution control and, A-83, A-97 to A-98
endrin, V-15
energy efficiency, in isotherm turbocompressors, C-190 to C-191
energy recovery, synchronous clutches and, P-107 to P-108, P-114 to P-115
energy use, pulp and paper industry, P-204 to P-205
Enertran (heat pumps), H-2 to H-6
engine condition monitoring (ECM) (*See* condition monitoring)
engine condition–monitoring systems (ECMS) (*See* condition monitoring)
engines, gasoline powered, E-38 to E-40
coolant in, C-408 to C-412
diesel, E-38
emissions from, E-39
four stroke, E-38, E-40
reciprocating, E-38 to E-39
scavenging in, E-39
torque converters and in, P-172 to P-174
turbochargers for, E-38 to E-39
two stroke, E-38, E-40
enthalpy, steam turbines and, T-82, T-84
Enthone Inc., P-4
entrained air/gases, vs. pump performance., P-271 to P-272, P-278 to P-281
ENVAR biodegradable plastics, P-2
Environment Canada, A-13, O-4 to O-26, P-8, R-1 to R-10
environmental accountability, A-40 to A-49
accidents/incidents and, E-46
environmental economics and, E-49 to E-56
feeding management (livestock) and, E-44
ISO VILJA technology in, E-46, E-48
life cycle inventories (LCIs) in, E-43
performance table for, E-45
reports of, E-47 to E-48
risk management and, E-44
environmental air monitoring (*See* air pollution control; emissions)
environmental controls, computational fluid dynamics to model, C-7
environmental degradation of plastic, P-1
environmental economics, E-47 to E-56
biological diversity and, E-50
cost-benefit analysis for, E-51 to E-56
ecolabeling in, E-54 to E-55
emission reduction and, E-50 to E-52, E-54
environmental accountability and (*See* environmental accountability)
environmental management systems (EMS) in, E-52
forest preservation and, E-52
key ratios in, E-51
life cycle assessment (LCA) in, E-55 to E-56
packaging, resource efficient, E-52 to E-54
profitability, long term, E-52 to E-55
research and development in, E-52
taxes and charges for environmental protection and, E-51
waste reduction, E-54
"yardsticks" of, E-50 to E-54
Environmental Management System (EMS) standard in, F-13 to F-16
environmental management systems (EMS), E-52
environmental protection, Sp-1 to Sp-2
accidents/incidents and, E-46
agricultural and food products, A-49
biological diversity and, E-50
biotopes, F-13
condensers and, C-233
deforestation effects and, F-12
ecolabeling in, E-54
ecolabeling in, E-55
ecological parks, E-1
ecosystems, E-1 to E-3
emission reduction and, E-50 to E-52
emission reduction and, E-54
environmental accountability, E-40 to E-49
environmental economics in, E-49 to E-56
Environmental Management System (EMS) standard in, F-13 to F-16
environmental management systems in, E-52

forest preservation and, E-52
forest product industry, F-12 to F-24
key ratios in, E-51
life cycle assessment (LCA) in, E-55 to E-56
packaging, resource efficient, E-52 to E-54
plastics and, P-1 to P-2
"red tide" algae (*Ptychodiscus brevis*) and, C-78
reindeer, F-16
taxes and charges for, E-51
ultrasonic cleaners and, T-5 to T-6
waste management and, W-1 to W-12
waste reduction, E-54
"yardsticks" of, E-50 to E-54
Environmental Protection Agency (EPA), E-40 to E-49, P-8
enzymes, E-43
epichlorodyndrin, P-11 to P-31, P-36, P-40 to P-57
epoxy, as interior tank coating, T-4
Epson, P-5
equivalent series resistance (ESR) polymers, P-6
erosion, in air filtration systems, A-75 to A-78
erythrobates, E-44
ethane, V-16
ethanol, C-418, F-37, V-15
ethanolamine, V-15
ethoxyethanols, P-11 to P-31, P-40 to P-57
ethyl acrylate, P-11 to P-31, P-36, P-40 to P-57
ethyl chloride, V-15
ethyl ether, C-415
ethylbenzene, P-11 to P-31, P-40 to P-57
ethylene, C-418, P-11 to P-31, P-40 to P-57, V-16 to V-17
ethylene dichloride, V-7, V-15
ethylene glycol (EG), C-409 to C-412, P-11 to P-31, P-40 to P-57
ethylene oxide, C-415, P-11 to P-31, P-36, P-40 to P-57, V-7, V-15
eutrophication, P-64 to P-65
evaporative coolers (*See* chillers; dairy coolers)
evaporative cooling towers, C-423 to C-426
exhaust gas recovery, C-78
exhaust gas temperature monitoring, C-260 to C-261
exhaust plume, computational fluid dynamics modeling of, C-3
exhaust stacks (*See* stacks)
exhausters, centrifugal gas, E-56 to E-57
existing refinery, R-8
expanded refinery, R-8
expanders, turboexpanders, C-151 to C-152, T-114 to T-115
synchronous clutches and, P-107
expansion joints, D-18, E-57 to E-67
ambient temperatures and, E-64 to E-65
axial/vs. axial movement, E-63
bias cutting of, E-65
cold-to-cold, E-58 to E-59
composition of, E-59 to E-60
cracking damage vs., E-60 to E-61
differential thermal expansion in, E-58
distortion damage vs., E-61
fabric vs. metallic, E-60
flexible elements of, E-60
floating sleeves in, E-65 to E-66
heat/vs. heat damage, E-61 to E-62
hot-to-cold, E-58 to E-59
hot-to-hot, E-58 to E-59
insulation and, E-66
insulation pillows and, E-65 to E-66
mounting flanges and, E-63 to E-64
noise vs., E-66 to E-67
outer cover of, E-60
pantograph control mechanism and, E-63
proximity of building to, E-64 to E-65
PTFE in, E-60
radial growth of steel parts vs., E-64 to E-65
support/vs. support frame damage, E-60
thermal barrier of, E-60
water washing and, E-64
explosion hazard assessment, E-67 to E-87
abrasive materials, A-8
air movement studies in, E-67, E-71
American National Standards Institute (ANSI) codes for, E-87
American Petroleum Institute (API) code for, E-86
as low as reasonably practicable (ALARP) risks and, E-80
British Gas Code for, E-86 to E-87
case study in, gas turbine power plants (UK), E-81 to E-85
combined cycle gas turbine (CCGT) plants and, E-67, E-70 to E-71, E-74, E-81 to E-85
combined heat and power (CHP) plants and, E-67, E-70 to E-71, E-81 to E-85
dilution ventilation vs., E-87
electric heater, H-17 to H-18
fault tree logic diagrams for, E-79 to E-80
flammable materials and, E-67, E-76 to E-78, E-84, E-86
fuel leaks and E-69 to E-70, E-76, E-82 to E-85
full air flow balance tests in, E-71
gas detection vs., E-70
gas turbine (GT) enclosures and, E-67, E-70 to E-71
hazard analysis (HAZAN) in, E-68, E-379 to E-80
hazardous operations (HAZOP) review in, E-75 to E-76
Health and Safety at Work (HSWA) Act (UK) and, E-87
identification of hazards in, E-71 to E-76
ignition sources and, E-67, E-78 to E-379
incidence of explosion and, E-84 to E-85
Institute of Petroleum (IP) code for, E-87
liquefied natural gas (LNG), F-39
lower explosion limit (LEL) in, E-70, E-76
methane and, E-76 to E-77
naphtha, F-42 to F-43, E-76
National Fire Protection Association (NFPA) codes for, E-86
natural gas and, E-76
operating conditions vs., E-74
process hazard review (PHR) in, E-67, E-74 to E-76
propane and, E-76
risk based approach to, E-71, E-75 to E-78, E-80 to E-81
smoke tests in, E-73
standards for, E-86 to E-87
static pressure checks in, E-72
tanks and, T-28 to T-29
Teeside Power Station (UK) example of, E-67 to E-87
3–D velocity grid test in, E-72 to E-73
ventilation vs., E-70, E-76, E-87
external gear pumps, P-260 to P-264
extraction governors, T-106
extraction processes, E-44
agitators in, A-38
liquid-solid, E-87
extraction steam turbines, T-98 to T-100
extrusion die, computational fluid dynamics modeling of, C-4
extrusions, computational fluid dynamics to model, C-7
fabric filters, in air pollution control, A-83, A-85 to A-86
fabricated metal industries pollutants and, P-20 to P-21
Fahrenheit temperature conversion, M-12
fans, C-136, F-1 to F-5
acceleration of, F-3
air density vs., F-1, F-3
air horsepower (ahp) of, F-5
applications for, F-2
blades in, F-1 to F-3
capacity (cfm) of, F-1 to F-3
centrifugal fan types, F-1
computational fluid dynamics to model shapes in, C-5
cooling towers and, F-4
cylinder velocity recovery capability of, F-4
drying, D-4
in electric motors, for cooling, E-17, E-28 to E-29
forward vs. backward curved blades in, F-1
in generators and turbogenerators, G-6, G-8, G-11
horsepower of, F-1, F-5
laws of operation in, F-1 to F-2
mass flow rate of, F-4
net flow area of, F-3
performance range, by blade type, F-2 to F-3
speed (rpm) of, F-1
static pressure (hs) of, F-1 to F-5
total pressure differential of, F-4
variable pitch, F-3
velocity of, F-3 to F-4
velocity pressure of, F-4
water density vs., F-3
far fields, in sound, A-20
fast fourier transform (FFT), sound intensity meters using, A-16
fast response valves, C-405 to C-408
fasteners, Sp-8 to Sp-10
fatal accident rate (FAR), E-80
fatigue, M-40
fatty acids, crystallization of, C-68 to C-72
fault tolerance, in control systems, C-355
fault tree logic diagrams, in explosion hazard assessment, E-79 to E-80
federal government service industries, pollutants and, P-31
feeding management (livestock), environmental accountability and, E-44
feedwater heating, in steam turbines, regenerative, T-85 to T-87
fermentation, E-44
ferric oxide (Fe_2O_3), C-25
ferrous alloys, abrasives and, A-2
fertilizers, V-15
fiber forming, computational fluid dynamics to model, C-7
filled system thermometers, M-8
fillers, A-1
Filpac TCR system, A-100 to A-101
Filsorption system, A-96 to A-97, A-100 to A-101
filter designs and materials (*See also* air filtration; air pollution control), C-109, F-5 to F-12
air (*See* air filtration; air pollution control)
air purification using, A-102
carbon-graphite mix products and, C-13
cartridges for, F-6 to F-8
closures for, F-6
computation fluid dynamics in, C-1 to C-7
construction of, F-10
fuel gas conditioning systems, F-25, F-28
gas filters, F-6 to F-11
in line gas, F-10 to F-11

multitube, F-9 to F-10, F-28
optional features in, F-10
separators and, S-13 to S-16
sizing and performance of, F-9 to F-10
filter in/filter out amplitude vs. rpm, in vibrational analysis, C-276 to C-278
filtered signal, vibrational analysis (VA), C-263
fin tube type electric heaters, H-9 to H-12
fire hazards
tanks and, T-28 to T-29
toxic chemicals, C-59
fire ratings, soundproofing materials, A-36
first aid procedures for toxic chemicals, C-61
fittings, Sp-22 to Sp-23
flame arrestors, V-31
flammable materials, in explosion hazard assessment, E-67, E-76 to E-78, E-84 to E-86
flanges, Sp-22 to Sp-23
synchronous clutches and, P-118
flapper valves, D-3
flare carbon, vs. air filtration systems, A-74 to A-75
flare stacks (*See also* stacks), F-12
flashbox, condensers, C-238 to C-239
flexible AC transmission system (FACTS), C-353 to C-354
flexible connectors, in soundproofing and noise control, A-24 to A-25
flexible couplings, synchronous clutches and, P-118
float and tape tank gauges, T-18
float valves, vaporizers, V-25 to V-28
floating covers, F-12
floating sleeve expansion joints, E-65 to E-66
flow control, C-333 to C-334
in hydrodynamic power transmission, P-128, P-131, P-133
in fuel systems, F-29 to F-37
in stacks, design parameters affecting, S-30 to S-33
in turbocompressors, C-333
pulsation dampeners in, P-214 to P-218
flow errors, M-15
flow measurement (*See also* liquid level/liquid flow control), C-389 to C-390, M-24 to M-31, Sp-10
adiabatic expansion factor in, M-25
Bourdon tube in, P-224
capacity of flow in, M-24 to M-25
coefficient of discharge in, M-24
combined flow coefficient in, M-24
compressible fluid, M-28 to M-29
Delaval applications and, M-29 to M-30
differential pressure in, M-24
flow rate calculations in, M-27 to M-30
incompressible fluid, M-28
manometer differential pressure in, M-24
net expansion factor in, M-25
nomenclature and terms in, M-24 to M-25
nozzle and, M-23, M-25 to M-26, M-29
numerical factor in, M-24
orifice and, M-23, M-25, M-29 to M-30
pipe diameter in, M-24
primary element construction and installation for, M-25 to M-27
rate of flow in, M-25
Reynolds number in, M-25
specific heats ratio in, M-24
specific weights in, M-25
thermal expansion factor in, M-24
throat area, M-24
throat diameter in, M-24
velocity of approach factor in, M-24
venturi tube and, M-23, M-26, M-29
flow modeling, computational fluid dynamics, C-3
flow nozzles (ASME), C-330, M-23 to M-29
flow rate calculations, M-27 to M-30
flow resistivity in absorptive materials, A-34
flow stream pressure, M-15
flow switches, C-389
flow-through bioassay, R-9
flow transducers, C-328 to C-330
flowmeters, C-330
Flowserve, USA, S-1 to S-9
flue gas desulfurization (FGD), A-83, A-86, A-89 to A-90
Fluent project, C-6 to C-7
fluid bed dryers, D-4
fluid catalytic cracker (FCC), pulsation dampener in, P-217
fluid handling, computation fluid dynamics in, C-1 to C-7
fluidization, computation fluid dynamics in, C-1 to C-7
fluidized bed, F-12
fluids, hydrodynamic power transmission and, P-123
fluoride ion cleaning, M-34
flux level, motor flux, E-32
fly-ash handling, A-97 to A-98, C-28
foaming cleaners, T-67 to T-68
foams, chlorofluorocarbons in, O-14, O-16 to O-26
food industries (*See also* agriculture and agricultural products)
dairy coolers, C-412 to C-413, C-420
irradiation technology for, I-3 to I-5
pollutants and, P-14 to P-15, P-29
Food Technology Service Inc., food irradiation facility, I-3 to I-5
forced feed type lubrication, L-25
forced vibration, C-264, C-301, C-313 to C-316, C-322 to C-323
forcing frequency in, C-264
forest preservation, E-52
forest product industry (*See also* pulp and paper mills), A-48, F-12 to F-24
AssiDomän forestry management case study in, F-13 to F-18
barrier coating production and, F-15, F-21 to F-23
biotopes in, F-13
carbon dioxide emissions, F-18
contemporary forest and timber industry practices in, F-12 to F-21
corporate environmental objectives of, F-14
deforestation effects and, F-12
ecological balance sheet in, F-16 to F-18
ecological "footprint" of, F-22
emissions control in, F-20
Environmental Management System (EMS) standard in, F-13 to F-16
environmental work list, sample year 1997, F-20
Forest Stewardship Council (FSC) (Sweden) in, F-13
forestry work in, F-13 to F-16
key environmental data for AssiDoman, F-25
packaging and paper products in, F-21 to F-26
reindeer herding as adjunct to, F-16
sawmill and wood processing operations, F-16, F-19
seedlings for, F-21
thinning and regeneration in, F-14 to F-18
timber production in, F-18 to F-21
waste management in, F-20
water conservation and, F-17
Forest Stewardship Council (FSC) (Sweden), F-13
forging, M-43
form scan geometry gauges, A-2
formaldehyde, P-4, P-11 to P-31, P-36, P-40 to P-57, V-7, V-15
forming, computation fluid dynamics in, C-1 to C-7
fosfomycin sodium (FOS), freeze drying and, D-13, D-17
fossil fuel, F-37
ASME certification program for, Sp-30
emissions from, E-36
fouling
in air filtration systems and, A-76, A-78
in condensers, C-243 to C-244, C-248
four-stroke engines, E-38 to E-40
fractional distillation, D-2 to D-3
fractionating column, D-3
fractionator, R-8
fracture, material M-39 to M-40
free vibration, C-264, C-301, C-305 to C-306
freeze drying (*See also* drying), D-4 to D-18
applications for, D-8
bioabsorbable polymer production using, D-17
clothing for TBA handling operators in, D-5 to D-7
ejectors in, E-2 to E-3
fosfomycin sodium (FOS) production using, D-13, D-17
frozen TBA water mixtures in, D-8 to D-9
glass powder production using, D-18
lipids and liposomes, with TBA, D-13 to D-17
liposoluble platinum (II) production using, D-17
macromonomer solution and TBA in, D-18
mass transfer–accelerator effect of TBA in, D-8
nanoparticle drug dispersant production using, D-17
N-methyl pyrrolidone (NMP) as solvent in, D-13
pharmaceutical industry use of, D-4 to D-8
physical properties of, D-5
porous resistance of dry product layer vs. TBA in, D-8, D-10 to D-11
scanning electron microscopic (SEM) studies of TBA solutions in, D-8, D-15 to D-16
sublimation in, D-8
sublimation rates of TBA/water solution in, D-5, D-7
sugar/sucrose and TBA solutions in, D-8 to D-18
TBA–water phase diagram in, D-5 to D-6
tertiary butyl alcohol (TBA) in, D-5 to D-18
vacuum dryer for, D-12 to D-13
water as solvent in, drawbacks of, D-4 to D-5
water-unstable drugs and, D-17
freeze protection, in vaporizers, V-12 to V-13, V-17, V-19
frequency, C-270 to C-272
electric motors requirements, E-13 to E-14, E-23 to E-24, E-29, E-35
sound, A-28
subsynchronrous frequency vibration, C-291 to C-294
in vibrational analysis (VA), C-262, C-272, C-287, C-301 to C-302
frequency control, variable AC induction motors used for, E-18 to E-35
friability of abrasives, A-3 to A-4, A-6
friction, coefficient of, carbon-graphite mix products, C-9
friction, friction loss
in electric motors, E-5 to E-6
in pumps, P-222
in worm gears, P-86 to P-88
friction (coulomb) damping, C-307

Frövi Light paperboard, E-52
FTSI, food irradiation facility, I-2 to I-3
fuel, F-37 to F-45
  oil sands and, O-1
  synthetic crude, O-1 to O-2
fuel and fuel systems
  life cycle assessment (LCA) for, L-8 to L-10
fuel gas conditioning systems
  dampeners, F-27
  filter separator in, F-25
  filter separators in, F-28
  heater, F-27
  line separators in, F-29
  metering, F-27
  multitube filters in, F-28
  pressure regulation station/valves (PRS/PRV) in, F-25 to F-26
  separators for, range of, F-28 to F-29
  vertical absolute separators in, F-29
  vertical dry scrubber in, F-25, F-28 to F-29
  vertical gas separators in, F-28
fuel gas conditioning systems, F-24 to F-29
fuel leaks
  explosion hazard assessment and, E-69 to E-70
  explosion hazard assessment and, E-76
  explosion hazard assessment and, E-82 to E-85
fuel regulation, C-403 to C-408
fuel regulators
  current to pressure (I/P) converters for, C-392 to C-401
  eroderivative fuel regulator for gas turbine, C-354 to C-358
fuel skid, F-34 to F-36
fuel systems and control
  actuators for, F-30, F-31, F-33 to F-34
  emissions control in, F-30
  features of, F-30 to F-33
  schematic of, F-32
  servomotor actuated plug metering valve in, F-36
  skid for, fuel skid, F-33 to F-34
  testing of, F-34 to F-36
  valves for, F-30, F-32 to F-33
fuel systems and control, F-29 to F-37
fuels, V-16
  biomass, P-205 to P-214
  black liquor, P-205 to P-214
  crude oil, F-38, F-43 to F-45
  fossil fuels, F-37
  gaseous fuels, F-37
  heavy fuel oil, F-38, F-43 to F-45
  impurities in, L-9
  liquefied natural gas (LNG), F-38 to F-45
  liquefied petroleum gas (LPG), F-38, F-41 to F-42
  liquid fuels, F-37
  liquid natural gas (LNG), L-21 to L-24
  naphtha, F-38, F-42 to F-43
  water in, L-9
  white liquor, P-205 to P-214
fumigation process, ejectors in, E-2 to E-3
functional food, E-44
furniture and fixture industries, pollutants and, P-18

gamma radiation, food irradiation using, I-1 to I-5
gas actuated sensors, C-378
gas exhausters, E-56 to E-57
gas filters, F-6 to F-11
gas path component management systems, M-41
gas path monitoring, C-299
gas pipeline (*See* pipeline and pipeline pumping stations)
gas seals (*See also* seals), S-1 to S-9
  advanced pattern groove (APG) in, S-3 to S-4
  GASPAC compressor seals, S-1 to S-9
  J ring, S-5 to S-9
  O rings, S-1, S-5
  polytetrafluoroethylene (PTFE) in, S-5 to S-9
  retrofitting of, S-8 to S-9
  symmetrical T groove (SMT) in, S-4 to S-5
gas separator, horizontal, S-27 to S-29
gas separators, vertical, S-22 to S-24
gas treating, R-5
gas turbines (*See* turbines, gas)
gaseous fuels, F-37
gaskets, E-17, Sp-22 to Sp-23
gasoline, C-418
gasoline hydro treatment modeling (OLMES project) using CFD, C-7
GASPAC compressor seals, S-1 to S-9
gate valve, A-43
GateCycle software to monitor biomass fuel use at paper mill, P-211 to P-212
gauge blanks, Sp-11
gauge pressure, M-15
gauge type instruments, pressure switches, C-383
gauges, Sp-11
gearboxes (*See also* power transmission), G-1
gears (*See also* helical gears; power transmission; worm gears), G-1, P-74 to P-98
  adjustment of, P-92
  in agricultural production, A-48
  American Gear Manufacturers Association (AGMA) and
  automatic turning, synchronous clutches and, P-107 to P-113
  cross sectional views of, P-78 to P-79
  design requirements/specifications for, P-75 to P-76
  distress in, P-74 to P-75
  durability of, P-75 to P-76
  in gas turbines, C-343, C-345
  in generators and turbogenerators, G-16
  hardness (Sc) values of, P-78
  helical, P-74 to P-82
  high speed, P-98 to P-107
  hydrodynamic power transmission and, P-131
  installation of, P-82, P-97 to P-98
  in isotherm turbocompressors, C-200
  lubrication of, P-81 to P-82, P-96 to P-97
  maintenance of, P-82
  number of teeth in, P-92 to P-93
  overload in, P-74 to P-75
  pitting in, P-74 to P-75
  reduction type, P-77, P-79 to P-81
  scoring in, P-74 to P-75
  scuffing in, P-75
  self-locking, P-88
  sizing of, P-75 to P-76
  in steam turbines, T-79
  synchronous clutches and, P-107, P-113, P-118
  tooth breakage in, P-74 to P-75
  tooth contact in, P-90 to P-92
  tooth form in, P-88 to P-90
  vibrational analysis (VA) and, C-294 to C-296, P-82
  worm, P-82 to P-98
General Electric, M-31
General Electric LM-2500 gas generator, M-33
generators and turbogenerators, G-1 to G-20
  accessories for, G-3 to G-4
  arrangement forms for, G-3
  axial fan in, G-11
  balancing of, G-11
  base frame, G-20
  bearing end shields in, G-6 to G-7
  bearings in, G-7 to G-9
  brush gear in, G-16
  brushless excitation in, G-12 to G-16
  carbon-graphite mix products for brushes in, C-13
  configuration of, G-1
  constant winding temperature operations in, G-20
  control and protection in, G-18 to G-20
  cooling in, G-1 to G-3, G-11 to G-12, G-16
  current transformers in, G-19
  excitation system for, G-3
  exciter housing in, G-15
  fan and fan covers for, G-6, G-8
  heating elements for, G-19
  inspection and testing of, G-17 to G-18
  insulation in, G-6, G-10 to G-11
  low ambient temperature operation in, G-20
  main exciter in, G-12 to G-13
  noise reduction in, G-20
  operating characteristics of, G-20
  overspeed running of, G-11
  pilot exciter in, G-15
  protective equipment for, G-19 to G-20
  rotating rectifier in, G-13
  rotor cooling in, G-11 to G-12
  rotor in, G-3, G-9 to G-10, G-16 to G-17
  rotor retaining and support rings, G-11
  rotor terminals in, G-9 to G-10
  rotor winding in, G-10 to G-11
  slip ring housing in, G-16 to G-17
  slip ring shaft in, G-16
  standards, standard design for, G-1
  static excitation in, G-16 to G-17
  static thyristor rectifier as excitation system in, G-3
  stator core in, G-4 to G-5
  stator terminals in, G-6
  stator winding in, G-5
  stators and stator frame for, G-4 to G-5
  support bearing in, G-15
  surface treatment for, G-16
  synchronous clutches and, P-115 to P-121
  synchronous compensator operation in, G-20
  technical data for, G-4
  temperature monitoring and control, G-18 to G-19
  vibration measurement, in bearing, G-19
genetic modification, E-44
genetically modified micro-organism (GMM), E-44
genetically modified organism (GMO), E-44
GeoCenters Inc., D-17
geothermal heat pumps, H-1 to H-6
geothermal plants, centrifugal gas exhausters in, E-56 to E-67
glass bead peening, A-1
glass powder, freeze dry production of, D-18
global warming and GWP, C-13, C-28 to C-34, R-11, R-13 to R-14
globe valves, CE-18 to CE-392
glycerine, V-15
glycol, as coolant, C-413
governors
  dead band in, T-102
  electrohydraulic, T-104 to T-106
  extraction type, T-106
  mechanical, T-101 to T-102
  mechanical-hydraulic, T-102 to T-104
  steam turbines and, T-100 to T-108
grade of abrasives, A-5 to A-7
grain, in materials, M-39
grain boundaries, M-39
grain growth, M-39
grain size of abrasives, A-1, A-6

graphite, carbon-graphite mix products, C-7 to C-13
Gravi Merge Blending Silos, T-8
gravity blending, in tanks, T-4 to T-8
gravity mixers, agitators, A-38
grease lubrication, L-25 to L-29
Great Lakes toxic chemical pollution, P-65 to P-72
Greater Vancouver Regional District (GVRD), R-8
greenhouse gases, C-13 to C-14
  chlorofluorocarbons (CFCs) in, C-413
  IEA Greenhouse Gas Implementing Agreement, C-14
grinding (*See also* abrasives), A-1, A-3 to A-5, M-35
grinding wheels, A-1 to A-7
grit, for shot peening, A-7 to A-9
grit size of abrasives, A-5 to A-6
gross observed volume (GOV), T-25 to T-27
gross standard volume (GSV), T-25 to T-27
ground injection of CO_2, in Sweden, C-14 to C-21
grounding, tanks and, T-31 to T-32
GT 7 gas turbine (Alstom), C-79
guards, machine, Sp-11
gust loading, stacks and, S-35 to S-38

halocarbons, R-13
Halon, ozone depletion and, O-14
hardness of abrasives, A-5 to A-7
hardness (Sc) values, gears, P-78
hardware production pollutants, P-30
harmonic distortion, in power supplies, C-351 to C-352
harmonics
  in electric motors, E-14, E-33 to E-34
  in vibrational analysis (VA), C-261 to C-262, C-266 to C-267, C-282, C-302, C-305, C-310 to C-311, C-314
Hastelloy, M-41
hazard analysis (HAZAN), in explosion hazard assessment, E-68, E-79 to E-80
hazardous operations (HAZOP), explosion hazard assessment, E-75 to E-76
hazardous or toxic materials (*See also* chemicals; explosion hazard assessment; pollution), C-42 to C-62
  abrasives, abrasive materials, A-8
  ASME certification program for, Sp-30
  cement using, C-33
  Chemical Manufacturers Association (CMA) guidelines for, C-61
  CHEMTREC information service, telephone number, C-61
  container for, C-45
  cooling and, C-415, C-418
  cylinders for shipment of, unloading, C-52 to C-54
  decontamination of empty containers and, C-62
  disposal of, C-62
  drums for shipment of, unloading, C-54
  explosion hazard assessment, E-67 to E-88
  fire hazards of, C-59
  first aid procedures for, C-61
  health hazards of, C-59
  incineration plant for, A-97
  Material Safety Data Sheets (MSDS), C-43, C-58
  metallurgy and, M-32
  protective clothing for handling of, C-59 to C-61
  reactivity of, C-43 to C-45
  respirators for safety in handling, C-59 to C-61
  shipping and handling of, C-45 to C-54, C-58 to C-61
  spills and leaks of, cleanup, C-61 to C-62
  storage containers for, C-56 to C-58
  tank car unloading guidelines for, C-46 to C-52
  tank truck unloading guidelines for, C-51 to C-52
  Threshold Limit Value of, C-61
  toluene diisocyanate (TDI), C-42 to C-62
  transport of, T-40 to T-41
  unloading of, C-45 to C-54
  waste management and, W-1 to W-12
hazards (*See* color coding; explosion; specifications lists)
head, P-222, P-224 to P-228
Health and Safety at Work (HSWA) Act (UK), explosion hazard assessment, E-87
health hazards of toxic chemicals, C-59
heat, air purification using, A-102
heat and power (H&P) applications, C-81
heat exchanger control, hydrodynamic brakes, B-17
heat exchangers (*See also* cogeneration; regenerator; vaporizers), D-4, H-1
heat pumps, H-1 to H-6
  dehumidification (mechanical) and, H-4 to H-5
  earth loop in, H-1 to H-3
  heat reclaim options/opportunities shown for, H-5 to H-6
  high humidity conditions and, H-4
  horizontal earth loop in, H-2
  hybrid systems using, H-5
  liquid cooled air conditioning, H-4
  liquid to air systems, H-3
  liquid to liquid systems, H-3
  on demand hot water (ODHW) heating system using, H-4
  open loop (well system), H-1 to H-2
  operating costs comparisons, H-4
  partial hot water (PHW) heating systems using, H-3 to H-4
  pond and lake loop in, H-1
  vertical earth loop in, H-2
heat recovery steam generator (HRSG) (*See also* cogeneration), C-78, P-208
heat recovery systems, nitrogen oxide (NO_x) emission vs., E-37 to E-38
heat sinks, C-419 to C-421
heat transfer
  computation fluid dynamics in, C-1 to C-7
  condensers, C-239, C-242 to C-243
  electric motors and, E-25 to E-28
  expansion joints and, E-61 to E-62
  impedance type electric heaters, H-8
heat treatment (*See* metallurgy)
heaters, electric, H-6 to H-23
  bayonet type, H-13
  coil type stab in type, H-14
  controls for, H-16 to H-20
  cooling for control systems in, H-18
  Dowtherm type, H-22
  Electrofin type, H-8 to H-12
  explosion proof control panels for, H-17 to H-18
  fin tube type, H-9 to H-12
  heat transfer/pressure drop impedance type, H-8
  horizontal in tank type, H-14
  hot pipe tank heaters, H-12 to H-15
  impedance type, H-6 to H-12, V-1 to V-5
  Inconel type, H-10
  line type, H-13
  ohmic type, H-19 to H-23
  process type, H-15 to H-16
  radiant type, H-22 to H-23
  resistance type elements in, H-11, V-1 to V-5
  sheathed element process type, H-15 to H-16
  suction type, H-13
  thyristor controls in, H-16 to H-19
  vapor type, H-11, V-1 to V-5
  vertical cluster in tank type, H-14
heating elements, in generators and turbogenerators, G-19
heating pollutants, P-30
heating systems/heat pumps, H-1 to H-6
heavy fuel oil, F-37 to F-38, F-43 to F-45
heavy metals, A-96 to A-97
helical gears, P-74 to P-82
  arrangements of, P-76 to P-81
  cross sectional views of, P-78 to P-79
  design requirements/specifications for, P-75 to P-76
  distress in, P-74 to P-75
  durability of, P-75 to P-76
  hardness (Sc) values of, P-78
  installation of, P-82
  lubrication of, P-81 to P-82
  maintenance of, P-82
  overload in, P-74 to P-75
  pitting in, P-74 to P-75
  reduction type, P-77, P-79 to P-81
  scoring in, P-74 to P-75
  scuffing in, P-75
  service factor values for, P-77
  sizing of, P-75 to P-76
  tooth breakage in, P-74 to P-75
  vibrational analysis of, P-82
  worm type, reduction, P-84 to P-87
helical tube sensors, C-386
helium, C-95, C-99
Hepralidae moth in air filter system, A-51
Hercules filters, A-81
hertzian stress, P-76
hexane, C-418
Hickman butyl phthalate manometer, M-22
high efficiency (Hercules) filters, A-81 to A-82
high pressure systems, Sp-11
high speed drive systems (*See* power transmission)
high speed gears, P-98 to P-107
  axial tilting pad bearings in, P-104 to P-107
  back-to-back test bed for, P-99 to P-101
  deformation analysis in, P-102, P-104
  design considerations in, P-107
  radial tilting pad bearings in, P-101 to P-104
high velocity oxygen fuel (HVOF) coatings, M-35
high voltage direct current (HVDC), C-353 to C-354
high voltage suppression systems, T-30
high vs. low velocity air filtration systems, A-74
Hoechst Acetyls Singapore, C-41
Hoechst AG, P-5
homogenization, A-38, A-48
Homprera moth in air filter system, A-51 to A-52
horizontal gas separator, S-27 to S-29
horizontal in tank electric heater, H-14
horsepower
  in electric motors, E-23 to E-24
  in fans, F-1, F-5
  in gears, loss of
  pump requirements for, P-225 to P-226
  in rotary pumps, P-269 to P-270
  in steam turbines and, T-81
hot corrosion, M-40
hot isostatic pressing (HIP), M-41
hot-to-cold expansion joints, E-58 to E-59
hot-to-hot expansion joints, E-58 to E-59
hot water, pumps and NPSH, for, P-236
hotwell, condensers, C-238
housings for air filtration systems and, A-53 to A-58, A-70
huff-and-puff design, A-49 to A-50

humidification process, ejectors in, E-2 to E-3
humidity
  vs. air filtration systems., A-51 to A-53, A-57, A-74
  vs. axial compressors., C-156
  gas turbines and, T-58, T-65 to T-67
  heat pump and, H-4
  in isotherm turbocompressors vs., C-191, C-194
hybrid heat pump systems, H-5
hybrid inventory measurement system (HIMS) tank gauges, T-22 to T-24, T-26 to T-27, T-36, T-39
Hydra filters, A-81
hydraulic amplification, A-41
hydraulic filters (*See* filters)
Hydraulic Institute Standards, pumps and, P-231
hydraulic pressure pumps, P-242 to P-243
hydraulics, pump, P-221 to P-241
hydrazine, P-11 to P-31, P-36, P-40 to P-57
hydrocarbons, C-85, O-17 to O-18
  as refrigerants, R-15 to R-18
  toxic byproducts of, C-42 to C-62
hydrochloric acid, A-9, P-11 to P-31, P-40 to P-57
hydrochlorofluorcarbons (HCFCS), R-10 to R-15
hydrocracking, in petroleum industry, R-3
hydrodynamic brakes (*See* brakes, hydrodynamic)
hydrodynamic power transmission, P-121 to P-166
  annulus in, P-123 to P-127
  arrangement of, P-128
  blades in, P-123, P-138
  boiler feed pumps use of, P-152 to P-164
  characteristics of, P-134 to P-136
  converter circuit in, P-138
  cost of operation of, P-137 to P-138
  coupling selection for, P-164 to P-165
  cutaway view of, P-130
  features of, P-128
  fluid flow through, P-128, P-131, P-133
  fluids in, P-123
  gears in, P-131
  impeller in, P-123 to P-127
  load in, P-128
  mechanical elements to influence converter circuit in, P-138
  offshore pumping station use of, P-151 to P-152
  operation of, P-128 to P-134
  petrochemical plant use of, P-138 to P-147
  pipeline pumping station use of, P-147 to P-152
  power absorption in, P-138
  scope for varying characteristics in, P-138
  speed regulation using, P-155 to P-164
  torque converter in, P-123 to P-127, P-134 to P-136, P-166 to P-184
  torque in, P-128, P-133 to P-134
  transmission ratio in, P-128
  Trilok converters for, P-138
  Type RK variable speed coupling in, P-147 to P-149
  Type SVL variable speed coupling in, P-145 to P-147
  Type SVNL variable speed coupling in, P-142 to P-145
  variable speed couplings in, P-138 to P-147, P-155 to P-164
  vibration in, P-128, P-132
hydrodynamic principle, in turbocouplings, C-346 to C-347
hydrofluorocarbons (HFCS), R-10
hydrogen, C-15, C-85, C-95, C-415, F-37
hydrogen bonding, C-72
hydrogen bromide, V-16
hydrogen chloride, A-9, C-86, V-15
hydrogen cyanide, P-11 to P-31, P-40 to P-57, V-15
hydrogen fluoride, P-11 to P-31, P-40 to P-57, V-16
hydrogen sulfide, C-15, C-86, V-16
hydrokinetic drives (*See* hydrodynamic power transmission)
hydroquinone, P-11 to P-31, P-40 to P-57
hydrostatic power transmission, P-121 to P-124
hydrostatic tank gauges (HTG), T-21 to T-22, T-25 to T-26, T-35 to T-36, T-39
hydrotreating, in petroleum industry, R-4 to R-5
hysteresis, A-41

ice cream, C-76
IcePak, C-5, C-7
icing conditions vs. air filtration, A-49 to A-50
ideal vs. real systems in, C-264
idler rotors, in rotary pumps, P-264
IEA Greenhouse Gas Implementing Agreement, C-14
ignition sources for explosion hazard, E-67, E-78 to E-79, E-83
imbalance (*See* balancing)
impact tubes, in pressure measurement, M-15
impedance type electric heaters, H-6 to H-12, V-1 to V-5
impellers
  in centrifugal compressors, C-172, C-174 to C-177, C-186
  in centrifugal pumps and, P-219, P-226 to P-228
  in hydrodynamic power transmission, P-123 to P-127
  in isotherm turbocompressors, C-191, C-193, C-207, C-223
  in pumps, P-219
  vibrational analysis (VA) of, C-325
in-house power production (IPP), P-72 to P-73
incineration, computational fluid dynamics modeling of, C-1 to C-7
incompressible fluid, flow measurement, M-28
Inconel electric heater, H-10
Inconel, M-40 to M-41
induction electric motors, E-10 to E-15
  AC induction motors used for variable frequency control, E-18 to E-35
induction steam turbines T-98 to T-100
industrial ecological park (*See* ecological parks)
industrial ecology (*See* ecological parks)
industrial effluent, P-71 to P-72
industrial platinum resistance thermometer, M-4
industrial trucks, Sp-11
inertia, vibrational analysis (VA), C-314 to C-315
influence coefficient balancing, C-325 to C-327
Information Handling Services (IHS), ASME, Sp-32
inks, environmentally friendly, F-22
inlet ducts and silencers (*See* acoustic enclosures; air filtration; ducting)
inlet guide vane assembly
  in centrifugal compressors, C-181
  in isotherm turbocompressors, C-207 to C-208, C-214 to C-216
inorganic chemicals, C-41 to C-42, C-64
insect control, insecticides, A-49, A-51, A-66, I-2
inspections
  authorized, Sp-4
  computer aided tomography (CAT) scan in, M-37
  eddy current testing in, M-36
  laser techniques in, M-37
  liquid penetrant (LPI) in, M-36 to M-37
  magnetic particle, M-36
  metallurgy and, M-35
  nondestructive test/ (NDT/NDI), M-35 to M-36
  radiographic, M-35
  ultrasonic, M-36
instrumentation (*See also* condition monitoring; control systems; measurement), Sp-1 to Sp-3
  in reciprocating (labyrinthine) compressors, C-103, C-108, C-111
  in sound measurement, A-14, A-16 to A-17
  in synchronous clutches, P-118 to P-121
insulated gate bipolar transistors (IGBT), E-29
insulation, I-1
  brick lining as, B-23
  chlorofluorocarbons (CFC) in, O-14, O-16 to O-26
  electric motors and, E-14 to E-16, E-25 to E-28, E-31 to E-32
  expansion joints and, E-60, E-66
  generators and turbogenerators, G-6, G-10 to G-11
  MICAFOLD, G-6
  MICAREX, G-5
insulation pillows, expansion joints and, E-65 to E-66
integrally geared compressors, C-219 to C-228
  applications for, C-220 to C-222, C-225 to C-228
  compact (CVK) design, C-227 to C-228
  HVK series, C-220
  internal view of, C-229, C-233, C-235, C-237
  multistage design in, C-219
  operating envelope for, C-230, C-234, C-236 to C-237
  performance data for, C-220 to C-228
  process radial gas turbines (PRT series), C-222 to C-225
  SEE series, C-220
  SEZ series, C-220
  single stage design (direct driven centrifugal), C-220 to C-222
  testing of, C-226, C-234
  VK series, C-220
integrated control system (Series 9500) for gas turbine generators, C-359 to C-361, C-374 to C-377
integrated gasification combined cycle (IGCC), CO_2 and, C-14 to C-16
intensity, of sound, A-11
intercoolers, C-340
  in isotherm turbocompressors, C-193 to C-196, C-207 to C-208, C-215 to C-216, C-219, C-222, C-226
internal gear pumps, P-260 to P-264
International Atomic Energy Agency (IAEA), D-2
International Organization for Standardization (ISO), E-44
inventory control
  hybrid inventory measurement system (HIMS) tank gauges in, T-22 to T-24
  tank gauging in, T-15
  tanks and, central inventory management system for, T-36 to T-39
ionization gauges, M-22
irradiation, food product, I-1 to I-5
ISO 3740–3747, A-18, A-22
ISO 9000 registration program, Sp-29
ISO VILJA technology, E-46, E-48
ISO/DP 9614, A-18
isobutylene, V-16
isodrin, V-15
isoprene rubber, V-15
isopropyl alcohol, P-11 to P-31, P-40 to P-57

isopropylidenediphenol, P-11 to P-31, P-40 to P-57
isotherm turbocompressors (*See also* compressors; turbocompressors), C-136, C-145 to C-146, C-190 to C-219
  applications for, C-190, C-192
  ARF design and performance, C-209 to C-219
  ARI design and performance, C-221
  axial radial concept in, C-209 to C-211
  axial thrust bearings in, C-200 to C-202
  axial thrust transfer in, C-195, C-199 to C-200
  bearings in, C-211, C-214
  blade carrier in, C-211 to C-215
  casing for, C-205 to C-206, C-210 to C-211
  corrosion resistance in, C-191 to C-193
  diffuser in, cross section of, C-213
  discharge temperature in, C-204
  erection and dismantling of, C-200
  features of, C-190 to C-208
  gears in, C-200
  horizontally split casing in, C-210 to C-211
  humidity vs., C-191, C-194
  impellers in, C-191, C-193, C-207, C-223
  inlet guide vanes in, C-207 to C-208, C-214 to C-216
  intercooler in, C-193 to C-196, C-207 to C-208, C-215 to C-216, C-219, C-222, C-226
  journal bearings in, C-200 to C-202
  maintenance of, C-200
  materials used in, C-210
  noise level of, C-194
  operating conditions vs. performance in, C-203
  operating range of, C-190, C-192
  performance data for, C-202 to C-208
  power consumption in, C-190 to C-191
  power oil supply in, C-217 to C-218, C-225
  reliability of, C-201
  RIK design and performance, C-208, C-211 to C-212, C-217
  RIO design and performance, C-208 to C-209, C-217 to C-219
  rotor in, C-193 to C-194, C-197, C-207, C-214, C-218 to C-219, C-225
  seals in, C-206 to C-207, C-217, C-224
  selection charts for, C-205 to C-206
  shaft string drive in, C-194 to C-200
  skid mount for, C-205
  solid couplings in, C-195 to C-201
  space requirements of, C-200 to C-201
  stator blade adjustment in, C-215 to C-217, C-223 to C-224
  stator blades in, C-211 to C-215
  suction side bearing seal in, C-217
  technical data and dimensions chart, C-213

J ring seals, S-5 to S-9
J.M. Voith GmbH, A-39 to A-48, C-392 to C-401, F-29 to F-37, P-121 to P-184
jet A fuel, F-37
jet vacuum pumps, P-321 to P-325
Johns Hopkins Applied Physics Lab, P-4
journal bearings, B-2 to B-6
  in axial compressors and, C-163
  in centrifugal compressors, C-178 to C-181
  in isotherm turbocompressors, C-200 to C-202
  in steam turbines, T-78
  in synchronous clutches, P-116 to P-117

Kawasaki, C-81
Kaybob, C-81
Kemet Electronics Corp., P-6
kerosene, F-37
ketones, P-11 to P-31, P-40 to P-57
keys, Sp-11
kilovolt-ampere (KVA) rating, electric motor, E-8 to E-9
kinetic barriers, M-39
kinetic head, P-222
knife gate valves, C-399, C-402 to C-403
knock out drum (*See also* separators), S-10
Kodak Corporation and environmental accountability case, E-40 to E-42
kraft paper, eco white, E-52
Kronecker delta, C-327

labeling, ecolabeling, E-54 to E-55
labyrinth design pistons, C-112 to C-115, C-121
labyrinth seals, windage heating prediction using CFD, C-6 to C-7
labyrinthine (laby) compressors (*See* reciprocating or labyrinthine compressors)
lactic acid, C-412
lactose, C-68
landfill, R-8
laser cleaning, M-34
laser cutting (*See* metallurgy)
laser drilling, M-35
laser inspection, M-37
laser isokinetic sampling probe (LISP), liquid natural gas (LNG) and, L-23 to L-24
laser machining, M-35
laser repair, M-31
laser welding, M-37
lathes, C-1, M-35
laurolactam, C-68
LCA (*See* life cycle assessment)
leachate, R-8
leaching processes, agitators in, A-38
lead, P-7, P-11 to P-31, P-37, P-40 to P-57
lead angle, worm gears and, P-84
leakage and explosion hazard, fuel leaks and, E-69 to E-70, E-76, E-82 to E-85
leaks, storage tanks, T-16 to T-17
leather and allied products industries, pollutants, P-17
level sensing (*See* liquid level/liquid flow control)
levulinic acid, C-68
life cycle assessment (LCA), C-253, E-43 to E-44, E-55 to E-56, L-1 to L-21, M-31
  algorithms for, L-3
  codes and specifications in, L-19
  cyclic usage factor in, L-3
  design assessment for maximum TBOs and, L-8 to L-10
  fuel quality and machinery selection example of, L-8 to L-10
  gas turbine deterioration and diagnosis example of, L-10 to L-11
  importance of, L-1 to L-2
  life cycle defined in, L-2
  life usage and, L-2 to L-7
  measurement in, L-12 to L-15
  modifications to operating parameters based on, L-7 to L-8
  operational optimization audits in, L-19 to L-21
  performance analysis (PA) with gas path parameters in, L-11 to L-17
  risk and weighting factors in, L-10
  service bulletins (SBs) and, L-3 to L-6
  specific applications of case history in, L-6 to L-7
  steam turbine deterioration and diagnosis example of, L-11
  time between overhaul (TBO) adjustment based on, L-7 to L-10
  triple redundancy and, T-42 to T-43
  turbine example of, L-6 to L-7
  vibrational analysis (VA) and, L-18 to L-19
life cycle inventories (LCIs), E-43 to E-44
life usage, in life cycle assessment, L-2 to L-7
lift check valves, C-392
lifting devices, automotive, Sp-4
lightning strikes, tanks and, T-29 to T-30
lime/limestone FGD, air pollution control and, A-90
limestone ($CaCO_3$), C-14, C-28 to C-30
lindane, P-7
line separators, F-29, S-24 to S-27
line-type electric heater, H-13
linoleic acid, C-66
lipids, freeze drying, with TBA, D-13 to D-17
liposoluble platinum (II), freeze drying, D-17
liposomes, freeze drying, with TBA, D-13 to D-17
liquefied natural gas (LNG), F-37 to F-45, L-21 to L-24
  cold utilization of, F-40
  combustibility of, F-39
  computation fluid dynamic (CFD) studies in, L-21, L-23
  emissions of, F-41
  energy and project integration in, F-40 to F-41
  in line testing of, without plant shutdown, L-23 to L-24
  laser isokinetic sampling probe (LISP) for, L-23 to L-24
  process system for feed, liquefaction, and refrigeration of, L-22
  production of, F-38 to F-39
  safety and health issues in production/transport of, F-39 to F-41
  vaporizers, V-17
  vessel sizing, proprietary scheme for, L-23
liquefied petroleum gas (LPG), F-37 to F-38, R-8, V-16
liquid-cooled air conditioning, H-4
liquid eliminator (*See* separators)
liquid-filled sensors, C-378 to C-382
liquid fuels, F-37
liquid-in-glass thermometers, M-1 to M-2
liquid-jet vacuum pumps, P-321 to P-325
liquid-level gauges, M-17 to M-18
liquid-level measurement, C-388
  capacitance method in, C-388
  conductance method in, C-388
  differential pressure method in, C-389
  electromechanical (displacement) method in, C-389
  flow measurement sensing in, C-389 to C-390
  flow switches, C-389
  magnetic float principle in, C-388
  positive displacement flow measurement in, C-389
  pressure head flow measurement in, C-389
  reed switch, C-390
  single pole double throw (SPDT) reed switch, C-390
  single pole single throw (SPST) reed switch, C-390
  tank gauges for, T-24 to T-27
  turbine flow measurement, C-390
  ultrasonic flow measurement in, C-390
  ultrasonic method in, C-388
  variable area flow measurement in, C-390
  vortex flow measurement, C-390
liquid-liquid extraction, R-8
liquid penetrant inspection (LPI), M-36 to M-37
liquid phase methanol synthesis (LPMeOH) in, C-17 to C-18
liquid ring vacuum pumps, P-318 to P-321
liquid-solid extraction, E-87
liquid-to-air heat pump systems, H-3
liquid-to-liquid heat pump systems, H-3
Lissajous patterns, in vibrational analysis (VA), C-270, C-282 to C-286

LM-2500 gas generator, General Electric, M-33
LNG (*See* liquid natural gas)
load
in electric motors, E-5, E-7, E-11 to E-15
in hydrodynamic power transmission, P-128
on stacks, from wind, S-33 to S-38
torque converters and, fluctuation in, P-172 to P-174
in turbocompressors, control of, C-334
load bearing capacity, carbon-graphite mix products, C-9
load-commutated inverters, in centrifugal compressors, C-185
lobe pumps, P-260 to P-264
lobes, A-48
local-mount liquid filled sensors, C-379 to C-380
logging industry (*See also* forest products industry; pulp and paper mills), P-13
logic trees, in explosion hazard assessment, E-79 to E-80
Los Alamos National Laboratory, P-7
low density polyethylene (LDPE) plastics, P-2
low NO_x burners (LNBs), emission reduction in cement production, C-32 to C-33
low pressure loss filtration, in air filtration systems and, A-81 to A-83
low pressure steam turbines and (high pressure insert), T-100
lower explosion limit (LEL), E-70, E-76
lubrication, L-24 to L-29
actuators and, A-42
AD CARB, carbon–graphite mix products, C-8 to C-13
in axial compressor, C-339, C-341, C-343
carbon–graphite mix products for, C-7 to C-13
cold start requirements of, L-25 to L-26
control of, C-336
in electric motors, E-15 to E-16
forced feed type, L-25
in gears, P-81 to P-82, P-96 to P-97
grease, L-25 to L-29
impurities in, L-25 to L-26
in isotherm turbocompressors, C-217 to C-218, C-225
maintenance of oils and grease in, L-28
methods for, L-25
oil analysis in, O-1
oil rings, ring oiling in, L-25
oil selection chart for, L-24 to L-25, L-27
oils in, L-25 to L-26
piping for, L-29
in reciprocating (labyrinthine) compressors, C-94 to C-95, C-98 to C-99
seals, C-336
selection criteria for, L-24 to L-25
specifications for, L-24 to L-25
splash type, L-25
in steam turbines and, T-79, T-109
viscosity of oils in, L-25 to L-26
lyophilization (*See* freeze drying)

MAAG Gear Co., P-98 to P-107
machine guards, Sp-11
machinery industries pollution, P-21
machining, C-1, M-35
macromonomer solution, freeze dry production of, D-18
magnesium, as fuel impurity, L-9
magnesium oxide (MgO), C-25
magnetic bearings, B-5 to B-11
actuators in, B-5 to B-9
advantages and benefits of, B-10 to B-11
analog control systems for, B-9
axis of control for, B-6
centrifugal compressors, C-185 to C-186, C-189 to C-190
control algorithms for, B-5, B-10
control system for, B-5 to B-6, B-8 to B-11
noncontacting technology in, B-5
open loop control system in, B-10
plant controller for, B-8
position sensors in, B-6
power amplifier in, B-6, B-8
power supply for, B-8
radial type, B-6 to B-7
rotor dynamic analysis in, B-10
sensors in, B-5 to B-9
stator and rotor in, B-6 to B-7
system components and operation in, B-6 to B-11
thrust type, B-6, B-8
transfer function in, B-10
user interface in, B-9
magnetic drives, actuators, A-42, A-45
magnetic float principle, in level sensing, C-388
magnetic particle inspection, M-36
magnification factor (D/F/k), vibrational analysis (VA) and damping, C-316
magnitude, in vibrational analysis (VA), C-263
maintenance
gears, P-82
lubricating oils and grease, L-28 to L-29
service life for turbine components vs. material selection, M-41 to M-42
makeup water, in condensers, C-240
maleic anyhydride, P-11 to P-31, P-40 to P-57, V-15
malsynchronization, C-344
man-machine interface (MMI), in control systems, C-356
manganese, P-11 to P-31, P-40 to P-57
manifold flow (fuel), control of, C-403 to C-408
manlifts, Sp-12
manometer differential pressure, in flow measurement, M-24
manometers, C-329, M-17 to M-18, M-22
manufacturers, Sp-2
manufacturing industries, pollutants and, P-27
Mar M247, M-41
margarine, C-76
maritime use of air filtration systems, A-61
mass flow rate, fans, F-4
mass law performance, sound, A-33
mass transfer
agitators for, A-38
computation fluid dynamics in, C-1 to C-7
mass transfer–accelerator, TBA and freeze drying, D-8
material fundamentals, metallurgy, M-38 to M-40
Material Safety Data Sheets (MSDS), C-43, C-58
material characteristics (*See also* materials selection)
alloys in, M-39
aqueous corrosion in, M-40
compounds in, M-39
corrosion in, M-40
creep deformation in, M-39
creep deformation in, M-40 to M-41
crystalline nature of, M-39
deformation in, M-39
diffusion rate of, M-39
dislocations in, M-39
elastic deformation in, M-39
fatigue in, M-40
fractures in, M-39 to M-40
grain boundaries in, M-39
grain growth in, M-39
grains in, M-39
hot corrosion in, M-40
kinetic barriers in, M-39
metastable conditions in, M-39
microstructure of, M-40
orientation of crystals in, M-39
oxidation in, M-40
phases in, M-39
plastic deformation in, M-39
recrystallization in, M-39
selection of (*See* materials selection)
solid solutions in, M-39
materials processing, computation fluid dynamics modeling of, C-1 to C-7
material selection (gas turbines), M-40 to M-42
blades, M-41
combustion cans, M-40 to M-41
compressor rotor, M-40
pumps and, P-241
service life for turbine components vs., M-41 to M-42
steam turbine, M-42
turbine discs, M-40
turbine vanes, M-41
maximum daily amount, in petroleum industry, R-8
McLeod gauge, M-22
MDS Nordion, I-2
mean time between failure (MTBF), C-418
measurement (*See also* condition monitoring; control systems; flow measurement; pressure measurement; temperature measurement), M-1 to M-31, Sp-12
computation fluid dynamics in, C-1 to C-7
flow measurement (*See* flow measurement)
life cycle assessment (LCA) and, L-12 to L-15
pressure measurement (*See* pressure measurement)
tank gauges in, T-13 to T-40
temperature (*See* temperature measurement)
mechanical governors, T-101 to T-102
mechanical power transmission, P-121 to P-124
mechanical rating, worm gears, P-94
mechanical strength, in carbon-graphite mix products, C-9
mechanical thermal switches (MTS), P-202 to P-204
mechanical-hydraulic governors, T-102 to T-104
medical waste incinerators, ASME certification program for, Sp-30
menthol, C-68
mercaptans, R-8 to R-9
Merco centrifuge, C-36 to C-37
mercury, P-11 to P-31, P-37, P-40 to P-57
mercury micromanometer, M-22
metal diaphragm sensors, C-383
metal industries, pollutants and, P-19 to P-20
metallurgical refurbishment (*See also* metallurgy), M-31 to M-43
metallurgical repair (*See also* metallurgy), M-31 to M-43
metallurgy, L-9, M-31 to M-43, Sp-2
activated diffusion healing (ADH) in, M-32
air plasma coatings in, M-35
anticorrosion coatings, C-442
autoclaving in, M-34
brazing, M-32
casting in, M-43
cleaning and stripping in, M-32, M-34
coatings in, M-34 to M-35
computer aided tomography (CAT) scan in, M-37
dabber TIG welding in, M-38
drilling in, M-35
eddy current testing in, M-36
electrical discharge machining (EDM) in, M-35

fluoride ion cleaning in, M-34
forging in, M-43
grinding in, M-35
high velocity oxygen fuel (HVOF) coatings in, M-35
hot isostatic pressing (HIP) in, M-41
laser cleaning in, M-34
laser drilling in, M-35
laser inspection in, M-37
laser machining in, M-35
laser welding in, M-37
lathes in, M-35
liquid penetrant inspection (LPI) in, M-36 to M-37
machining in, M-35
magnetic particle inspection in, M-36
manufacture in, M-42 to M-43
material fundamentals in, M-38 to M-40
material selection (gas turbines) and, M-40 to M-42
milling in, M-35
nondestructive test/inspection (NDT/NDI), M-35 to M-36
partitioned allow component healing (PACH) in, M-32
physical vapor deposition (PVD) in, M-34
plasma transfer arc welding in, M-38
plastic cleaning in, M-34
powder, M-37
radiographic inspection in, M-35
refurbishment in, M-32 to M-35
repair in, M-32 to M-35
robotic welding in, M-38
service life for turbine components vs. material selection and, M-41 to M-42
superalloy welding in, M-38
thermally densified coatings (TDC) in, M-35
ultrasonic inspection, M-36
vapor coatings in, M-35
water jet cleaning in, M-34
water jet machining in, M-35
welding in, M-37 to M-38
metals as pollutants, P-30
metastable conditions, M-39
metering, fluids (*See* flow control; fuel systems)
methane, C-418
explosion hazard assessment and, E-76
flammability of, E-76 to E-77
methanol, C-16 to C-18, C-72, C-418, F-37, P-11 to P-31, P-40 to P-57, V-15
cost vs. energy efficiency for, C-18 to C-20
liquid phase methanol synthesis (LPMeOH) in, C-17 to C-18
methionine, C-68
methoxyethanl, P-11 to P-31, P-40 to P-57
methyl acrylate, P-11 to P-31, P-40 to P-57
methyl bromide, I-2
methyl chloride, V-15 to V-16
methyl ethyl ketone, P-11 to P-31, P-40 to P-57
methyl isobutyl ketone, P-11 to P-31, P-40 to P-57
methyl methacrylate, P-11 to P-31, P-40 to P-57
methyl tert-butyl ether, P-11 to P-31, P-40 to P-57
methylamine, V-16
methylenbis (phenylisocyanate), P-11 to P-31, P-40 to P-57
methylene chloride, V-16
methylenebis (2-choloraniline), P-11 to P-31, P-37, P-40 to P-57
methylmethacrylate (MMA) monomers, C-41
metric system, Sp-2, Sp-12
MICAFOLD insulation, G-6
MICAREX insulation, G-5
micrometers, laser, A-2
microstructure of materials, M-40
military standards/specifications (MIL-Std), cooling, C-414
Milliken & Co., P-4
milling, C-1, M-35
minimum flow through pumps, P-237 to P-240
mining industry pollutants, P-13
misalignment, vibrational analysis (VA) and, C-290 to C-291
mist eliminators (*See* separators)
mist extractor, separators and, S-18 to S-19, S-22
mixers (*See also* agitators; centrifuges), A-38
mixing process
in centrifuges, C-34 to C-38
computation fluid dynamics in, C-1 to C-7
modal balancing, C-327 to C-328
modeling, computational fluid dynamics for, C-5
modulation, pulse wave modulation (PWM), E-32
molds, computational fluid dynamics to model, C-7
molecular vacuum gauges, M-22
Mollier diagrams, steam turbines and, T-82, T-84
molybdenum alloys, M-42
molybdenum trioxide, P-11 to P-31, P-40 to P-57
monitoring (*See also* condition monitoring)
power transmission assemblies, P-202 to P-204
monitoring logic, in turbocompressors, C-336
monoethanol amine (MEA), C-15
monoglycerides, crystallization of, C-68 to C-71
monomers, P-4 to P-5
monthly amount, in petroleum industry, R-9
Montreal Protocol, O-15 to O-16
Moog valves, C-405
Mopico compressor, gas pipeline application, C-182 to C-186
motor flux, E-32
motor vehicle industries, pollutants and, P-29 to P-30
motors (*See* electric motors)
mounting flanges, expansion joints and, E-63 to E-64
MSHA, C-61
Multipulse electrostatic precipitator, A-84 to A-85
multiresistance thermometers for, T-34 to T-35
multistage condensing steam turbines, T-94 to T-96
multistage flash (MSF) desalination process, D-1
multistage pumps (*See* pumps)
multitemperature thermometers, T-34 to T-35
multitube filters, F-28
municipal solid waste (MWC), ASME certification program for, Sp-30

N-butyl alcohol, V-15
nanoparticle drug dispersant, freeze dry production of, D-17
naphtha, C-418, F-37 to F-38, F-42 to F-43
explosion hazard assessment and, E-69 to E-70, E-76
napthalene, C-72, P-11 to P-31, P-40 to P-57
National Fire Protection Association (NFPA), explosion hazard assessment, E-86
National Gas Turbine Establishment (NGTE), A-69
National Oceanic and Atmospheric Administration (NOAA), C-78
National Pollution Release Inventory (NPRI), P-8 to P-72
natural frequency, in vibrational analysis (VA), C-272 to C-273, C-307, C-316
natural gas, C-95, C-418, F-37
biomass gasification and, CO_2 removal, C-18
explosion hazard assessment and, E-76
in reciprocating (labyrinthine) compressors for, low gas temperature, C-121 to C-136
natural gas combined cycle (NGCC), CO_2 and, C-14 to C-16
NBS, M-12
near vs. far fields, in sound measurement, A-20
NEC design specifications, cooling, C-415, C-418
needle bearings, B-1 to B-2
needle globe valves, CE-18 to CE-392
negative spring rate sensor, C-386 to C-387
NEMA
cooling specifications, C-413 to C-414
generators and turbogenerators design, G-1
neoprene, V-15
net expansion factor in, M-25
net flow area, fans, F-3
net positive suction head (NPSH), P-232 to P-237, P-257
neutron bombardment, nonintrusive wear monitoring in, C-330
new refinery, R-9
nickel and nickel alloys, M-40 to M-42, P-11 to P-31, P-37, P-40 to P-57
abrasives and, A-2
nickel resistance thermometer, M-4
nickel sulfate, C-72
Nimonic, M-41
NIOSH, C-61
Nippon Shokubai, C-41
nitric acid, A-9, P-11 to P-31, P-40 to P-57, V-15
nitrilotriacetic acid, P-11 to P-31, P-38, P-40 to P-57
nitrochlorobenzene, C-68
nitrogen, C-95, V-16
nitrogen oxides (NOx), A-9 to A-11, A-83 to A-90, E-36 to E-38, E-50, F-43, F-45, L-9
in acid rain, A-9 to A-11
air pollution control and, A-83 to A-90
cement production and, C-29 to C-31
emission guidelines for, stationary combustion turbines, E-36 to E-38
oxygen analysis and, O-3 to O-4
ozone and, O-5 to O-6, O-11 to O-15
nitroglycerin, P-11 to P-31, P-40 to P-57
nitropropane, P-11 to P-31, P-38, P-40 to P-57
N-methyl pyrrolidone (NMP), as freeze drying solvent, D-13
noise (*See also* acoustic enclosures)
in electric motors, E-28, E-32 to E-34
expansion joints vs., E-66 to E-67
in generators and turbogenerators, G-20
in isotherm turbocompressors, C-194
nonclogging pumps, P-243
noncontact thermal switches (BTS), P-202 to P-204
noncontacting technology, magnetic type bearings, B-5
nondestructive test/inspection (NDT/NDI), M-35 to M-36
nonintrusive wear monitoring, C-330
nonlubricated sealing system, C-110 to C-112
nonmetallic mineral products industries, pollutants and, P-23
non-Newtonian fluids, pumps and, P-272
nonreturn or check valves, C-392
non-self-supporting hydrodynamic brakes, B-14 to B-16

nozzles, M-25 to M-26, M-29, N-1
in centrifuge, C-35 to C-38
in steam turbines, T-88 to T-89
nozzle rings, in steam turbines, T-76 to T-77
nuclear power plants and technology, Sp-13
ASME accreditation packages for, Sp-25 to Sp-28
ASME codes, Sp-34
desalination processes and, D-1 to D-2
numerical factor, in flow measurement, M-24
NUTEK study of CO_2 ground injection, C-14 to C-21
Nyquist plots, in vibrational analysis (VA), C-270, C-280 to C-282

O rings, S-1, S-5
octane, R-9
offshore or coastal environments, Sp-14
air filtration systems and, A-57, A-66 to A-84
stacks and, S-29 to S-38
offshore pumping, hydrodynamic power transmission and, P-151 to P-152
ohmic electric heaters, H-19 to H-23
oil analysis, O-1
oil mist lubrication, E-16
oil rings, ring oiling, L-25
oil sands, E-87, O-1
oil supply system
in steam turbines and, T-79
temperature monitoring of, C-261
oils for lubrication, L-25 to L-26
oleic acid, C-66
OLMES project, C-6 to C-7
on-demand hot water (ODHW) heating pump system, H-4
once-through cooling water, R-9
one day amount, in petroleum industry, R-9
onspeed balancing, rotors, B-1
open loop (well system), heat pump, H-1 to H-2
open loop control system, magnetic bearings, B-10
operating parameters, life cycle assessment (LCA) and modifications to, L-7 to L-8
operational optimization audits in, L-19 to L-21
operator qualifications and certification, Sp-14
optical comparators, A-2
optical pyrometers, M-11
optimization audits, L-19 to L-21
Optipulse air cleaning system, A-85 to A-86, A-89
orbits, in vibrational analysis (VA), C-270, C-282 to C-286
organic acids, C-72
organic chemicals, C-41 to C-42
organic production processes, E-44
Organon International B.V., D-12
orifice, in flow measurement, M-23, M-25, M-29 to M-30
orifice plates, C-330
oscillation, C-261 to C-262
steady-state, C-313 to C-314
OSHA, C-61
outages, power supply, C-350 to C-354
ovens, drying, D-4
overdamping, C-312
overhaul and repair processes, chemical cleaning, C-38
overload/overspeed control, A-47 to A-48, C-336, C-343 to C-350
brake properties at high speed and acceleration, C-347 to C-350
clutch type couplings in, C-347
constant fill couplings, C-347
fault conditions and, C-344
gas turbines, C-336, C-343 to C-350
gearboxes in, C-343, C-345
hydrodynamic principle in turbocouplings and, C-346 to C-347
limits for, C-345
malsynchronisation and, C-344
Safeset coupling in, C-344 to C-350
short circuits and, C-344
slip curves for, C-347 to C-348
in steam turbines and, C-368 to C-374, T-78, T-106 to T-108
torque limiting requirements for, C-344 to C-345
variable speed couplings in, C-347
overspeed (*See* overload/overspeed control)
overtorque (*See* overload/overspeed control)
oxidation, C-9, M-40
oxidative degradation of plastics, P-2
oxygen, V-16
oxygen analysis, O-1 to O-4
oxygen condensers, condensate content, C-240
oxygen demand, R-8
ozonation, R-9
ozone, C-413, O-4 to O-26
ambient air quality objective for, O-9 to O-10
chemical precursors to, O-5 to O-6
chlorofluorcarbons and, C-13 to C-14
coatings and CFCs vs., O-14, O-16 to O-26
corrosion effects of, O-8 to O-9
depletion of, in stratospheric level, O-12 to O-13, O-15 to O-16, R-10
emission control options for, O-10 to O-15
foams and CFCs vs., O-14, O-16 to O-26
ground level, effects of, O-6 to O-9
Halon and, O-14
insulation and CFCs vs., O-14, O-16 to O-26
materials damage caused by, O-9
Montreal Protocol and, O-15 to O-16
nitrogen oxides (NO_x) and, O-5 to O-6, O-11 to O-15
packaging and CFCs vs., O-14, O-16 to O-26
refrigerants and CFCs vs., O-14, O-16 to O-26
regional remedial measures for, examples of, O-12 to O-13
solvents and CFCs vs., O-14, O-16 to O-26
sterilants and CFCs vs., O-14, O-16 to O-26
United National Environment Programme (UNEP) and, O-13
vegetation effected by, O-8 to O-9
Vienna Convention for the Protection of the Ozone Layer and, O-12 to O-13
volatile organic compounds (VOCs), O-5 to O-6, O-10 to O-15

package, for reciprocating (labyrinthine) compressors, C-110
packaging and paper products, F-21 to F-26, P-1
barrier coatings for, F-15, F-21 to F-23
carton materials, F-24
chlorofluorocarbons (CFC) in, O-14, O-16 to O-26
forest product industry and, F-12 to F-24
inks and, environmentally friendly, F-22
recycling in, F-22
resource efficient, E-52 to E-54
sack manufacture and, F-23 to F-24
waste reduction in, E-54
wastewater management in, F-24
packed reactors, T-40
paint, blistering, A-70
pallets, Sp-14
palm oil, C-68
pantograph control mechanism, expansion joints and, E-63
paper (*See* packaging and paper products; pulp and paper)
paracresol, C-68
paracymene, V-16
paradicholorbenzene, C-68
paraffins, C-72
parasitic forces and noise, electric motors and, E-33 to E-34
parazylene, C-68
partial hot water (PHW) heat pump heating systems, H-3 to H-4
particle fusion, air filtration systems and, A-78
particulate emissions, E-36
partitioned allow component healing (PACH), M-32
p-dichlorobenzene, P-35
Peabody TecTank, T-1 to T-8
peak current rating, electric motors and, E-25 to E-26
peening operations, A-1, A-7 to A-9
Peerless USA, F-24 to F-29, L-21 to L-24, P-214 to P-218, S-10 to S-29, T-108 to T-112
pelletizers, A-48, B-28
Peltier Effect, C-418
pentaerythritol, C-68
peracetic acid, P-11 to P-31, P-40 to P-57
perchloroethylene, V-15
performance analysis (PA) (*See also* condition monitoring), C-253, C-299
performance test codes, ASME, Sp-34 to Sp-35
periodic frequency, vibrational analysis (VA), C-263
periodic motion, vibrational analysis (VA), C-11, C-302, C-305, C-310
personal and household service industries, pollutants and, P-31
pesticides, A-49, P-7
Petrochemical Company of Singapore (PCS), C-38 to C-41
petrochemical complex/plant, C-38 to C-41
hydrodynamic power transmission and, P-138 to P-147
Petrochemical Corporation of Singapore (PCS), C-403 to C-408
petroleum and natural gas industries
acid treatment in, R-5
additives and, R-5 to R-6
alkylation in, R-4
atmospheric distillation, R-2
blending processes in, R-2, R-5 to R-6
catalytic cracking in, P-2 to P-3
chemical treating in, R-5
coking processes in, R-4
combining processes in, R-4
conversion process in, R-2 to R-4
cracking processes in, P-2 to P-4
crude oil composition and, R-1 to R-2
deposits and compliance assessment for, R-5 to R-6
gas treating in, R-5
glossary of terms for, P-6 to P-10
hydrocracking in, R-3
hydrotreating in, R-4 to R-5
physical treatments in, R-5
pollutants and, P-13 to P-14
polymerization in, R-4
rearranging processes in, R-4
separation process in, R-2
solvent extraction in, R-5
sweetening processes in, R-5
tanks and, T-16
treatment process in, R-2, R-4 to R-5
vacuum distillation in, R-2
visbreaking in, R-3
Petrotech Inc. (retrofit controls), C-354 to C-408

pharmaceuticals, Sp-2
freeze drying process in, D-4 to D-18
phase, in vibrational analysis (VA), C-270, C-272 to C-282
phase angle, in vibrational analysis (VA), C-265, C-305 to C-306, C-315 to C-318
phases, of materials, M-39
phathalate, P-10 to P-31, P-33, P-40 to P-57
phenol, P-11 to P-31, P-40 to P-57
phenylphenol, P-12 to P-31, P-40 to P-57
Philips Electronics, P-5
Philips Pening gauges, M-22
phosgene, V-15
phosphates, C-78
phosphoric acid, P-12 to P-31, P-40 to P-57
phosphorus, P-12 to P-31, P-40 to P-57, P-64 to P-65
photodegradation of plastics, P-2
photosynthetic action, R-9
phtalic anhydride, P-12 to P-31, P-40 to P-57
physical vapor deposition (PVD), M-34
pinch valves, CE-18 to CE-392
pipe diameter, in flow measurement, M-24
pipelines and pipeline pumping stations
centrifugal compressors used in, C-182 to C-186
hydrodynamic power transmission used in, P-147 to P-152
pollutants and, P-28
piping, Sp-16
centrifugal pump, P-257
lubrication line, L-29
pressure measurement in, M-15 to M-16
in reciprocating (labyrinthine) compressors, C-103
rotary pump, P-273 to P-274
Pirani gauges, M-22
piston rods, in reciprocating (labyrinthine) compressors, C-91 to C-94
piston type pressure switches, C-385 to C-386
pistons
in actuators, A-42
labyrinth design, C-112 to C-115, C-121
in reciprocating (labyrinthine) compressors, C-86 to C-88, C-93 to C-97, C-112 to C-115, C-121, C-126 to C-132
pitch, L-25, R-9
pitot static tubes, C-329, M-15
pitting, gear, P-74 to P-75
plankton, C-78
plant controller, magnetic bearings, B-8
plasma transfer arc welding, M-38
plastic cleaning, M-34
plastic deformation, M-39
plastics, P-1 to P-7, Sp-2
biodegradation of, P-2 to P-3
biodistegrable, P-2
conductive polymers in, P-4 to P-7
electric, P-3 to P-7
ENVAR biodegradable, P-2
environmental degradation of, P-1
equivalent series resistance (ESR) polymers, P-6
low density polyethylene (LDPE), P-2
monomers, P-4 to P-5
oxidative degradation of, P-2
photodegradation of, P-2
pollutants and, P-16 to P-17
polyacetylene, P-5 to P-7
polyaniline, P-5 to P-7
polyethylene, P-2
polypyrrole, P-4
polythiophene, P-3 to P-7
smart membranes using conductive polymers, P-7
thermoset, Sp-18
platinum resistance thermometer, M-4
plug valves, CE-18 to CE-392
plumbing, P-30, Sp-17
plume abatement cooling towers, C-424 to C-426
poisoning of catalyst, in air pollution control, A-96
polar solutes/solvents, C-72
pollution (*See also* air pollution control; environmental accountability; water pollution), P-7 to P-22
accidents/incidents and, E-46
acid rain and, A-9 to A-13, P-60 to P-62
agricultural chemicals and, A-49, P-12
air emissions, P-57 to P-60
beverage industries and, P-15, P-29
biochemical oxygen demand (BOD) in, P-69
business service industries and, P-31
Canadian studies in, P-8 to P-72
carbon dioxide (CO_2), C-13 to C-21
carbon monoxide and, P-62 to P-64
cement kilns and, C-22, C-28 to C-34
chemical and chemical products industries and, P-25 to P-26
chlorofluorocarbons (CFCs) in, C-413
communication industries and, P-29
condensers and, C-233
construction industries and, P-27 to P-28
continuous emissions monitoring (CEM) in, C-34
drug industries and, P-29
ecological parks, E-1
ecosystems, E-1 to E-3
electrical and electronics products industries and, P-22 to P-23
environmental accountability, E-40 to E-49
eutrophication in, P-64 to P-65
fabricated metal industries and, P-20 to P-21
federal government service industries and, P-31
food industries and, P-14 to P-15, P-29
furniture and fixture industries and, P-18
IEA Greenhouse Gas Implementing Agreement, C-14
industrial effluent in, P-71 to P-72
leather and allied products industries, P-17
logging industry and, P-13
machinery industries and, P-21
manufacturing industries (other), P-27
metal industries and, P-19 to P-20
metals, hardware, plumbing, heating, building materials industries, P-30
mining industry and, P-13
motor vehicle industries and, P-29 to P-30
National Pollution Release Inventory (NPRI) for (Canada), P-8 to P-72
nonmetallic mineral products industries and, P-23
ozone, O-4 to O-26
personal and household service industries and, P-31
petroleum and natural gas industries and, P-13 to P-14
phosphates and, C-78
phosphorus in, P-64 to P-65
pipeline transport industries and, P-28
plastics and, P-16 to P-17
printing, publishing and allied industries, P-19
pulp and paper industries and, P-18 to P-19
recovery, reuse, recycling in, P-10, P-54 to P-58
"red tide" algae (*Ptychodiscus brevis*) and, C-78
releases of, P-8 to P-9
rubber products industries and, P-15 to P-16
service industries and, P-28, P-31
service industries of mineral extraction and, P-14
sewage (*See also* wastewater), P-69 to P-72
storage and warehousing industries and, P-29
sulfur dioxide in, P-60 to P-62
surface waters and, P-40 to P-48
textile industries and, P-17 to P-18
tobacco industries and, P-29
toxic chemicals and, in Great Lakes, P-65 to P-72
trade contracting industries and, P-28
transfer of, P-9, P-49 to P-54
transportation equipment industries and, P-22
transportation industries and, P-28
utility industries and, P-29
waste management and, W-1 to W-12
wastewater flows in, P-69 to P-72
water pollution and, P-64 to P-65
wholesale products and industries and, P-30 to P-31
wood industries and, P-18
polyacetylene, P-5 to P-7
polyamide epoxy primer, as tank coatings, T-4 to T-8
polyaniline, P-5 to P-7
polychlorinated biphenyls (PCBs), P-7
polyene, freeze dry production of, D-17
polyethylene plastics, P-2
polymerization, C-43, C-58
computation fluid dynamics in, C-1 to C-7
in petroleum industry, R-4
polypyrrole, P-4
polytetrafluoroethylene (PTFE) seals, S-5 to S-9
polythiophene, P-3 to P-7
pond and lake loop heat pump, H-1
Portland cement (*See* cement)
position sensors, A-46, B-6
positive displacement flow measurement, C-389
positive lubrication system (PLS), electric motors and, E-15 to E-16
potassium, as fuel impurity, L-9
potassium chloride, C-68
potassium nitrate, C-68, C-72
potential head, P-222
powder metallurgy, M-37
power
in gas turbines, atmospheric conditions vs. degradation in, T-65 to T-67
in hydrodynamic brakes, B-16, B-20 to B-21
in hydrodynamic power transmission, P-138
in pumps, requirement for, P-225 to P-226, P-269 to P-270
in steam turbines, T-81
of sound (PWL), A-14 to A-19
power amplifiers, magnetic bearings, B-6, B-8
power control, in turbocompressors, C-333
power factor, electric motor, E-7 to E-9
power generation
ASME certification program for, Sp-30
explosion hazard assessment in, E-67 to E-87
gas turbines and, T-47
oxygen analysis in, O-2 to O-4
power production (IPP/SPP), P-72 to P-73
power rotor, in rotary pumps, P-264
power stations, synchronous clutches and, P-107 to P-113
power supplies
control of, C-350 to C-354
demand charge and, energy costs of, E-9 to E-10
electric motors and, E-3 to E-5
flexible AC transmission system (FACTS) for, C-353 to C-354
harmonic distortion in, C-351 to C-352
high voltage direct current (HVDC) in, C-353 to C-354
kilovolt-ampere (KVA) rating in, E-8 to E-9

magnetic bearings, B-8
outages and disturbances in, C-350 to C-354
peak current rating in, E-25 to E-26
power factor in, E-7 to E-9
PQ2000 system for, C-351 to C-354
reactive kilovolt-ampere (KVAR) rating in, E-8 to E-9
real power, in calculating energy consumption costs, E-7 to E-8
RMS current rating in, E-25 to E-26, E-28
ultrasonic cleaning, U-1 to U-2
voltage sags in, C-350 to C-354
voltage unbalance and, electric motor operating costs, E-6 to E-7
power transistors, in electric motors, E-29 to E-32
power transmission (*See also* gears; hydrodynamic power transmission; synchronous clutch), P-73 to P-204
gears in, P-74 to P-98
hydrodynamic, P-121 to P-166
hydrostatic, P-121 to P-124
mechanical, P-121 to P-124
monitoring devices in, P-202 to P-204
Safeset couplings in, P-184 to P-191
synchronous clutch couplings and, P-107 to P-115
torque converters and, P-166 to P-184
torque limiting safety coupling in, P-184 to P-191
turbocouplings (type T) in, P-191 to P-196
types of, P-121 to P-127
powered platforms, Sp-17
PQ2000 power supply system, C-351 to C-354
Pratt and Whitney, M-31
precision platinum resistance thermometer, M-4
predictive strategies, in condition monitoring, C-254
pressure angle, worm gears, P-84
pressure control, C-333 to C-334
in steam turbines, C-368 to C-374
in turbocompressors, C-333
pressure differential transmitter (PDT), C-333 to C-334
pressure gauge, C-107
pressure head, P-221 to P-224
pressure head flow measurement, C-389
pressure measurement (*See also* pressure switches), C-328 to C-339, M-14 to M-31
absolute pressure in, M-14 to M-15
alphatron gauges in, M-22
ambient air pressure (P_0), T-55 to T-58
ambient air temperature (T_0) in, T-58
barometers in, M-18
bellows gauges in, M-21
Bourdon tube in, M-21
calibration of, M-16
connections in, M-15 to M-17
consolidated diaphragm comparator in, M-22
deadweight tester and gauges in, M-18 to M-20
diaphragm gauges in, M-21
differential pressure in, M-14, M-18 to M-22
elastic gauges in, M-20 to M-22
electric transducers for, M-22 to M-24
errors in, M-15 to M-16
flow errors in, M-15
flow stream pressure in, M-15
gauge pressure in, M-15
Hickman butyl phthalate manometer, M-22
impact tubes and, M-15
ionization gauges in, M-22
liquid level gauges in, M-17 to M-18
manometers in, M-17 to M-18, M-22
McLeod gauge for, M-22
mercury micromanometer in, M-22
molecular vacuum gauges in, M-22
Philips Pening gauges in, M-22
piping and, M-15 to M-16
Pirani gauges in, M-22
pitot tubes in, M-15
pressure defined for, M-14
pressure relationships defined for, M-14
static pressure in, M-15
static taps and, M-15
static tubes and, M-15
thermal conductivity gauges in, M-22
thermocouple gauges in, M-22
total pressure in, M-15
turbulence errors in, M-15
vacuum pressure in, M-15
velocity head in, M-15
velocity pressure in, M-15
pressure regulation valves (PRV), F-25 to F-26
pressure relationships, M-14
pressure switches (*See also* pressure measurement), C-383 to C-388
absolute pressure, C-383
accuracy and lifespan of, C-387
applications and cost of, C-387 to C-388
Belleville washer negative spring rate sensor in, C-386 to C-387
bellows type sensors in, C-386
Bourdon tube sensors in, C-383 to C-385
differential pressure-actuated, C-383
gauge type instruments in, C-383
helical tube sensors in, C-386
metal diaphragm sensors in, C-383
negative spring rate sensor in, C-386 to C-387
piston type, C-385 to C-386
proof pressure in, C-383
unsealed piston sensors in, C-387
pressure transducers, C-329
pressure transmitter, C-329, C-333 to C-334
pressure vessels, Sp-17
ASME accreditation packages for, Sp-25 to Sp-28
ASME CD-ROM for, Sp-32
ASME codes, Sp-35
books of interest on, Sp-37 to Sp-38
data report forms for, Sp-36 to Sp-37
international code for, Sp-24
ISO 9000 registration program, Sp-29
pressure-intensity (P-I) index of sound, A-16, A-18 to A-19
pressurized storage tanks, vaporizers and, V-22 to V-25
preventive strategies, in condition monitoring, C-254 to C-261
priming pumps, P-254
printing, publishing and allied industries, pollutants and, P-19
priority pollutants, R-9
probes
sound intensity, A-16 to A-17
ultrasonic cleaners and, U-2 to U-3
vibrational analysis (VA), C-266 to C-270
process control, in turbocompressors, C-333 to C-334
process hazard review (PHR), in explosion hazard assessment, E-67, E-74 to E-76
process industry, pumps and for, P-285 to P-309
process radial gas turbines (PRT series), C-222 to C-225
process type electric heaters, H-15 to E-16
programmable logic controllers (PLCs), N-1
condition monitoring (CM) and, C-259
propagation of sound, A-27
propane, C-418, E-76, F-41, R-11, V-15 to V-17
propanol, V-15
proportional conversion of electrical control signals, A-40
propylene, P-12 to P-31, P-40 to P-57, V-16
propylene glycol (PG), C-408 to C-412
propylene oxide, C-415, P-12 to P-31, P-38, P-40 to P-57, V-15 to V-16
prostaglandin E1 (PGE-1), freeze dry production of, D-17
protective clothing
freeze drying and TBA handling, D-5 to D-7
toxic chemicals handling, C-59 to C-61
proximity probes, vibrational analysis (VA), C-266 to C-270
PTFE, expansion joints using, E-60
pulp and paper industries (*See also* forest products), P-204 to P-214, Sp-2
back-pressure turbine in, P-210
biomass fuel from, P-205 to P-214
biomass gasifier (BIG) in, P-208 to P-210
black liquor gasifier (BLG) in, P-209
black liquor production in, P-205 to P-214
combined heat and power (CHP) in, P-208, P-210
duct burner in, biomass fuel, P-210
energy consumption by, P-204 to P-205
gas turbine in, using biomass fuels, P-210 to P-211
GateCycle software to monitor biomass fuel use at, P-211 to P-212
heat recovery steam generator (HRSG) in, P-208, P-210
pollutants and, P-18 to P-19
Tomlinson boiler in, P-210
white liquor production in, P-205 to P-214
pulsating vibration, C-264
pulsation dampeners, P-214 to P-218
pulse jet–type air filters, A-49
pulse wave modulation (PWM), electric motors and, E-32
pumping stations, hydrodynamic power transmission and, P-151 to P-152
pumps, P-214 to P-325, Sp-17
abrasive fluids and, P-274
agricultural and food products, A-48
API 610 double suction type, P-296 to P-299
API 610 rigid or flexible coupled, P-296
applications for, P-241 to P-255, P-281 to P-285
axial balance in, P-243 to P-247, P-275 to P-276
axial piston, P-260 to P-264
Bernoulli's theorem and, P-222
boiler feed, P-242
Bourdon tube to measure flow in, P-224
CAP 8 type (API 610 process pumps), P-285 to P-292
casings for, P-276
cavitation in, P-232
centrifugal (*See* centrifugal pumps)
characteristic curves of performance for, P-228
condensate, NPSH and, P-236 to P-237
condensate, P-243
continuity equation in, P-221 to P-224
CVA8 type (API 610) overhung vertical inline integral bearing frame flexibly coupled, P-292 to P-296
CVAR type rigid coupled pumps, P-296
displacement of, P-260, P-267 to P-268
double suction, P-296 to P-299
double volute, P-247 to P-248
electric motors and in, E-22
entrained air/gases and, P-271 to P-272, P-278 to P-281
external gear, P-260 to P-264
friction loss in, P-222
head in, P-222, P-224 to P-228
heavy duty double case multistage, P-299 to P-309
hot water, NPSH and, P-236
Hydraulic Institute Standards for, P-231

hydraulic pressure, P-242 to P-243
hydraulics of, P-221 to P-241
hydrodynamic brakes, controller in, B-17 to B-19
hydrodynamic power transmission and, P-152 to P-164
impellers in, P-219, P-226 to P-228
inlet pressure in, P-276
installation of, P-257 to P-258, P-277 to P-278
internal gear, P-260 to P-264
kinetic head in, P-222
lobe, P-260 to P-264
materials selection, by liquids to be pumped, P-241
minimum flow through, P-237 to P-240
non-Newtonian fluids and, P-272
nonclogging, P-243
operation of, P-257 to P-258, P-277 to P-278
parallel operations for, P-228 to P-230
performance analysis of, P-226 to P-228, P-265 to P-281
piping for, P-257, P-273 to P-274
potential head in, P-222
power/horsepower requirement calculation for, P-225 to P-226, P-269 to P-270
pressure head in, P-221 to P-224
priming of, P-254
process industry uses of, P-285 to P-309
recirculation connection for, P-240 to P-241
rotary (*See* rotary pumps), P-259 to P-281
rotors in, P-276 to P-277
screw, P-260 to P-264
seals in, P-250 to P-254, P-276
series operations for, P-228 to P-230
single stage, double suction type, P-241 to P-242
sliding vane, P-260 to P-264
specific speed of, P-219 to P-221
static head in, P-221 to P-224
stuffing boxes for, P-248 to P-254
suction conditions, suction lift in, P-272 to P-273
suction head in, net positive (NPSH), P-232 to P-237, P-257
suction head in, P-222
suction lift in, P-222
temperature rise in, P-237 to P-240
temperature vs. efficiency in, P-254 to P-255
theoretical capacity (Q_t) of
theory of operation for, P-218 to P-281
timed screw, P-260 to P-264
total discharge head in, P-223 to P-224
total head in, P-222
troubleshooting, P-258 to P-259
types of, by application, P-281 to P-285
untimed screw, P-260 to P-264
vacuum type (*See* vacuum pumps), P-309 to P-325
valves in, P-257
velocity head in, P-221 to P-224
vibration limits for, P-255 to P-258
viscous fluids and, P-230 to P-232, P-265 to P-267, P-270 to P-273
warmup procedures for, P-254
purification
of air, A-101 to A-102
centrifugal, C-35 to C-38
pyridine, V-16
pyrometers, M-11
pyrometry, M-10
pyruvic acid, C-412

quality control, in reciprocating (labyrinthine) compressors, C-116 to C-121
quill shafts, synchronous clutches and, P-118

RA search online service, ASME, Sp-33
radar tank gauges, T-19 to T-21, T-33 to T-34
radial magnetic bearing, B-6 to B-7
radial tilting pad bearings, in high speed gears, P-101 to P-104
radiant electric heaters, H-22 to H-23
radiation error, in temperature measurement, M-13
radiation thermometers, M-10 to M-11
radiographic inspection, M-35
rainfall, vs. air filtration systems, A-50 to A-63
random vibration, in vibrational analysis (VA), C-263
Rankine cycle, T-82 to T-84
rate of flow in, M-25
reaction, computation fluid dynamics in, C-1 to C-7
reactive kilovolt-ampere (KVAR) rating, E-8 to E-9
reactive sound fields, A-20
reactive strategies vs. condition monitoring, C-254
reactivity index of sound, A-12, A-16, A-18 to A-19
reactors, R-1
computational fluid dynamics modeling for, C-3 to C-4
packed type, T-40
tray type, T-40
reaffirmation (R) design, Sp-3
rearranging processes, in petroleum industry, R-4
reciprocating (labyrinthine) compressors, C-81, C-84 to C-136
acoustic calculations for, C-116
applications for, C-85, C-95 to C-110
arrangement and drive in, C-103 to C-108
balanced opposed type, C-88
bearings in, C-86, C-92
benefits of, C-88 to C-95
bone dry natural gas and, C-126
coding of, C-104
condensate receiving tank for, C-109
cooling and piping in, C-103
cooling valve in, C-96
crankcase deformation at cryogenic temperatures, C-132
crankcase of, C-86, C-92, C-107
crankshaft in, C-99, C-108
crankshaft seal in, C-114 to C-115, C-123 to C-124
crosshead fastening in, C-86, C-93
cryogenic temperatures and, C-121 to C-126
cutaway view of, showing features, C-91
cylinders in, C-103, C-126 to C-132
D type, C-117, C-119
design and maintenance features of, C-86 to C-88
dimensions of, C-89, C-104, C-119 to C-120
double acting piston in, C-130
dynamic crank throw behavior in, C-116, C-126
E type, C-117, C-119
electrical system of, C-110
features of, design/special, C-99 to C-108
filter system for, C-109
gas stream through, C-108
gases compressed by, C-85 to C-86, C-95
instrumentation in, C-103, C-108, C-111
K type, C-118, C-120, C-129
labyrinth piston design in, C-112 to C-115, C-121
leaks in, at cryogenic temperatures, C-132 to C-135
liquefied natural gas service (low gas temp application), C-121 to C-136
lubrication in, C-99
maintenance of, C-126, C-128, C-135
materials used in, C-104, C-116, C-126, C-129 to C-131
nonlubricated sealing system in, C-110 to C-112
oil wiper packing in, C-94 to C-95, C-98
operating data for, C-105 to C-106
optional features for, C-110
package for, C-110
performance parameters of, C-119 to C-120
performance range of, typical, C-89
piston rod gland in, C-114, C-123
piston rod guide bearing in, C-114, C-123
piston rod packing in, C-91 to C-94, C-97
piston rod runout in, C-86 to C-88, C-93
pistons of, C-88, C-94, C-126 to C-132
pressure gauge in, C-107
process conditions for, C-85 to C-86
quality control for, C-116 to C-121
remote controls for, C-109
research and development in, labyrinth design, C-115 to C-121
safety valve in, C-107
sealing labyrinth design in, C-116, C-125
seals in, C-110 to C-112, C-134
selection charts for, C-90, C-104
skid mount for, C-108, C-128
startup procedures for, C-135 to C-136
supply scope of, standard, C-108 to C-110
temperature control in cryogenic applications, C-132
testing of, C-110
tools used/supplied with, C-91
valves in, C-88 to C-91, C-95, C-107, C-115, C-122, C-124, C-135
vertical, inline type, C-87
water system for, C-109
reciprocating engines, E-38 to E-39
recirculation connection, in pumps, P-240 to P-241
recorders/recording systems, in vibrational analysis (VA), C-270 to C-286
recovery, reuse, recycling, P-10, P-54 to P-58
recrystallization, M-39
rectifiers
generators and turbogenerators, G-13
static thyristor, G-3
recycling, P-10, P-54 to P-58
packaging and paper products, F-22
plastic, P-1
resource efficient packaging and, E-52 to E-54
waste reduction, E-54
"red tide" algae (*Ptychodiscus brevis*), C-78
reducers, worm gears, P-84 to P-87
reduction gears, helical, P-77, P-79 to P-81
reduction gears, worm, P-84 to P-87
redundancy, triple, T-41 to P-43
reed switch, C-390 to E-18
reference crude rate (RCR), R-9
refineries, petroleum (*See also* petroleum industries), R-1 to R-10
refinery gas, F-37
refrigerants (*See also* cooling), R-10 to R-19, V-16
air conditioning and, R-11 to R-13
alternative types, R-10 to R-14
American Society of Heating, Refrigeration, Air Conditioning Engineers (ASHRAE) classification, R-11
ammonia as, R-13 to R-18
azeotrope mixtures for, R-11
balance between ozone depleting fluids and other emissions, R-13
carbon dioxide as, R-13 to R-18
chloroluorcarbons (CFCs) in, O-14, O-16 to O-26, R-10 to R-15
coefficient of performance (COP) in, R-18
comfort conditioning and, R-11 to R-13
design constraints and, R-18 to R-19

development of refrigeration systems and, R-14 to R-15
global warming potential (GWP) of, R-11, R-13 to R-14
halocarbons in, R-13
hydrocarbons as, R-15 to R-18
hydrochlorofluorcarbons (HCFCs) in, R-10
hydrofluorocarbons (HFCs) in, R-10
mixtures for, R-10 to R-11
natural types, R-14 to R-18
ozone depletion by, R-10
propane and, R-11
R22 and, R-17 to R-18
R22 in comfort conditioning, R-11 to R-13
thermoelectric cooling, C-418 to C-423
threshold limit value (TLV) of, R-19
time weighted average (TWA) exposure to, R-19
total equivalent warming impact (TEWI) of, R-12 to R-14
water as, R-15 to R-18
zeotrope mixtures for, R-11
regenerative cycle, in steam turbines, T-85
regenerators, R-19, T-40
regulators, electrohydraulic actuator in, A-44
regulatory affairs, in environmental accountability, E-44
reheat cycle, in steam turbines, T-84 to T-85
reindeer, environmental protection, F-16
reinforced thermoset plastics, Sp-18
relative humidity, gas turbines and, T-58, T-65 to T-67
release of pollutants, P-8 to P-9
Reliance Electric (electric motors), E-3 to E-36
relief valves, ASME accreditation packages for, Sp-25 to Sp-28
remote controls, C-109
Rene 80, M-41
residual pitch, R-9
resin, in lubricating oil, L-25
resistance thermometers in, M-2 to M-5
resistance type electric heating elements, H-11, V-1 to V-5
resistive thermal detectors (RTDs), C-261
resonance, A-27 to A-28, C-272 to C-273
resonant couplings, C-317
respirators for safety in handling toxic chemicals, C-59 to C-61
response time, in temperature switches, C-380
retrofit control systems (*See also* control systems), C-354 to C-408
aeroderivative fuel regulator for gas turbine, C-354 to C-358
antisurge control in, C-359 to C-368
fault tolerance in, C-355
integrated control system (Series 9500) for gas turbine generators, C-359 to C-361, C-374 to C-377
overload/overspeed protection for steam turbines, C-368 to C-374
pressure control for steam turbines, C-368 to C-374
retrofitting, condition monitoring (CM) and, C-255
reverse osmosis (RO) desalination process, D-1
Reynolds number in, M-25
risers, solids concentration with baffle plate, using computational fluid dynamics, C-2
risk management/assessment
as-low-as-reasonably-practicable (ALARP) risks and, E-80
condition monitoring (CM) and, C-254
environmental accountability and, E-44
explosion hazard assessment and, E-71
life cycle assessment (LCA) and, L-10
RMS current rating, electric motors, E-25 to E-26, E-28
robotics, welding using, M-38
roller bearings, B-1 to B-2
rolling contact type bearings, B-1 to B-2
Rolls Royce, M-31 to M-34
rotary piston blowers, C-232
rotary pumps, P-259 to P-281
abrasive fluids and, P-274
applications for, P-270
axial balance in, P-275 to P-276
axial piston, P-260 to P-264
capacity of, P-267 to P-268
casings for, P-276
classification of, P-260 to P-265
design details of, P-274 to P-277
displacement of, P-267 to P-268
displacement pumps and, P-260
entrained air/gases and, P-271 to P-272, P-278 to P-281
external gear, P-260 to P-264
idler rotors in, P-264
inlet conditions vs. performance in, P-265 to P-267
inlet pressure in, P-276
installation of, P-277 to P-278
internal gear, P-260 to P-264
journal bearing theory in, P-264 to P-265
lobe, P-260 to P-264
non Newtonian fluids and, P-272
operation of, P-277 to P-278
performance analysis of, P-265 to P-281
pipe size for, P-273 to P-274
power rotor in, P-264
power/horsepower requirements of, P-269 to P-270
pressure in, P-268 to P-269
rotors in, P-264, P-276 to P-277
screw, P-260 to P-264
sealing rotor in, P-264
seals in, P-276
selection criteria for, P-270
sliding vane, P-260 to P-264
speed of, P-265, P-273
suction conditions, suction lift in, P-272 to P-273
theoretical capacity (Q_t) of, P-265
timed screw, P-260 to P-264
untimed screw, P-260 to P-264
viscous fluids and, P-265 to P-267, P-270 to P-273
rotating machinery, Sp-3
rotating positive displacement compressors, C-228 to C-232
applications for, C-230 to C-232
rotors in, C-229
rotating stall, C-328
rotational velocity, vibrational analysis (VA), C-265
rotors
in axial compressors, C-160
balancing, onspeed, B-1
balancing of (*See also* vibrational analysis), C-288 to C-291, C-318
damping in (*See also* vibrational analysis), C-318
in centrifugal compressors, C-189 to C-190
in electric motors, E-20
in generators and turbogenerators, G-3, G-9 to G-10, G-16 to G-17
in isotherm turbocompressors, C-193 to C-194, C-197, C-207, C-214, C-218 to C-219, C-225
influence coefficient balancing and, C-325 to C-327
magnetic bearings and, B-5 to B-7
material selection for, M-40
misalignment and, C-290 to C-291
modal balancing in, C-327 to C-328
in rotating positive displacement compressors, C-229
in rotary pumps, P-264, P-276 to P-277
in steam turbines, T-76
subsynchronrous frequency vibration in, C-291 to C-294
vibrational analysis (VA) of, C-288 to C-291, C-301 to C-325
RPM sensors, A-46
rubber products industries, pollutants and, P-15 to P-16
rubbing, vibrational analysis (VA) and, C-287

sack (paper) manufacture, F-23 to F-24
sacrificial anodes vs., C-247
Safeset couplings in, P-184 to P-191, C-344 to C-350
safety, Sp-3
accident management, A-9
ASME codes, Sp-35
tanks and, T-28 to T-32
safety systems, in turbocompressors, C-332 to C-333
safety valve, C-107, Sp-25 to Sp-28
salt corrosion, in air filtration systems, A-57 to A-64, A-69, A-72 to A-74
sawmills and wood processing operations, F-16, F-19
Saybolt, P-272
SCADA, control systems and, C-355
scanning electron microscopic (SEM), freeze drying of TBA solutions, D-8, D-15 to D-16
scavenging, in gasoline powered engines, E-39
Schott Glaswerke, D-18
scoring, gear, P-74 to P-75
screw compressors (*See* rotating positive displacement compressors)
screw pumps, P-260 to P-264
screw threads, Sp-18
scrubbers, E-40 to E-42, T-40
dry (*See* dry scrubbers)
fuel gas conditioning systems, F-25, F-28 to F-29
scuffing, gear, P-75
sealing rotor, rotary pumps, P-264
seals (*See also* gas seals), S-1 to S-9
advanced pattern groove (APG) in, S-3 to S-4
axial compressors and, C-162 to C-163, C-342
barrier type, S-8
carbon-graphite mix products and, C-12
in centrifugal compressors, C-180
centrifugal pumps and, P-250 to P-254
control panels and, S-8
electric motors and, E-17
GASPAC compressor seals, S-1 to S-9
in isotherm turbocompressors, C-206 to C-207, C-217, C-224
J ring, S-5 to S-9
lubrication of, C-336
O rings, S-1, S-5
polytetrafluoroethylene (PTFE) in, S-5 to S-9
in reciprocating (labyrinthine) compressors, C-110 to C-112, C-134
retrofitting of, S-8 to S-9
rotary pumps, P-276
in steam turbines, T-77 to T-80
symmetrical T groove (SMT) in, S-4 to S-5
seawater FGD, A-90, A-92
sebacic acid, C-68
sedimentation
centrifugal, C-35
computation fluid dynamics modeling of, C-1 to C-7
Seiko Co., P-4 to P-5

selective catalytic reduction (SCR), A-83, A-90, A-93 to A-96
selective noncatalytic reduction (SNCR), emission reduction in cement production, C-33
selenium, P-12 to P-31, P-40 to P-57
Selexol process, CO_2, C-15 to C-17
self-assessment, in environmental accountability, E-44
self-excited vibration, C-322 to C-323
self-locking worm gears, P-88
semibatch reactors, R-1
semiconductors, C-419
  conductive polymers in, P-5 to P-7
  thermoelectric cooling devices and, C-419
  thermoelectric cooling process in manufacture of, C-418 to C-423
Semipulse electrostatic precipitator, A-84 to A-85
SEN, generators and turbogenerators design, G-1
sensors (*See also* measurement; switches), A-40, A-46, C-377 to E-18
  Belleville washer negative spring rate sensor, C-386 to C-387
  bellows type, C-386
  bimetallic, C-377
  Bourdon tube, C-383 to C-385
  bulb and capillary, C-378 to C-379
  conductive polymers in, P-6
  electric transducer, M-23
  electrohydraulic actuators and, A-46
  gas actuated sensors in, C-378
  helical tube in, C-386
  liquid filled, C-378 to C-382
  liquid level/liquid flow control in, C-388 to E-18
  local mount liquid filled, C-379 to C-380
  magnetic bearings using, B-5 to B-9
  metal diaphragm, C-383
  negative spring rate, C-386 to C-387
  piston type pressure switches in, C-385 to C-386
  position, A-46
  pressure switches and, C-383 to C-388
  proof pressure in, C-383
  proof temperatures for, C-380 to C-381
  response time in, C-380
  RPM, A-46
  temperature switches, C-377 to C-382
  thermowells in, C-381 to C-382
  unsealed piston, C-387
  vapor pressure sensors in, C-377 to C-378
separation
  computation fluid dynamics modeling, C-1 to C-7
  in petroleum industry, R-2
separators, F-28 to F-29, S-10 to S-29
  capacity of, S-11 to S-12
  computational fluid dynamics to model, C-7
  cyclone tube in, S-12, S-17 to S-18
  design of, S-11 to S-12
  dry scrubbers, S-10 to S-29
  filter for, S-13 to S-16
  filter separators in, F-28
  fuel gas conditioning, F-28 to F-29
  high pressure type, S-27
  horizontal gas, S-27 to S-29
  knockout drums as, S-10
  line separators, F-29, S-24 to S-27
  mist extractor in, S-18 to S-19, S-22
  optional features of, S-12 to S-13
  pressure loss in, S-12
  removable tube section in, S-13
  steam type, for steam drum applications, T-108 to T-112
  vane type, S-19 to S-22
  vari line type, S-26 to S-27
  vertical absolute, F-29, S-16 to S-19
  vertical gas, F-27 to F-28, S-22 to S-24
Seraya Chemicals Singapore, C-41
service bulletins (SBs) and life cycle assessment, L-3 to L-6
service industries, pollutants and, P-28, P-31
service life for turbine components vs. material selection, M-41 to M-42
service symbols, Sp-3
servo tank gauges, T-18 to T-19, T-32 to T-33
servomotors
  in axial compressors and, C-161 to C-162
  in fuel system control, F-36
severity levels of accidents, A-9
sewage (*See also* wastewater), P-69 to P-72
sex hormone–related diseases, E-44
shaft gland seal ring, in steam turbines, T-78
shaft string drive, in isotherm turbocompressors, C-194 to C-200
shafts, synchronous clutches and, P-118
shale, O-1
shear, pumping applications and non Newtonian fluids, P-272
shear tubes, P-191 to P-192
sheathed element electric heaters, H-15 to H-16
short circuit, C-344
shortcircuiting, in chemical reactors, R-1
shot, A-1
  military specification (MIL-13165-C), A-7
shot peening, A-7 to A-9
Siemens, M-31
signal-to-noise (S/N) ratio, C-271
silencer outlet, air filtration systems and, A-64
silicon carbide, A-4, A-6
silicon controlled rectifiers (SCRs), H-18 to H-19
silicon dioxide (SiO_2), C-25
silos (*See* tanks)
silver nitrate, C-68
silver, P-12 to P-31, P-40 to P-57
Simmers, O-2 to O-4
Singapore Acrylic Pte Ltd (SAA), C-41
Singapore Monomer Pte Ltd (SMM), C-41
single pole double throw (SPDT) reed switch, C-390 to E-18
single pole single throw (SPST) reed switch, C-390 to E-18
sintering of catalyst, in air pollution control, A-96
sinusoidal waveforms, in vibrational analysis (VA), C-261 to C-262
site classification categories, waste management, W-1 to W-12
skid mount
  in isotherm turbocompressors, C-205
  in reciprocating (labyrinthine) compressors, C-108, C-128
skid, fuel skid, F-34 to F-36
slide valve, flow through, using computational fluid dynamics, C-2
sliding contact bearings, B-2 to B-5
sliding vane pumps, P-260 to P-264
slip curves, for gas turbines, C-347 to C-348
smart membranes, conductive polymers in, P-7
smoke test, in explosion hazard assessment, E-74 to E-75
snubbers (*See* pulsation dampeners)
SOAP, C-261
soap, in lubricating oil, L-25
Society of Automotive Engineers (SAE), A-7
sodium, as fuel impurity, L-9
sodium carbonate, C-68, C-72
sodium lauryl sulfate, C-68
sodium sulfate, C-68, C-72
solid or structural damping, C-307
solid solutions, M-39
solid-state technology, cooling products using, C-413
solubility curves of chemicals in crystallization of, C-71 to C-75
solvent extraction, R-5
solvents, C-72
  chlorofluorocarbons in, O-14, O-16 to O-26
  crystallization of, C-70
sorbic acid, C-68
sound (*See* soundproofing and noise control)
sound intensity levels (SIL), A-18 to A-19, A-22 to A-26
sound intensity meters, A-16
sound intensity probes, A-16 to A-17, A-21
sound pressure levels (SPL), A-11 to A-12, A-14 to A-16, A-18 to A-19
sound reduction index (SRI), A-22, A-27 to A-28, A-36
soundproofing and noise control, A-10 to A-37
  absorption of sound, A-11
  absorptive linings in, A-33 to A-37
  acoustic behavior of products in, A-27 to A-28
  acoustic design in, lightweight gas turbine enclosures, A-26 to A-37
  acoustic performance, A-37
  active vs. reactive sound fields in, A-20
  attenuation of sound, A-33 to A-37
  background noise measurement in, A-20 to A-21
  bending stiffness of panels vs., A-29 to A-31
  "coincidence frequency" in, A-27, A-30
  corrugated panels in, lined, A-33 to A-37
  corrugated panels in, unlined, A-29 to A-33
  damping in, A-28, A-33
  distance between source and measurement probe in, A-19 to A-20
  environment, environment's effects on, A-21
  fire ratings for, A-36
  flat panels in, lined, A-33 to A-37
  flat panels in, unlined, A-27 to A-29
  flexible connectors and, A-24 to A-25
  flow resistivity in absorptive materials and, A-34
  frequency in, A-28
  instrumentation and meters to measure sound in, A-14, A-16 to A-17
  intensity probe, A-21
  limitations to measurement in, A-20 to A-22
  mass law performance in, A-33
  measurement guidelines and standards for, A-17 to A-20
  motors, relative noise of, A-24
  near vs. far fields in, A-20
  nomenclature of, A-26
  pressure-intensity (P-I) index of sound in, A-16, A-18 to A-19
  propagation of sound in, A-27
  rank ordering of components in terms of A-weighted SPLs, A-23
  reactivity index of sound, A-12, A-16, A-18 to A-19
  resonance in, A-27 to A-28
  sound intensity (SIL) in, A-14 to A-19, A-22 to A-26
  sound intensity measurement in, A-11 to A-14
  sound power (PWL) in, A-14 to A-16, A-18 to A-19
  sound pressure levels (SPL) in, A-11 to A-12, A-14 to A-19
  sound reduction index (SRI) in, A-22, A-27 to A-28, A-36
  sound reference levels of, A-15 to A-16
  soundproofing panels for, testing of, A-21 to A-22, A-26 to A-33
  source location and, A-21, A-25 to A-26
  standards for, A-17 to A-20

subareas in measurement of, A-19
surface area vs., A-12
sweep vs. point measurement of sound in, A-19
tonal noise source and, measurement of, A-22
transmission loss in, A-27 to A-28, A-31 to A-33
uncertainty of determination of sound pressure levels, A-18
soundproofing panels, A-21 to A-22, A-26 to A-33
sour water, R-9
specific heats ratio, in flow measurement, M-24
specific weights in, M-25
specifications
American Society of Mechanical Engineers (ASME), A-1
commonly used, Sp-1 to Sp-3
gears, P-75 to P-76
life cycle assessment (LCA) and, L-19
lubrication and, L-24 to L-25
tanks, for potable water, T-9 to T-13
spectrum analyzer
automatic peak hold in, C-272
transient data capture in, C-271
in vibrational analysis (VA), C-266, C-270 to C-272, C-298 to C-299
speed (rpm), fans, F-1
speed protection device
electrohydraulic actuator in, A-47 to A-48
trigger criteria in, A-48
speed regulation
in hydrodynamic power transmission, P-155 to P-164
in torque converters, P-174
spin tubes, air filtration systems and, A-57
splash type lubrication, L-25
sponge ball cleaning system, condensers, C-248 to C-250
squirrel cage induction motors, E-19
stacks, S-29 to S-38
cleaning of, C-78
cost analysis for, height vs. cost, S-30 to S-32
design parameters for, S-30 to S-33
gust loading on, S-35 to S-38
height of, S-30 to S-33
nomenclature used with, S-30 to S-31
steady-state wind loading on, S-33 to S-35
steel, Sp-18
supports for, S-30 to S-33
weight of, S-30 to S-33
staged air combustion, emission reduction in cement production, C-33
stainless steel, M-40
in air filtration systems and housing, A-58 to A-63, A-71 to A-72
in condensers, C-245 to C-246
stall, C-328, T-48 to T-49
standards, Sp-1 to Sp-3
American Society of Mechanical Engineers (ASME), A-1, Sp-36
starters
synchronous clutches and, P-107 to P-108
torque converters and, P-171 to P-172
static bioassay, R-9 to R-10
static excitation, in generators and turbogenerators, G-16 to G-17
static head, P-219, P-221 to P-224
static pressure (hs), F-1 to F-5, M-15
static taps, pressure measurement and, M-15
static thyristor rectifier, in generators and turbogenerators, G-3
static tubes, in pressure measurement, M-15
Statoil, C-14
stators
in electric motors, E-19
in generators and turbogenerators, G-4 to G-5
in isotherm turbocompressors, C-211 to C-215, C-223 to C-224
in magnetic bearings, B-5 to B-7
steady-state oscillations, vibrational analysis (VA) and, C-313 to C-314
steady-state wind loading, stacks and, S-33 to S-35
steam drums, steam separators for, T-108 to T-112
steam dump device (SDD), condensers, C-248 to C-250
steam generators (*See also* boilers), S-38
steam rate calculations, for steam turbines, T-92 to T-94
steam regulators, current to pressure (I/P) converters for, C-392 to C-401
steam seal system, in steam turbines, T-79 to T-80
steam supply, S-38
steam turbines (*See* turbines, steam)
steel belt conveyors, C-408
steels and iron, M-40
Stefan Boltzmann constant, M-10
stepper motor–driven valves, C-403 to C-405
SteriGenics food irradiation facility, I-1 to I-5
sterilants, chlorofluorocarbons in, O-14, O-16 to O-26
sterilization, of air, A-101 to A-102
Sterling Drug, D-17
sterols, C-68
stiffness coefficient, vibrational analysis (VA), C-305 to C-306
Stokes Vacuum Inc. dryer, D-12 to D-13
storage and warehousing industries, pollutants and, P-29
storage tanks (*See* tanks)
stress tables, ASME, Sp-33
strip chart recorders, vibrational analysis (VA), C-270
strippers, T-40
stripping, metallurgy, M-32, M-34, R-10
strobe lights, vibrational analysis (VA) and balancing, C-278 to C-280
stroboscopes, vibrational analysis (VA), C-282 to C-286
stuffing boxes, centrifugal pumps and, P-248 to P-254
styrene, P-12 to P-31, P-38, P-40 to P-57
styrene monomer/propylene oxide (SMPO), C-41
styrene oxide, P-12 to P-31, P-38, P-40 to P-57
subcooling, condensers, C-240
sublimation, freeze drying, D-8
subsynchronrous frequency vibration, C-291 to C-294
suction head, P-222
net positive (NPSH), P-232 to P-237, P-257
suction lift, P-222, P-272 to P-273
suction type electric heater, H-13
sugar/sucrose and TBA in freeze drying, D-8 to D-18
sulfate levels in lakes, Ontario and Quebec, A-13
sulfur, C-15, L-9
sulfur dioxide (SO_2), A-83, C-34, E-38, F-43, F-45, L-9, P-60 to P-62, V-16
sulfur hexafluoride, V-16
sulfur trioxide (SO_3), C-25, V-16
sulfuric acid, A-9, P-12 to P-31, P-40 to P-57
sulfur oxides (SO_x), A-9 to A-10, E-36, E-50
Sulzer–Burckhardt, C-84 to C-232, E-42
Sumitomo Chemical, C-41
Sumitomo Pharmaceuticals Ltd., D-17
Suomen Rehu, E-46
superabrasive wheels, vitrified, A-2
superabrasives, A-1
superalloys, M-42 to M-43
abrasives and, A-2
cobalt based, A-2
nickel based, A-2
welding, M-38
superheaters, vaporizers, V-1 to V-5, V-13 to V-17
superposed steam turbines, T-96 to T-98
support systems
flexible, vibrational analysis of, C-318 to C-321
rigid, vibrational analysis of, C-318 to C-321
surface finish, A-1
surface quality, Sp-19
surge, C-264, T-48 to T-49
surge control (*See also* antisurge control), C-328 to C-330
sweetening processes, R-5
swing check valves, C-392
switches (*See also* measurement; sensors), switches, C-377 to C-391
absolute pressure switches in, C-383
actuation values in, C-377
Belleville washer negative spring rate sensor in, C-386 to C-387
bellows type sensors in, C-386
bimetallic type sensors, C-377
Bourdon tube sensors in, C-383 to C-385
bulb and capillary sensors in, C-378 to C-379
differential pressure-actuated, C-383
flow measurement sensors, C-389
gas actuated sensors in, C-378
gauge type instruments in, C-383
helical tube sensors in, C-386
liquid filled sensors in, C-378 to C-382
liquid flow instrumentation, C-388
liquid level instrumentation, C-388
local mount liquid filled sensors in, C-379 to C-380
mechanical thermal (MTS), P-202 to P-204
metal diaphragm sensors, C-383
negative spring rate sensor in, C-386 to C-387
noncontact thermal (BTS), P-202 to P-204
piston type pressure switches in, C-385 to C-386
pressure switches, C-383 to C-388
proof pressure in, C-383
proof temperatures for, C-380 to C-381
reed switch, C-390
response time in, C-380
sensors and, C-377
single pole double throw (SPDT) reed switch, C-390
single pole single throw (SPST) reed switch, C-390
temperature switches, C-377 to C-382
thermowells in, C-381 to C-382
unsealed piston sensors in, C-387
vapor pressure sensors in, C-377 to C-378
symmetrical T groove (SMT) seals, S-4 to S-5
synchronous clutch couplings, P-107 to P-115
alternators and, P-107, P-112
automatic turning gears and, P-107 to P-113
clutch assembly in, P-118
cogeneration and, P-107 to P-108, P-114 to P-115
design lettering designations for, P-111
elements of, P-107 to P-108
energy recovery and, P-107 to P-108, P-114 to P-115
flanges in, rigid, P-118
flexible couplings in, P-118
freestanding (type MS), P-113 to P-114
geared systems for, P-118
generators and, P-115 to P-121

instrumentation for, P-118 to P-121
journal bearings in, P-116 to P-117
power station, power generator use of, P-107 to P-113
quill shafts in, P-118
sectional view of, P-109
teeth in, P-116
thrust bearings in, P-117
thrust cones in, P-117 to P-118
type designations of (MS, DS, HS), P-110
syngas, C-17 to C-18
synthetic crude, O-1 to O-2
System 3 tank exterior coating, T-5

tank capacity tables (TCTs), T-24
tank cars, toxic chemical unloading procedures for, C-46 to C-52
tank gauges, T-13 to T-40
central inventory management system for, T-36 to T-39
diversion systems for, T-30 to T-31
float and tape, T-18
future developments in, T-39 to T-40
grounding for, T-31 to T-32
hybrid inventory measurement system (HIMS), T-22 to T-24, T-26 to T-27, T-36, T-39
hydrostatic (HTG), T-21 to T-22, T-25 to T-26, T-35 to T-36, T-39
manual, T-17 to T-18
quantity assessment using, T-24 to T-27
radar, T-19 to T-21, T-33 to T-34
servo, T-18 to T-19, T-32 to T-33
suppression systems for, T-30
temperature type, T-34 to T-35
uncertainties and errors in, T-27 to T-28
tank heaters, H-12 to H-15
tanker trucks, toxic chemical unloading guidelines for, C-51 to C-52
tanks, storage, T-1 to T-40, Sp-18
central inventory management system for, T-36 to T-39
density at reference temperature (DENS.REF) in, T-25 to T-27
errors and uncertainties in tank gauging and, T-27 to T-28
exterior coatings for, T-5
fire and explosion hazards, T-28 to T-29
flame arrestors for, V-31
float and tape gauges in, T-18
gauges for, T-13 to T-40
general storage type, T-1 to T-8
Gravi Merge Blending Silos, T-8
gravity blending in, T-4 to T-8
gross observed volume (GOV) in, T-25 to T-27
gross standard volume (GSV) in, T-25 to T-27
grounding for, T-31 to T-32
high voltage diversion systems for, T-30 to T-31
high voltage suppression systems for, T-30
hybrid inventory measurement system (HIMS) tank gauges in, T-22 to T-24, T-26 to T-27, T-36, T-39
hydrostatic tank gauges (HTG) in, T-21 to T-22, T-25 to T-26, T-35 to T-36, T-39
interior coatings for, T-4 to T-8
inventory control in, tank gauging for, T-15
leak detection and repair in, T-16 to T-17
lightning strikes and, T-29 to T-30
liquid storage, T-9 to T-13
manual gauging in, T-17 to T-18
oil movement, T-16
pneumatic conveying fill system for, T-7
quantity assessment, using tank gauges, T-24 to T-27
radar tank gauges in, T-19 to T-21, T-33 to T-34
recirculation blending while filling in, T-8
safety and, T-28 to T-32
servo gauges in, T-18 to T-19, T-32 to T-33
specifications sheets for, sample of, T-9 to T-13
tank capacity tables (TCTs) for, T-24
temperature gauges for, T-34 to T-35
toxic chemicals, C-56 to C-58
transfer of product, tank gauging for, T-15 to T-16
vaporizers, pressurized, V-22 to V-25
vents in, V-31
volume correction factor (VCF) in, T-25 to T-27
water storage (potable), T-9 to T-13
welded exterior coatings for, T-5
welded interior coatings for, T-4 to T-8
tar sands, O-1
TCR, air pollution control and, A-99 to A-100
TECA (cooling), C-413 to C-423
teeth, gear, P-88 to P-93, P-116
temperature control
in generators and turbogenerators, G-18 to G-19
in turbocompressors, C-333
temperature measurement (*See also* temperature switches), C-260 to C-261, M-1 to M-14, M-30 to M-31
absorptivity in, M-10
bimetallic thermometers for, M-8 to M-9
calibration of instruments for, M-11 to M-12
Celsius scale in, M-12
conduction error in, M-13
emissivity in, M-10
Fahrenheit scale in, M-12
filled system thermometers in, M-8
liquid in glass thermometers in, M-1 to M-2
multiresistance thermometers for, T-34 to T-35
multitemperature thermometers, T-34 to T-35
optical pyrometer in, M-11
pyrometers in, M-11
pyrometry in, M-10
radiation error in, M-13
radiation thermometers in, M-10 to M-11
in rapidly moving gas, M-13 to M-14
resistance thermometers in, M-2 to M-5
steam turbines and, T-80
Stefan Boltzmann constant in, M-10
tank gauges for, T-34 to T-35
thermistors in, M-4 to M-5
thermocouple thermometers for, M-5 to M-8
thermometers for, M-1 to M-8
tolerance in thermometers for, M-2 to M-3
temperature switches (*See also* temperature measurement), C-377 to C-382
actuation values in, C-377
bimetallic sensors in, C-377
bulb and capillary sensors in, C-378 to C-379
liquid filled sensors in, C-378 to C-382
local mount liquid filled sensors in, C-379 to C-380
proof temperatures for, C-380 to C-381
response time in, C-380
thermowells in, C-381 to C-382
vapor pressure sensors in, C-377 to C-378
temperature transmitter, C-333 to C-334
tertiary butyl alcohol (TBA) (*See also* freeze drying), D-5 to D-18
test codes, ASME, Sp-34 to Sp-35
testing
of centrifugal compressors, C-188 to C-189
eddy current, M-36
of fuel systems and controls, F-34 to F-36
of generators and turbogenerators, G-17 to G-18
of integrally geared compressors, C-226
liquid natural gas (LNG) and, L-23 to L-24
of reciprocating (labyrinthine) compressors, C-110
tetracalcium aluminoferrite (C4AF), C-25 to C-26
tetrachloroethylene, P-12 to P-31, P-39 to P-57
tetracholorbenzene, C-68
tetrahedral mesh modeling, C-5
tetramethy benzene, C-68
textile industries, pollutants and, P-17 to P-18
texts, Sp-1 to Sp-3
theoretical capacity (Q_t), pumps and, P-265
thermal conductivity gauges, M-22
thermal efficiency, steam turbine, T-82, T-84
thermal expansion factor, in flow measurement, M-24
thermal expansion, coefficient of, carbon-graphite mix products, C-9
thermal rating, worm gear, P-94
thermally densified coatings (TDC), M-35
thermistors, M-4 to M-5
thermocouples, M-5 to M-8, M-22
pressure measurement, M-22
synchronous clutches and, P-118
temperature monitoring, C-260 to C-261
thermometers for, M-5 to M-8
thermodynamics
in condensers, C-242 to C-244
in gas turbines, T-55
thermoelectric cooling process, C-418 to C-423
thermoforming, computational fluid dynamics to model, C-7
thermometers, M-1 to M-8
multiresistance, T-34 to T-35
multitemperature, T-34 to T-35
thermoset plastics, Sp-18
thermowells, C-381 to C-382
thin film processors (*See* chillers)
thiourea, P-12 to P-31, P-39 to P-57
threads, Sp-18
three-lobe journal bearings, B-3
threshold limit value (TLV), C-61, R-19
throat area, in flow measurement, M-24
throat diameter, in flow measurement, M-24
through-flow control, in hydrodynamic brakes, B-18
thrust bearings (*See also* axial thrust bearings), B-4, B-6, B-8
in axial compressors, C-163 to C-164
in centrifugal compressors, C-178 to C-181
in steam turbines, T-78
in synchronous clutches, P-117
thrust cones, in synchronous clutches, P-117 to P-118
thyristors
electric heater, H-16 to H-19
rectifiers using, G-3
tilting pad journal bearings, B-4, L-25
timber production (*See also* forest products), F-18 to F-21
time between overhaul (TBO), L-7 to L-10
adjustment to, based on LCA, L-7 to L-8
condition monitoring (CM) and, C-256
time waveform analysis, in vibrational analysis (VA), C-270, C-280 to C-282
time weighted average (TWA) exposure, refrigerants, R-19
timed screw pumps, P-260 to P-264
titanium and titanium alloys, C-246, C-251, M-40, M-42
titanium dioxide, V-15
titanium tetrachloride, P-12 to P-31, P-40 to P-57
Toagoseia, C-41
tobacco industries pollutants, P-29
tolerance in degrees, thermometers, M-2

tolerances
ASME certification program for, Sp-30
carbon-graphite mix products and, C-10
toluene diisocyanate (TDI), C-42 to C-62, P-12 to P-31, P-39 to P-57, V-15
Chemical Manufacturers Association (CMA) guidelines for, C-61
CHEMTREC information service, telephone number, C-61
color/discoloration of, at unloading, C-49
container for, C-45
cylinders for shipment of, unloading, C-52 to C-54
decontamination of empty toxic chemical containers, C-62
disposal of, C-62
drum unloading of, C-54
fire hazards of, C-59
first aid procedures for, C-61
health hazards of, C-59
heat and, reactivity of, C-43, C-45
Material Safety Data Sheets (MSDS) for, C-43, C-58
physical properties and toxicity of, C-43
protective clothing for handling of, C-59 to C-61
reactivity of, C-43 to C-45, C-58 to C-59
respirators for safety in handling, C-59 to C-61
shipping and handling of, C-45 to C-54, C-58 to C-61
spills and leaks of, C-61 to C-62
storage containers for, C-56 to C-58
tank car unloading, C-46 to C-52
tank truck unloading of, C-51 to C-52
thawing of, C-54 to C-56
Threshold Limit Value of, C-61
Types I and II, C-43
unloading of, C-45 to C-54
water and, reactivity of, C-43
white precipitates of, at unloading, C-48 to C-49
toluene, P-12 to P-31, P-39 to P-57
toluene-2,4-diisocyanate, P-12 to P-31, P-40 to P-57
Tomlinson boiler, P-210
tonal noise source, measurement of, A-22
tools, Sp-19 to Sp-22
tooth breakage, gear, P-74 to P-75
topping turbines, T-97
torque, C-344
in electric motors, E-3 to E-4, E-10 to E-15, E-23 to E-25
hydrodynamic braking and, B-16, B-20 to B-21
in hydrodynamic power transmission, P-128, P-133 to P-134
in torque converters, P-170 to P-171, P-174
torque converter (*See also* hydrodynamic power transmission), P-166 to P-184
adjustable, P-169
characteristics of, P-134 to P-136
compact design (Vorecon type RWE) in, P-174 to P-177
components of, P-170 to P-171
expanded speed range applications (Vorecon type RWS) in, P-177
filling and draining facilities in, P-172 to P-173
gas turbine starter for, P-171 to P-172
hydrodynamic power transmission and, P-123 to P-127
models available, P-174
multistage variable speed drives and, P-181 to P-184
operation of, P-170 to P-171
power vs. efficiency in, P-177
relieve startup application (Vorecon type RW) in, P-177 to P-181
schematic of, P-172
speed regulation in, P-174
symbols for, P-169
torque in, P-170 to P-171, P-174
variable speed engine and fluctuating load in, P-172 to P-174
torque limiting safety coupling, torque converters and, P-184 to P-191
total equivalent warming impact (TEWI), refrigerants and, R-12 to R-14
total head, P-222
total pressure, M-15
total quality management (TQM), environmental accountability and, E-40 to E-49
total signal, in vibrational analysis (VA), C-263
towers and columns, T-40
toxaphene, P-7
toxic chemicals (*See* chemicals; hazardous materials)
toxic gases, C-86
trade contracting industries, pollutants and, P-28
trajectories, computational fluid dynamics to calculate, C-3
transducers
pressure measurement, M-22 to M-24
vibrational analysis (VA) and, C-266 to C-270
transfer of pollutants, P-9, P-49 to P-54
transformers, in generators and turbogenerators, G-19
transient data capture, spectrum analyzers, C-271
transient vibration in, C-264
transistors, power transistors and electric motors, E-29 to E-32
transition voltage (dV/dt) time, electric motors and, E-30
translational velocity, vibrational analysis (VA), C-265
transmission loss, sound, A-27 to A-33
transmissions (*See* power transmission)
transmitters, electric transducer, M-23
transportation equipment industries, pollutants and, P-22
transportation industries, pollutants and, P-28
transportation of bulk chemicals, T-40 to T-41
tray-type reactors, T-40
treatment process, in petroleum industry, R-2, R-4 to R-5
tricalcium aluminate (C3A), C-25 to C-26
tricalcium silicate (C3S), C-25 to C-26
trichloroethane, P-12 to P-31, P-39 to P-57, V-15
Trilok converters, hydrodynamic power transmission and, P-138
trimethylbenzene, P-12 to P-31, P-40 to P-57
triple redundancy, T-41 to T-43
tropical environments vs. air filtration, A-50 to A-61
trucks, industrial, Sp-11
tube bundles, tubes
condensers, C-234, C-240 to C-242, C-244 to C-246
vaporizers, V-7 to V-10
tubesheet condensers, C-246, C-251
tubular reactors, R-1
tungesten inert gas (TIG) welding, M-38
tungsten alloys, M-42
TURBAIR vacuum blowers, C-149 to C-150
turbine damage, in air filtration systems, A-58
turbine discs, material selection for, M-40
turbine flow measurement, C-390
turbine inlet temperatures (TIT), monitoring of, C-261
turbine vanes (*See* vanes, turbine)
turbines, gas, T-43 to T-76, Sp-22
acoustic design in, lightweight gas turbine enclosures, A-26 to A-37
acoustic enclosure for soundproofing and noise control in, A-10 to A-37
aeroderivative fuel regulator for, C-354 to C-358
air filtration for (*See* air filtration)
air/gas conditions in, T-53 to T-55
airflow heating in, T-45 to T-46
airflow through, T-44 to T-46, T-49, T-53 to T-55
ambient air pressure (P_0) in, T-55 to T-58
ambient air temperature (T_0) in, T-58
ambient air relative humidity in, T-58, T-65 to T-67
atmospheric conditions vs. power degradation in, T-65 to T-67
axial flow compressor design in, T-47 to T-48
axial flow through, T-51 to T-53
biomass fuel fed, from pulp and paper mill, P-210 to P-211
brake properties at high speed and acceleration, C-347 to C-350
cleaning or washing of, T-59 to T-75
closed cycle, T-44 to T-45
clutch type couplings in, C-347
combustion chamber in, T-49 to T-51
compressor for, T-47
condensers connection to, C-239
configurations for, T-53
constant fill couplings, C-347
description of, T-43 to T-60
deterioration of, L-10
efficiency of, T-43 to T-44
fouling of compressors in, T-63 to T-65, T-67
gearboxes in, C-343
gears in, C-345
hydrodynamic principle in turbocouplings and, C-346 to C-347
integrated control system (Series 9500) for, C-359 to C-361, C-374 to C-377
life cycle assessment (LCA) and, L-15 to L-17
offshore environments for, A-26
onshore environments for, A-26
open cycle, T-44 to T-45
operating conditions for, T-58 to T-60
operating cycle in main parts of, T-44
overload/overspeed control, C-336, C-343 to C-350
performance analysis of, T-55
power generation in, T-47
Safeset overspeed/overload protective coupling, C-344 to C-350
self-sustaining speed of, T-46 to T-47
slip curves for, C-347 to C-348
stall in, of compressor, T-48 to T-49
surging in, T-48 to T-49
thermodynamic factors in, T-55
torque converters and for starter in, P-171 to P-172
turbine in, T-51 to T-53
variable speed couplings in, C-347
witness testing of soundproofing for, A-22 to A-26
turbines, steam, T-76 to T-112, Sp-22
auxiliary systems for, T-78 to T-80
back pressure, T-96 to T-98
bearing housings in, T-78
blades for, T-76, T-88 to T-90
casing for, T-76
classification of, T-87
control systems for, T-78 to T-79
deterioration of, L-11
diaphragms in, T-77
efficiency of, T-82, T-84, T-89 to T-90

enthalpy in, T-82, T-84
extraction type, T-98 to T-100
feedwater heating in, regenerative, T-85 to T-87
governors for, T-100 to T-108
helical flow stage in, T-91
impulse stage in, T-89
induction type, T-98 to T-100
interstage steam sealing in, T-77
journal bearings in, T-78
low pressure type (high pressure insert), T-100
lubrication system for, T-79, T-109 to T-110
material selection for, M-42
Mollier diagrams for, T-82, T-84
multistage condensing, T-94 to T-96
nozzle rings for, T-76 to T-77
nozzles in, T-88 to T-89
oil supply system in, T-79
operational theory of, T-80 to T-108
overload/overspeed protection in, C-368 to C-374, T-78, T-106 to T-108
p5 type, T-111
power output from, T-81
pressure control in, C-368 to C-374
Rankine cycle in, T-82 to T-84
reaction stage in, T-88 to T-89
regenerative cycle in, T-85
reheat cycle in, T-84 to T-85
rotors in, T-76
seals in, T-77
shaft gland seal ring in, T-78
single stage, T-91 to T-94
steam cycles in, T-82 to T-87
steam path design in, T-87 to T-91
steam rate calculations for, T-92 to T-94
steam seal system in, T-79 to T-80
steam separators for steam drum applications, T-108 to T-112
superposed, T-96 to T-98
temperature sensors in, T-80
thermal efficiency in, T-82, T-84
thrust bearings in, T-78
topping, T-97
turning gear system in, T-79
valve gear assembly for, T-78
vibration monitoring in, T-80
turbochargers, E-38 to E-39, T-112 to T-114
turbocompressors, C-136 to C-232
antisurge control in, C-332 to C-335, C-359 to C-368
control systems for, C-331 to C-336
discharge pressure control in, C-333
flow control in, C-333
load limit control in, C-334
monitoring logic in, C-336
power limitation in, C-333
pressure control in, C-333
pressure limitation in, C-333
process control in, C-333 to C-334
safety systems and, C-332 to C-333
suction pressure control in, C-333
temperature limitation in, C-333
vibration limitation in, C-333
turbocouplings, hydrodynamic principle in, C-346 to C-347, P-191 to P-196
turboexpanders, C-151 to C-152, T-114 to T-115
turbogenerators (*See* generators), G-1 to G-20
turbomachinery, vibrational analysis (VA) of, C-268, C-297 to C-298, C-308
turbotransmission (*See* hydrodynamic power transmission)
turbulence errors, M-15
Turcon CTo electrohydraulic actuator, A-47
turning gear system, in steam turbines and, T-79
two-stroke engines, E-38 to E-40
Udimen, M-41
UL/CSA design standards, cooling, C-418
ultrasonic cleaning, U-1 to U-9
booster for, U-3
converter for, U-2
EPA environmental testing packages using, T-5 to T-6
1500 watt processors, T-7 to T-9
400 and 600 watt processors, T-6 to T-7
power supply for, U-1 to U-2
probes for, U-2 to U-3
ultrasonic flow measurement, C-390
ultrasonic method, in level sensing, C-388
ultraviolet irradiation
air purification using, A-102
ozone and, O-5 to O-6
underdamping, C-312 to C-313
Uniax, P-5
United National Environment Programme (UNEP), ozone and
United States Surgical Corporation, D-17
unsealed piston sensors, C-387
untimed screw pumps, P-260 to P-264
Upjohn Company, D-17
urea, C-33
urethane, C-58
user interface, in magnetic bearings, B-9
utility industries pollutants, P-29

vacuum blowers, TURBAIR, C-149 to C-150
vacuum crystallization process, C-71
vacuum devices, ejectors, E-2 to E-3
vacuum distillation, petroleum industry, R-2
vacuum drying, D-4, D-12 to D-13, E-2 to E-3
vacuum pressure, M-15
vacuum pumps, P-309 to P-325
dry types, P-312
liquid jet, P-321 to P-325
liquid types, P-312
single stage, liquid ring, P-318 to P-321
valves, A-40, C-107, Sp-22 to Sp-23
antisurge, C-332 to C-335
ASME accreditation packages for, Sp-25 to Sp-28
ball, CE-18 to CE-392
butterfly, C-392
centrifugal pumps and, P-257
control, CE-18 to CE-392
current to pressure (I/P) converters and, C-392 to C-401
diaphragm, CE-18 to CE-392
diverters, D-3
electrohydraulic actuators in, A-43
fast response, C-405 to C-408
flapper, D-3
float type, V-25 to V-28
fuel regulation using, C-403 to C-408, F-30, F-32 to F-33
gate, A-43
globe, CE-18 to CE-392
knife gate, C-399, C-402 to C-403
lift check, C-392
manifold flow (fuel) control using, C-403 to C-408
Moog, C-405
needle globe, CE-18 to CE-392
nonreturn or check, C-392
pinch, CE-18 to CE-392
plug, CE-18 to CE-392
pressure regulation (PRV) in, F-25 to F-26
in reciprocating (labyrinthine) compressors, C-88 to C-91, C-95, C-115, C-122 to C-124, C-135
in steam turbines, T-78
stepper motor–driven in, C-403 to C-405
swing check, C-392
Van T'Hoff equation, crystallization, C-72 to C-75
vanadium, A-6, L-9, P-12 to P-31, P-40 to P-57
vane type separators, A-63, A-74, S-19 to S-22
vanes, turbine (*See also* metallurgy)
material selection for, M-41
vapor coatings, M-35
vapor pressure sensors, C-377 to C-378
vapor type electric heater, H-11
vaporization, C-62
vaporizers, V-1 to V-31
ammonia in, V-16
applications for, V-15 to V-16
chlorine in, V-16
combination electric indirect, V-20 to V-22
constant vapor outlet temperature in, V-29 to V-31
control systems for, V-25 to V-31
control systems in, V-1 to V-5
conversion factors for
cryogenic, V-5 to V-7, V-17 to V-19
design criteria, by fluid type, V-16 to V-17
design features of, V-9 to V-12
direct steam heated types, V-19
electric resistance, V-1 to V-5
electrical radiant furnace, V-5 to V-6
ethylene in, V-17
float valves in, V-25 to V-28
fluids vaporized in, V-15 to V-16
freeze up protection in, V-12 to V-13, V-17 to V-19
impedance electric heater, V-1 to V-5
indirect fluid heater, V-1 to V-5
lethal fluids in, V-17
liquid to vaporizer–vapor direct to load system in, V-28 to V-29
liquefied natural gas (LNG) in, V-17
operation of, V-10 to V-12
pressurizing storage tanks for, V-22 to V-25
propane and butane in, V-17
small electric indirectly heated, V-19 to V-20
specifications and process parameters for, V-6 to V-8
steam feed for, V-10
superheater, V-1 to V-5, V-13 to V-17
tube bundles in, V-7 to V-10
tubular low temperature, V-1 to V-5
types of, V-1 to V-5
vertical, V-1 to V-5
vertical bayonet type, V-7 to V-12
working pressure in, V-9
vari-line separators, S-26 to S-27
variable area flow measurement, C-390
variable frequency control, AC induction motors used for, E-18 to E-35
variable frequency drives (VFD), in centrifugal compressors, C-185
variable pitch fan, F-3
variable speed couplings, C-347
in hydrodynamic power transmission, P-138 to P-147, P-155 to P-164
Type RK variable speed coupling in, P-147 to P-149
Type SVL variable speed coupling in, P-145 to P-147
Type SVNL, P-142 to P-145
variable speed drives, torque converters and, P-181 to P-184
vegetation
acid rain and, P-60 to P-62
ozone and damage to, O-8 to O-9
velocity
in fans, F-3 to F-4
in vibrational analysis (VA), C-287, C-305, C-311
velocity head, M-15, P-221 to P-224
velocity of approach factor, flow measurement, M-24
velocity pressure, F-4, M-15

velocity probes, in vibrational analysis (VA), C-268
ventilation
  computation fluid dynamics in, C-1 to C-7
  vs. explosion hazard, E-70, E-76, E-87
venting, in condensers, C-246 to C-250
vents and flame arrestors, V-31
Venturi tubes, C-330, C-333, M-23, M-26, M-29
verification, in environmental accountability, E-44
vertical absolute separators, F-29, S-16 to S-19
vertical bayonet vaporizers, V-7 to V-12
vertical cluster in tank electric heater, H-14
vertical deflection, in vibrational analysis (VA), C-262
vertical dry scrubbers, F-28 to F-29
vertical gas separators in, F-28, S-22 to S-24
VHBI hydrodynamic brakes, B-14 to B-16, B-25 to B-28
VHBK hydrodynamic brakes, B-27
VHBM hydrodynamic brakes, B-14 to B-16
vibrational analysis (VA) (*See also* condition monitoring), C-253, C-257 to C-258, C-261 to C-328, O-1
  acceleration, conversion factors for, C-265
  acceleration in, C-287, C-305, C-311
  accelerometers in, C-269, C-324
  amplitude in, C-263, C-268, C-302, C-315 to C-316
  amplitude vs. frequency in, C-270 to C-272
  amplitude vs. frequency vs. time in, C-270, C-272
  amplitude vs. phase in, C-280 to C-282
  amplitude vs. phase vs. machine rpm in, C-270, C-272 to C-280, C-300
  amplitude vs. time in, C-270
  anti node in plots of, C-276
  balancing and, C-278 to C-280, C-288 to C-291, C-325 to C-328
  blade tip rub and, C-294
  Bode plots in, C-270, C-272 to C-280
  Campbell diagrams for, C-323 to C-326
  cavitation in, C-264
  centrifugal pumps and, P-255 to P-258
  complex vibration in, C-263
  compressors and, C-264
  critical damping in, C-312 to C-313, C-315
  critical speed map for, C-322
  damped systems in, C-307 to C-313
  damping in, C-264, C-305 to C-306
  degree of freedom in, C-301 to C-302, C-310
  deterministic vibration in, C-263
  displacement and, C-287, C-314 to C-315
  displacement or proximity probes for, C-266 to C-270
  drive belts and, C-296
  dynamic absorbers and, C-276
  eddy current probes in, C-267 to C-268
  electric motors and, E-18 to E-22, E-33
  electronic data loggers in, C-270
  filter in/filter out amplitude vs. rpm in, C-276 to C-278
  filtered signal in, C-263
  flexible supports and, C-318 to C-322
  forced vibration in, C-264, C-301, C-313 to C-316, C-322 to C-323
  forcing frequency in, C-264
  free vibration in, C-264, C-301 to C-307
  frequency components of, C-87
  frequency in, C-262, C-300 to C-302
  friction (coulomb) damping in, C-307
  gear rpm and, C-294 to C-296
  gears, P-82
  generators and turbogenerators, G-19
  guideline limits for, C-300
  harmonic waveforms in, C-261 to C-262
  harmonics in, C-266 to C-267, C-282, C-302, C-305, C-310, C-314
  hydrodynamic power transmission and, P-128, P-132
  ideal vs. real systems in, C-264
  impeller and, C-325
  inertia in, C-314 to C-315
  influence coefficient balancing and, C-325 to C-327
  life cycle assessment (LCA) and, L-18 to L-19
  limitations in, C-269
  Lissajous patterns in, C-270, C-282 to C-286
  loose assemblies and, C-296
  magnification factor (D/F/k) in, C-316
  magnitude in, C-263
  mature units and, contingency corrective measures for, C-296 to C-297
  misalignment and, C-290 to C-291
  modal balancing in, C-327 to C-328
  natural frequency in, C-264 to C-265, C-307, C-316
  nonintrusive wear monitoring in, C-330
  Nyquist plots in, C-270, C-280 to C-282
  orbits in, C-270, C-282 to C-286
  oscillations of vibration in, C-261 to C-262
  overdamping and decay in, C-312
  packaged monitoring systems for, C-286 to C-287
  periodic frequency in, C-263
  periodic motion in, C-302, C-305, C-310
  phase analysis in, C-278 to C-280
  phase angle in, C-265, C-305 to C-306, C-315 to C-316
  principal modes of vibration in, C-302
  pulsating vibration in, C-264
  random vibration in, C-263
  recorders for, C-270 to C-286
  remote applications for, probe selection, C-270
  resonance in, C-264 to C-265
  rigid supports and, C-318 to C-321
  rotational velocity, conversion factors for, C-265
  rotor dynamics in, C-301 to C-325
  rotor systems and, C-318
  rubbing and, C-287
  self-excited vibration in, C-322 to C-323
  severity charts for, C-287
  signal-to-noise (S/N) ratio in, C-271
  "signatures" of, C-296 to C-297
  simple harmonic motion, C-266 to C-267
  sinusoidal waveforms in, C-261 to C-262
  solid or structural damping in, C-307
  spectrum analyzer for, C-266, C-270 to C-272, C-298 to C-299
  spring mass system and, C-305, C-311
  steady-state oscillations and, C-313 to C-314
  steam turbines and, T-80
  stiffness coefficient in, C-305 to C-306
  strip chart recorders in, C-270
  strobe lights for balancing and, C-278 to C-280
  stroboscopes for, C-282 to C-286
  subsynchronrous frequency orders and, C-291 to C-294
  surge in, C-264
  time waveform analysis in, C-270, C-280 to C-282
  total signal in, C-263
  transducers or probes used in, C-266 to C-270
  transient vibration in, C-264
  translational velocity, conversion factors for, C-265
  troubleshooting using, C-287 to C-298
  in turbocompressors, C-333
  turbomachinery and, C-268, C-297 to C-298, C-308
  underdamping and decay in, C-312 to C-313
  velocity in, C-287, C-305, C-311
  velocity probes in, C-268
  vertical deflection in, C-262
  viscous damping in, C-308 to C-311
  waterfall graphs or cascade diagrams in, C-272 to C-273
  waveforms in, C-261 to C-262
Vienna Convention for the Protection of the Ozone Layer, O-12 to O-13
vinyl acetate monomer (VAM), C-41
vinyl acetate, P-12 to P-31, P-40 to P-57, V-15
vinyl chloride, P-12 to P-31, P-40 to P-57, V-15 to V-16
vinylidene chloride, P-12 to P-31, P-40 to P-57, V-15 to V-16
visbreaking, in petroleum industry, R-3
viscosity, lubrication and, L-25 to L-26
viscous damping, C-308 to C-311
viscous fluids
  crystallization of, C-67
  pumps and pumping of, P-230 to P-232
  in rotary pumps, P-265 to P-267, P-270 to P-273
vitamins, C-68
Voith (*See* J.M. Voith GmbH)
Voith hydrodynamic brakes, B-14 to B-16
volatile organic compounds (VOCs), O-5 to O-6, O-10 to O-15
voltage requirements, electric motors, E-15, E-23 to E-24, E-29, E-32
voltage sags, in power supplies, C-350 to C-354
voltage unbalance, in electric motors, operating cost in, E-6 to E-7
volume correction factor (VCF), T-25 to T-27
Vorecon type RW torque converter, P-177 to P-181
Vorecon type RWE torque converter, P-174 to P-177
Vorecon type RWS torque converter, P-177
vortex flow measurement, C-390
Vosper Thornycroft, C-405
V-ring shaft slinger (Buna–N shaft seal), electric motors and, E-17

W.R. Grace, P-4
washing
  centrifugal, C-35 to C-38
  chemical cleaning and, C-38
waste heat recovery (WHR) (*See also* cogeneration), C-78, C-84, V-31
waste management (*See also* pollutants; recycling), E-54, F-20, W-1 to W-12, Sp-30
wastewater flow, P-69 to P-72
water
  in lubricating oil, L-25
  as refrigerant, coolant, R-15 to R-18
  tanks for, T-9 to T-13
water conservation, C-423 to C-426, F-17
water density, for fans, F-3
water eliminator, in air filtration systems, A-82 to A-83
water jet cleaning, M-34
water jet machining, M-35
water pollution, C-78, E-52, P-40 to P-48, P-64 to P-65
  accidents/incidents and, E-46
  biochemical oxygen demand (BOD) in, P-71
  condensers and, C-233
  desalination, D-1 to D-2
  environmental accountability, E-40 to E-49
  eutrophication in, P-64 to P-65
  forest conservation and, F-17
  Great Lakes, toxic chemicals and, P-65 to P-72
  industrial effluent in, P-71 to P-72

packaging and paper production process vs., F-24
phosphorus in, P-64 to P-65
"red tide" algae (*Ptychodiscus brevis*) and, C-78
sewage (*See also* wastewater), P-69 to P-72
waste management and, W-1 to W-12
wastewater flows in, P-69 to P-72
water system, C-109
waterboxes, condensers, C-238, C-248
waterfall graphs, in vibrational analysis (VA), C-272 to C-273
waveforms, in vibrational analysis (VA), C-261 to C-262
waxes, C-68, C-70
wear monitoring, nonintrusive, C-330
weather louvers/weather hoods, in air filtration systems, A-55 to A-61, A-74
weighting factors, in life cycle assessment (LCA), L-10
Weir Westgarth (WW), D-1
welding, C-1, M-31, M-37 to M-38
dabber TIG welding, M-38
laser, M-37
plasma transfer arc, M-38
robotics in, M-38
superalloy, M-38
tank exteriors, T-5
tank interiors, T-4 to T-8
Weldolet couplings, M-15
wet/dry flue gas desulfurization, A-89 to A-91, A-100
Wetpac TCR system, A-100 to A-101
white liquor, P-205 to P-214
Whittaker Controls (fuel systems), F-34 to F-36
wholesale products and industries pollutants, P-30 to P-31
wind loading, stacks and, S-33 to S-38
windings, computational fluid dynamics to analyze heating curves in, C-6 to C-7
wood industries pollutants, P-18
Worldwide Standard Service, ASME, Sp-32
worm gears, P-82 to P-98
adjustment of, P-92
bearing loading in, P-94
bolt stress in, P-94
circular pitch in, P-84
classification of, P-84 to P-85
coefficient of friction curves for, P-86 to P-88
design considerations for, P-92 to P-94
double reduction, P-84 to P-87
efficiency of, P-85 to P-88
gear blank under rim diameter in, P-93
gear face width in, P-93
gear pitch diameter in, P-93
gear ratio in, P-93
helical reduction, P-84 to P-87
horizontal shaft type, P-84
inner diameter of worm in, P-83
installation of, P-97 to P-98
lead angle in, P-84
lead of worm in, P-84
lubrication of, P-96 to P-97
mechanical elements of, P-83 to P-84
mechanical rating of, P-94
number of teeth in, P-92 to P-93
operation of, P-97 to P-98
outer diameter of gear in, P-83
outer diameter of worm in, P-83
performance of, P-89 to P-90, P-94 to P-96
pitch diameter of gear in, P-84
pitch of, P-93
pressure angle in, P-84, P-93
ratio of transmission in, P-84
reducers for
self-locking, P-88
shaft mount type, P-85
shaft stress in, P-94
single reduction, P-84 to P-87
speed ratios in, P-83
starting load for
thermal rating of, P-94 to P-96
throat diameter of gear in, P-83
tooth contact in, P-90 to P-92
tooth form in, P-88 to P-90
vertical output shaft type, P-84
worm face in, P-94
worm pitch diameter in, P-93

X rays (*See also* irradiation, food product), I-5
xylene, C-66, C-72, P-12 to P-31, P-40 to P-57

zeolite catalyst, R-10
zeotrope refrigerant mixtures, R-11
zinc, P-12 to P-31, P-40 to P-57
zinc primer tank exterior coating, T-5
zirconium oxide probes, O-1
zirconium oxide, A-4