

POWER PLANT EQUIPMENT OPERATION AND MAINTENANCE GUIDE

Maximizing Efficiency
and Profitability

Philip Kiamah

This page intentionally left blank

Power Plant Equipment Operation and Maintenance Guide

This page intentionally left blank

Power Plant Equipment Operation and Maintenance Guide

**Maximizing Efficiency
and Profitability**

Philip Kiameh, M.A.Sc., B.Eng., D.Eng., P.Eng.



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

Copyright © 2012 by The McGraw-Hill Companies, Inc. All rights reserved. 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 database or retrieval system, without the prior written permission of the publisher.

ISBN: 978-0-07-177222-8

MHID: 0-07-177222-7

The material in this eBook also appears in the print version of this title: ISBN: 978-0-07-177221-1,
MHID: 0-07-177221-9.

All trademarks are trademarks of their respective owners. Rather than put a trademark symbol after every occurrence of a trademarked name, we use names in an editorial fashion only, and to the benefit of the trademark owner, with no intention of infringement of the trademark. Where such designations appear in this book, they have been printed with initial caps.

McGraw-Hill eBooks are available at special quantity discounts to use as premiums and sales promotions, or for use in corporate training programs. To contact a representative please e-mail us at bulksales@mcgraw-hill.com.

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.

TERMS OF USE

This is a copyrighted work and The McGraw-Hill Companies, Inc. ("McGrawHill") and its licensors reserve all rights in and to the work. Use of this work is subject to these terms. Except as permitted under the Copyright Act of 1976 and the right to store and retrieve one copy of the work, you may not decompile, disassemble, reverse engineer, reproduce, modify, create derivative works based upon, transmit, distribute, disseminate, sell, publish or sublicense the work or any part of it without McGraw-Hill's prior consent. You may use the work for your own noncommercial and personal use; any other use of the work is strictly prohibited. Your right to use the work may be terminated if you fail to comply with these terms.

THE WORK IS PROVIDED "AS IS." McGRAW-HILL AND ITS LICENSORS MAKE NO GUARANTEES OR WARRANTIES AS TO THE ACCURACY, ADEQUACY OR COMPLETENESS OF OR RESULTS TO BE OBTAINED FROM USING THE WORK, INCLUDING ANY INFORMATION THAT CAN BE ACCESSED THROUGH THE WORK VIA HYPERLINK OR OTHERWISE, AND EXPRESSLY DISCLAIM ANY WARRANTY, EXPRESS OR IMPLIED, INCLUDING BUT NOT LIMITED TO IMPLIED WARRANTIES OF MERCHANTABILITY OR FITNESS FOR A PARTICULAR PURPOSE. McGraw-Hill and its licensors do not warrant or guarantee that the functions contained in the work will meet your requirements or that its operation will be uninterrupted or error free. Neither McGraw-Hill nor its licensors shall be liable to you or anyone else for any inaccuracy, error or omission, regardless of cause, in the work or for any damages resulting therefrom. McGraw-Hill has no responsibility for the content of any information accessed through the work. Under no circumstances shall McGraw-Hill and/or its licensors be liable for any indirect, incidental, special, punitive, consequential or similar damages that result from the use of or inability to use the work, even if any of them has been advised of the possibility of such damages. This limitation of liability shall apply to any claim or cause whatsoever whether such claim or cause arises in contract, tort or otherwise.

To my parents and all my family members

This page intentionally left blank

About the Author

Philip Kiameh, M.A.Sc., B.Eng., D.Eng., P.Eng., has been a teacher at the University of Toronto, Canada, for 20 years. He also teaches courses and seminars to engineers across Europe, North America, and the Middle East. Dr. Kiameh has written four books for working engineers and has won excellence in teaching awards from the Professional Development Center at the University of Toronto and TUV Akademie. He performed research on power generation equipment with Atomic Energy of Canada Limited and has more than 27 years of practical engineering experience with Ontario Power Generation, formerly Ontario Hydro, the largest electric utility in North America.

This page intentionally left blank

Contents

Preface	xxv
Acknowledgments	xxix
1 Gas Turbine Applications in Power Stations, Gas Turbine Protective Systems, and Tests	1
1.1 Introduction	1
1.2 Working Cycle	4
1.2.1 Starting	4
1.2.2 Shutdown	6
1.3 Protection	6
1.4 Black Start	7
1.5 Routine Tests	8
1.6 Bibliography	9
2 Steam Turbine Selection for Combined-Cycle Power Systems	11
2.1 Abstract	11
2.2 Introduction	11
2.3 Steam Turbine Application to Steam and Gas Plants	11
2.3.1 Steam and Gas Plants Structure	11
2.3.2 Steam Turbine Exhaust Size Selection	12
2.3.3 Non-Exhaust Cycle-Steam Conditions	12
2.3.4 Reheat Cycle Steam Condition	16
2.4 Steam Turbine Product Structure	17
2.4.1 Performance	17
2.4.2 Casing Arrangements	19
2.4.3 Cogeneration Applications	29
2.5 Bibliography	29
3 Steam Turbine Maintenance	31
3.1 Life Cycle Operating Cost of a Steam Turbine	31
3.2 Steam Turbine Reliability	31
3.3 Boroscopic Inspection	31
3.4 Major Cause of Steam Turbine Repair and Maintenance	31
3.5 Maintenance Activities	32
3.6 Advanced Design Features for Steam Turbines	34
3.7 Bibliography	39
4 Frequently Asked Questions About Turbine-Generator Balancing, Vibration Analysis, and Maintenance	41
4.1 Balancing	41
4.2 Vibration Analysis—Cam Bell Diagram	42
4.3 Turbine-Generator Maintenance	42

5 Features Enhancing the Reliability and Maintainability of Steam Turbines	45
5.1 Steam Turbine Design Philosophy	45
5.2 Measures of Reliability, Availability, and Maintainability	46
5.3 Design Attributes Enhancing Reliability	47
5.3.1 Overall Mechanical Design Approach	47
5.3.2 Modern Steam Turbine Design Features	48
5.4 Design Attributes Enhancing Maintainability	53
5.4.1 Maintainability Features	53
5.4.2 Maintenance Recommendations	60
5.5 Cost/Benefit Analysis of High Reliability, Availability, and Maintenance Performance	61
5.5.1 Reliability, Availability, and Maintainability Value Calculation	61
5.6 Conclusion	61
5.7 Bibliography	61
6 Steam Generators	63
6.1 Introduction	63
6.2 The Fire-Tube Boiler	64
6.3 The Water-Tube Boiler	66
6.3.1 The Straight-Tube Boiler	66
6.3.2 The Bent-Tube Boiler	68
6.4 The Water-Tube Boiler: Recent Developments	69
6.4.1 The Boiler Walls	70
6.4.2 The Radiant Boiler	72
6.5 Water Circulation	72
6.6 The Steam Drum	74
6.7 Superheaters and Reheaters	75
6.7.1 Convection Superheater	76
6.7.2 Radiant Superheater	76
6.8 Once-Through Boilers	78
6.9 Economizers	79
6.10 Air Preheaters	80
6.11 Fans	83
6.11.1 Fan Control	85
6.11.2 The Stack	86
6.12 Steam Generator Control	88
6.12.1 Feedwater and Drum-Level Control	88
6.12.2 Steam-Pressure Control	88
6.12.3 Steam-Temperature Control	89
6.13 Bibliography	91
7 Boilers (Steam Generators), Heat Exchangers, and Condensers	93
7.1 Heat Transfer	93
7.1.1 Steady-State Conduction	93

7.2	Thermal Conductivities	95
7.2.1	Conduction Through Cylindrical Walls	95
7.3	Combination Heat-Transfer Effects	96
7.4	Convection Heat-Transfer Coefficients	97
7.4.1	Turbulent Forced-Convection Flow Inside Long Circular Tubes	98
7.4.2	Streamlined Forced-Convection Flow Inside Tubes (Water and Oils)	98
7.4.3	Turbulent Forced-Convection Flow Across N Onbaffled Tube Banks with Circular Tubes	98
7.5	Boiling Liquids and Condensing Vapors	99
7.6	Heat Exchangers	99
7.6.1	Shell-and-Tube Heat Exchangers	101
8	Integrated Gasification Combined Cycles	107
8.1	Introduction	107
8.2	IGCC Processes	107
8.3	IGCC Plant Considerations	108
8.3.1	Turnkey Cost	108
8.3.2	Size of IGCC	109
8.3.3	Output Enhancement	109
8.4	Emission Reduction	109
8.4.1	Nitrogen Oxides	109
8.4.2	Air Pollutants	109
8.4.3	Mercury	109
8.4.4	Carbon Dioxide	109
8.5	Reliability, Availability, and Maintenance	110
8.6	Bibliography	110
9	Single-Shaft Combined-Cycle Power Generation Plants	111
9.1	Introduction	111
9.2	Performance of Single-Shaft Combined-Cycle Plants	112
9.3	Environmental Impact	114
9.4	Equipment Configurations	115
9.5	Starting Systems	116
9.6	Auxiliary Steam Supply	116
9.7	Plant Arrangement	116
9.8	Maintenance	117
9.9	Advantages of Single-Shaft Combined-Cycle Plants	117
9.10	Bibliography	118
10	Selection of the Best Power Enhancement Option for Combined-Cycle Plants	119
10.1	Plant Description	119
10.2	Evaluation of Inlet-Air Pre-Cooling Option	119
10.3	Evaluation of Inlet-Air Chilling Option	122
10.4	Evaluation of Absorption Chilling System	123

10.5	Evaluation of the Steam and Water Injection Options	123
10.6	Evaluation of Supplementary Firing in HRSG Option	124
10.7	Comparison of All the Power Enhancement Options	124
10.8	Bibliography	124
11	Economics of Combined-Cycle and Cogeneration Plants	125
11.1	Introduction	125
11.2	Natural Gas Prices	125
11.3	Economic Growth	126
11.4	Financial Analysis	127
11.5	Base Case	127
11.6	Combined-Cycle Configuration	128
11.7	Capital Cost	128
11.8	Operating and Maintenance Cost	128
11.9	Economic Evaluation of Different Combined-Cycle Configurations	135
11.10	Electricity Purchase Rate	140
11.11	Economic Consideration	140
11.12	Conclusions	140
11.13	Bibliography	141
11.14	Appendix: Definitions of Terms Used in the Tables	141
11.15	Appendix: Financial Analysis of the Different Configurations of Combined-Cycle Plants	142
12	Wind Power Turbine Generators—Brushless Double-Feed Generators	167
12.1	Introduction	167
12.2	Basic System Configuration	168
12.3	Equivalent Circuit for the Brushless Double-Fed Machine ...	169
12.4	Parameter Extraction	171
12.5	Generator Operation	171
12.6	Converter Rating	172
12.7	Machine Control	174
12.8	Conclusions	174
12.9	Bibliography	174
13	Gas Laws and Compression Principles	175
13.1	Introduction	175
13.2	Symbols	175
13.2.1	Compressor Operation	175
13.3	First Law of Thermodynamics	179
13.4	Second Law of Thermodynamics	179
13.4.1	Ideal or Perfect Gas Laws	179
13.4.2	Property Relationships	182
13.4.3	Vapor Pressure	185
13.4.4	Partial Pressures	186
13.4.5	Critical Conditions	187
13.4.6	Gas Mixtures	187

13.4.7	The Mole	187
13.4.8	Volume Percent of Constituents	188
13.4.9	Molecular Weight of a Mixture	188
13.4.10	Specific Gravity and Partial Pressure	188
13.4.11	Specific Heats	189
13.4.12	Pseudo-Critical Conditions and Compressibility	190
13.4.13	Weight-Basis Item	191
13.4.14	Compression Cycles	191
13.4.15	Compressor Polytropic Efficiency	193
13.4.16	Compressor Power Requirement	194
13.4.17	Compressibility Correction	195
13.4.18	Multiple Staging	196
13.4.19	Compressor Volumetric Flow Rate	197
13.4.20	Cylinder Clearance and Volumetric Efficiency	198
13.4.21	Cylinder Clearance and Compression Efficiency	201
13.5	Bibliography	201
13.6	Appendix: List of Symbols	201
14	Compressor Types and Applications	203
14.1	Introduction	203
14.2	Positive Displacement Compressors	204
14.2.1	Rotary Compressors	204
14.2.2	Reciprocating Compressors	208
14.3	Dynamic Compressors	210
14.3.1	Centrifugal Compressors	210
14.3.2	Axial Flow Compressors	217
14.4	Bibliography	218
15	Compressors	219
15.1	Compressor Types	219
15.2	Compressor Operation	219
15.3	Gas Laws	219
15.4	Compressor Performance Measurement	220
15.4.1	Inlet Conditions	222
15.4.2	Compressor Performance	222
15.4.3	Energy Available for Recovery	222
15.4.4	Positive Displacement Compressors	223
15.4.5	Reciprocating Compressors	223
15.4.6	Trunk Piston Compressors	224
15.4.7	Sliding Crosshead Piston Compressors	225
15.4.8	Diaphragm Compressors	226
15.4.9	Bellows Compressors	227
15.4.10	Rotary Compressors	227
15.4.11	Rotary Screw Compressors	228
15.4.12	Lobe-Type Air Compressors	229
15.4.13	Sliding Vane Compressors	230
15.4.14	Liquid Ring Compressors	230

15.4.15	Dynamic Compressors	231
15.4.16	Centrifugal Compressors	231
15.4.17	Axial Compressors	232
15.4.18	Air Receivers	233
15.5	Compressor Control	233
15.6	Compressor Unloading System	233
15.7	Intercooler and Aftercoolers	234
15.8	Filters and Air Intake Screens	235
15.9	Preventive Maintenance and Housekeeping	235
15.10	Bibliography	236
16	Performance of Positive Displacement Compressors	237
16.1	Compressor Performance	237
16.1.1	Positive Displacement Compressors	237
16.1.2	Reciprocating Compressor Rating	237
16.1.3	Reciprocating Compressor Sizing	237
16.1.4	Capacity Control	240
16.1.5	Compressor Performance	246
16.2	Reciprocating Compressors	246
16.2.1	Compressor Valves	246
16.2.2	Reciprocating Compressors Leakage	248
16.2.3	Screw Compressors Leakage	249
16.3	Bibliography	251
17	Reciprocating Compressors	253
17.1	Introduction	253
17.2	Crankshaft Design	256
17.3	Bearings and Lubrication Systems	259
17.4	Connecting Rods	262
17.5	Crossheads	263
17.6	Frames and Cylinders	264
17.7	Compressor Cooling	269
17.8	Pistons	272
17.9	Piston and Rider Rings	272
17.10	Valves	274
17.11	Piston Rods	280
17.12	Packings	281
17.13	Cylinder Lubrication	282
17.14	Distance Pieces	282
17.15	Bibliography	286
18	Reciprocating Air Compressors Troubleshooting and Maintenance	287
18.1	Introduction	287
18.2	Location	287
18.3	Foundation	287
18.4	Air Filters and Suction Lines	289
18.5	Air Receiver Location and Capacity	289

18.6	Starting a New Compressor	290
18.7	Lubrication	292
18.8	Non-Lubricated Cylinders	293
18.9	Valves	293
18.10	Piston Rings	296
18.11	Intercoolers and Aftercoolers	296
18.12	Cleaning	297
18.13	Packing	297
18.14	Bibliography	300
19	Diaphragm Compressors	301
19.1	Introduction	301
19.2	Theory of Operation	301
19.3	Compressor Design	304
19.4	Materials of Construction	308
19.5	Accessories	309
19.6	Cleaning and Testing	310
19.7	Applications	311
19.7.1	Automotive Air Bag Filling	312
19.7.2	Petrochemical Industries	312
19.8	Limitations	312
19.9	Installation and Maintenance	312
19.10	Diaphragm Compressor Specification	315
19.11	Bibliography	315
20	Rotary Screw Compressors and Filter Separators	317
20.1	Twin-Screw Machines	317
20.1.1	Compressor Operation	317
20.1.2	Applications of Rotary Screw Compressors	318
20.1.3	Dry and Liquid Injected Compressors	321
20.1.4	Operating Principles	322
20.1.5	Flow Calculation	324
20.1.6	Power Calculation	324
20.1.7	Temperature Rise	328
20.1.8	Capacity Control	328
20.1.9	Mechanical Construction	332
20.1.10	Industry Experience	334
20.1.11	Maintenance History	337
20.1.12	Performance Summary	338
20.2	Oil-Flooded Single-Screw Compressors	339
20.3	Selection of Modern Reverse-Flow Filter Separators	343
20.3.1	Conventional Filter Separators and Self-Cleaning Coalescers	343
20.3.2	Removal Efficiencies	344
20.3.3	Filter Quality	344
20.3.4	Selection of the Most Suitable Gas Filtration Equipment	345

20.3.5	Evaluation of the Proposed Filtration Configurations	346
20.3.6	Life-Cycle-Cost Calculations	346
20.4	Conclusions	347
20.5	Bibliography	347
20.6	Appendix: Coke Fuel	348
20.6.1	Introduction	348
20.6.2	Properties and Usage	348
20.6.3	Other Coking Processes	348
20.6.4	Bibliography	348
21	Straight Lobe Compressors	349
21.1	Applications	349
21.1.1	Operating Characteristics	349
21.2	Operating Principle	349
21.3	Pulsation Characteristics	351
21.4	Noise Characteristics	351
21.5	Torque Characteristics	352
21.6	Construction	352
21.6.1	Rotors	352
21.6.2	Casing	352
21.6.3	Timing Gears	352
21.6.4	Bearings	353
21.7	Staging	353
21.7.1	Higher Compression Ratios	353
21.7.2	Power Reduction	353
21.8	Installation	354
21.9	Bibliography	355
22	Recent Developments in Separating Liquid from Gases	357
22.1	Introduction	357
22.2	Removal Mechanisms	358
22.3	Liquid/Gas Separation Technologies	359
22.3.1	Gravity Separators	359
22.3.2	Centrifugal Separators	359
22.3.3	Mist Eliminators	359
22.3.4	Filter Vane Separators	359
22.3.5	Liquid/Gas Coalescers	359
22.3.6	Selection of Liquid/Gas Separation Equipment	361
22.4	Formation of Fine Aerosols	361
22.5	Ratings and Sizing of Separation Equipment	361
22.6	Bibliography	363
23	Dynamic Compressors Technology	365
23.1	Introduction	365
23.2	Centrifugal Compressor Overview	365
23.3	Axial Compressors Overview	368
23.4	Bibliography	371

24	Simplified Equations for Determining the Performance of Dynamic Compressors	373
24.1	Nonoverloading Characteristics of Centrifugal Compressors	373
24.2	Stability	373
24.3	Speedy Change	373
24.4	Compressor Drive	375
24.5	Calculations	376
24.6	Bibliography	379
25	Centrifugal Compressors—Components, Performance Characteristics, Balancing, Surge Prevention Systems, and Testing	381
25.1	Introduction	381
25.2	Casing Configuration	381
25.3	Construction Features	381
25.3.1	Diaphragms	383
25.3.2	Interstage Seals	391
25.3.3	Balance Piston Seal	395
25.3.4	Impeller Thrust	396
25.4	Performance Characteristics	396
25.4.1	Slope of the Centrifugal Compressor Head Curve	397
25.4.2	Stonewall	399
25.4.3	Surge	401
25.4.4	Off-Design Operation	404
25.5	Rotor Dynamics	406
25.6	Rotor Balancing	406
25.7	Surge Prevention Systems	408
25.8	Surge Identification	412
25.9	Liquid Entrainment	412
25.10	Instrumentation	413
25.11	Cleaning Centrifugal Compressors	413
25.12	Bibliography	415
25.13	Appendix: Boundary Layer	415
25.13.1	Definition	415
25.13.2	Description of the Boundary Layer	415
25.13.3	Separation: Wake	416
25.13.4	Bibliography	417
26	Compressor Auxiliaries, Off-Design Performance, Stall, and Surge	419
26.1	Introduction	419
26.2	Compressor Auxiliaries	419
26.3	Compressor Off-Design Performance	419
26.3.1	Low Rotational Speeds	422
26.3.2	High Rotational Speeds	423
26.4	Performance Degradation	423
26.5	Bibliography	424

27	Dynamic Compressors Performance		425
27.1	Description of a Centrifugal Compressor		425
27.2	Centrifugal Compressor Types		430
27.2.1	Compressors with Horizontally Split Casings		430
27.2.2	Centrifugal Compressors with Vertically Split Casings		430
27.2.3	Compressors with Bell Casings		433
27.2.4	Pipeline Compressors		433
27.2.5	SR Compressors		434
27.3	Performance Limitations		435
27.3.1	Surge Limit		436
27.3.2	Stonewall		438
27.3.3	Prevention of Surge		438
27.3.4	Anti-Surge Control Systems		438
27.4	Bibliography		440
28	Compressor Seal Systems		441
28.1	Introduction		441
28.2	The Supply System		441
28.3	The Seal Housing System		442
28.4	The Atmospheric Draining System		443
28.5	The Seal Leakage System		443
28.6	Gas Seals		445
28.7	Liquid Seals		446
28.8	Liquid Bushing Seals		446
28.9	Contact Seals		448
28.10	Restricted Bushing Seals		449
28.11	Seal Supply Systems		450
28.11.1	Flow Through the Gas Side Contact Seal		451
28.11.2	Flow Through the Atmospheric Side Bushing Seal ..		451
28.11.3	Flow Through the Seal Chamber		452
28.12	Seal Liquid Leakage System		452
28.13	Bibliography		452
29	Dry Seals, Advanced Sealing Mechanisms, and Magnetic Bearings ...		453
29.1	Introduction		453
29.2	Background		453
29.3	Dry Seals		454
29.3.1	Operating Principles		454
29.3.2	Operating Experience		457
29.3.3	Problems and Solutions		457
29.3.4	Upgrade Developments of Dry Seals		458
29.3.5	Prevention of Dry Gas Seal Failures by Gas Conditioning		459
29.4	Magnetic Bearings		461
29.4.1	Operating Principles		461
29.4.2	Operating Experience and Benefits		464

29.4.3	Problems and Solutions	465
29.4.4	Development Efforts	465
29.5	Thrust-Reducing Seals	465
29.6	Integrated Design	467
29.7	Bibliography	471
30	Compressor System Calculations	473
30.1	Calculations of Air Leaks from Compressed-Air Systems	473
30.1.1	Annual Cost of Air Leakage	474
30.2	Centrifugal Compressor Power Requirement	475
30.2.1	Compressor Selection	476
30.2.2	Selection of Compressor Drive	479
30.2.3	Selection of Air Distribution System	481
30.2.4	Water Cooling Requirements for Compressors	481
30.2.5	Variation of Compressor Delivery with Inlet Air Temperature	481
30.2.6	Sizing of Compressor System Components	482
30.2.7	Calculation of Receiver Pump-Up Time	483
30.3	Bibliography	484
31	Pumps	485
31.1	Introduction	485
31.2	Centrifugal Pumps	485
31.2.1	Theory of Operation of a Centrifugal Pump	487
31.2.2	Casings and Diffusers	489
31.2.3	Radial Thrust	492
31.2.4	Hydrostatic Pressure Tests	495
31.2.5	Impeller	496
31.2.6	Axial Thrust	500
31.2.7	Axial Thrust in Multistage Pumps	500
31.2.8	Hydraulic Balancing Devices	501
31.3	Mechanical Seals	504
31.4	Bearings	504
31.5	Couplings	504
31.6	Bedplates	504
31.7	Minimum Flow Requirement	506
31.8	Centrifugal Pumps: General Performance Characteristics	506
31.9	Cavitation	508
31.10	Net Positive Suction Head	509
31.11	Maintenance Recommended on Centrifugal Pumps	510
31.12	Recommended Pump Maintenance	511
31.13	Vibration Analysis	513
31.14	Bibliography	515
32	Centrifugal Pump Mechanical Seal	517
32.1	Introduction	517
32.2	Basic Components	517

32.2.1	Seal Balance	518
32.2.2	Face Pressure	519
32.2.3	Pressure-Velocity	521
32.2.4	Power Consumption	521
32.2.5	Temperature Control	522
32.2.6	Seal Lubrication/Leakage	527
33	Positive Displacement Pumps	555
33.1	Reciprocating Pumps	555
33.1.1	Piston Pumps	555
33.1.2	Plunger Pumps	557
33.1.3	Diaphragm Pumps	560
33.2	Rotary Pumps	562
33.2.1	Gear Pumps	562
33.2.2	Screw Pumps	563
33.2.3	Two- or Three-Lobe Pumps	564
33.2.4	Cam Pumps	564
33.2.5	Vane Pumps	565
33.3	Bibliography	570
34	Diaphragm Pumps	571
34.1	Introduction	571
34.2	Mechanically Driven Diaphragm Pumps	571
34.3	Hydraulically Actuated Diaphragm Pumps	573
34.4	Pneumatically Powered Diaphragm Pumps	573
34.5	Materials of Construction	576
34.5.1	Advantages and Limitations	577
34.5.2	Limitations of Diaphragm Pumps	577
34.5.3	Advantages of Diaphragm Pumps	578
34.6	Bibliography	578
35	Canned Motor Pumps	579
35.1	Canned Motor Pumps Design and Applications	579
35.2	Seal-Less Pump Motors	582
35.3	Bibliography	582
36	Troubleshooting of Pumps	583
36.1	Pump Maintenance	583
36.1.1	Daily Observations of Pump Operation	583
36.1.2	Semiannual Inspection	583
36.1.3	Annual Inspection	584
36.1.4	Complete Overhaul	584
36.1.5	Spare and Repair Parts	585
36.1.6	Record of Inspections and Repairs	585
36.1.7	Diagnoses of Pump Troubles	586
36.2	Troubleshooting of Centrifugal Pumps	586
36.3	Troubleshooting of Rotary Pumps	586

36.4	Troubleshooting of Reciprocating Pumps	586
36.5	Troubleshooting of Steam Pumps	586
36.6	Vibration Diagnostics	588
36.6.1	Analysis Symptoms	588
36.6.2	Impeller Unbalance	601
36.6.3	Hydraulic Unbalance	602
36.7	Bibliography	602
37	Water Hammer	603
37.1	Introduction	603
37.2	Nomenclature	603
37.3	Basic Assumptions	604
37.4	Effects of Water Hammer in High- and Low-Head Pumping Systems	605
37.4.1	Magnitude of the Pulse	605
37.4.2	Possible Causes of Water Hammer	606
37.4.3	Mitigating Measures to Water Hammer	606
37.4.4	Applications of Water Hammer	606
37.5	Power Failure at Pump Motors	607
37.5.1	Pumps with No Valves at the Pump	607
37.5.2	Pumps Equipped with Check Valves	610
37.5.3	Controlled Valve Closure	612
37.5.4	Surge Suppressors	612
37.5.5	Water Column Separation	613
37.5.6	Quick-Opening, Slow-Closing Valves	613
37.5.7	One-Way Surge Tanks	613
37.5.8	Air Chambers	614
37.5.9	Surge Tanks	614
37.5.10	Nonreverse Ratchets	614
37.6	Normal Pump Shutdown	615
37.7	Water Hammer Example	615
37.8	Steam Hammer	617
37.9	Bibliography	617
38	Selection and Procurement of Pumps	619
38.1	Introduction	619
38.2	Engineering of System Requirements	619
38.2.1	Fluid Type	619
38.2.2	System-Head Curves	619
38.3	Alternate Modes of Operation	620
38.4	Margins	620
38.5	Wear	621
38.6	Future System Changes	621
38.7	Selection of Pump and Driver	621
38.7.1	Pump Characteristics	622
38.7.2	Code Requirements	622

38.7.3	Fluid Characteristics	622
38.7.4	Pump Materials	623
38.7.5	Driver Type	623
38.8	Pump Specifications	623
38.8.1	Specification Types	623
38.8.2	Data Sheet	624
38.8.3	Codes and Standards	624
38.8.4	Bidding Documents	626
38.8.5	Technical Specification	626
38.8.6	Commercial Terms	627
38.9	Special Considerations	628
38.9.1	Performance Testing	628
38.9.2	Pump Drivers	628
38.9.3	Special Control Requirements	629
38.9.4	Drawing and Data Requirements Form	629
38.9.5	Quality Assurance and Quality Control	629
38.10	Bidding and Negotiation	630
38.10.1	Public and Private Sector	630
38.10.2	Bid List	631
38.10.3	Evaluation of Bids	631
38.10.4	Cost	631
38.10.5	Efficiency	631
38.10.6	Economic Life	631
38.10.7	Spare Parts	631
38.10.8	Guarantee/Warranty	631
38.10.9	Sample Bid Evaluation	632
38.11	Bibliography	635
39	Pumping System Calculations	637
39.1	Analysis of Pumps Installed in Series	637
39.2	Analysis of Pumps Installed in Parallel	637
39.3	Selection of Pump Driver Speed	642
39.4	Affinity Laws for Centrifugal Pumps	643
39.5	Centrifugal Pump Selection Using Similarity or Affinity Laws	644
39.6	Determination of Centrifugal Pump Capacity and Efficiency	646
39.7	Selection of the Best Operating Speed for a Centrifugal Pump ..	648
39.8	Calculate the Total Head of the Pump	650
39.9	Pump Selection Procedure	655
39.9.1	Draw the Proposed Piping Layout of the Pumping System	655
39.9.2	Determine the Required Pump Capacity	655
39.9.3	Determine the Total Head on the Pump	656
39.9.4	Obtain the Physical and Chemical Data of the Liquid Being Pumped	656

39.9.5	Select the Category and Type of Pump	657
39.9.6	Evaluate the Selected Pump	660
39.10	Bibliography	662
40	Bearings	663
40.1	Types of Bearings	663
40.1.1	Ball and Roller Bearings	663
40.2	Stresses During Rolling Contact	664
40.3	Statistical Nature of Bearing Life	665
40.4	Materials and Finish	666
40.5	Sizes of Bearings	666
40.6	Types of Rolling Bearings	666
40.6.1	Thrust Bearings	669
41	Lubrication	671
41.1	The Viscosity of Lubricants	671
41.1.1	Viscosity Units	671
41.1.2	Significance of Viscosity	672
41.1.3	Flow Through Pipes	673
41.2	Variation of Viscosity with Temperature and Pressure	674
41.2.1	Temperature Effect	674
41.2.2	Viscosity Index	674
41.2.3	Effect of Pressure on Viscosity	674
41.3	Non-Newtonian Fluids	674
41.3.1	Greases	674
41.3.2	VI-Improved Oils	674
41.3.3	Oils at Low Temperatures	675
41.4	Variation of Lubricant Viscosity with Use	675
41.4.1	Oxidation Reactions	675
41.4.2	Physical Reactions	675
41.5	Housing and Lubrication	675
41.6	Lubrication of Antifriction Bearings	677
41.7	Bibliography	679
42	Used Oil Analysis—A Vital Part of Maintenance	681
42.1	Proper Lube Oil Sampling Technique	681
42.1.1	Test Description and Significance	681
42.1.2	Visual and Sensory Inspections	681
42.1.3	Chemical and Physical Tests	683
42.2	Summary	689
42.3	Bibliography	689
43	Vibration Analysis	691
43.1	The Application of Sine Waves to Vibration	691
43.1.1	Multimass Systems	693
43.1.2	Resonance	693
43.1.3	Logarithms and Decibels	695

- 43.1.4 The Use of Filtering 695
 - 43.1.5 Vibration Instrumentation 695
 - 43.1.6 Transducer Selection 697
 - 43.1.7 Machinery Example 699
 - 43.1.8 Vibration Analysis 699
 - 43.1.9 Vibration Causes 699
 - 43.1.10 Forcing Frequency Causes 699
 - 43.1.11 Vibration Severity 702
- 43.2 Appendix: A Case History (Condensate Pump Misalignment) ... 703
 - 43.2.1 Problem 703
 - 43.2.2 Test Data and Observations 703
 - 43.2.3 Corrective Actions 703
 - 43.2.4 Final Results 703
 - 43.2.5 Conclusion 703
- Index 707**

Preface

Power Plant Equipment Operation and Maintenance Guide provides detailed coverage of different types of power plants, including the following: modern cogeneration, combined cycle,* and integrated gasification combined cycle (IGCC; plants that burn coal, petroleum coke, orimulsion, biomass, and municipal waste).

This book provides the reader with a comprehensive understanding of all the equipment used in different type of power plants. It includes detailed coverage of the sizing, selection considerations, calculations, operation, diagnostic testing, trouble-shooting, maintenance, and refurbishment of all types of steam turbines, steam generators, boilers, condensers, heat exchangers, gas turbines, compressors, pumps, advanced sealing mechanisms, magnetic bearings, and advanced generators. All the systems and methods used to maximize the efficiency, reliability, and longevity of this equipment are covered in detail. All the preventive and predictive maintenance required for this equipment, including vibration analysis and used oil analyses, are also covered thoroughly. The book also provides guidelines and rules that must be followed to minimize the failure rate and downtime of all power plant equipment.

This book includes an in-depth description of the economics of these power plants and provides information that assists in the selection of the best available power enhancement options such as duct burners, evaporative cooling, inlet-air chilling, absorption chilling, steam and water injection, and peak firing.

Deregulation of the electricity markets is sweeping across the world, which is increasing the necessity for highly efficient power generating plants, such as cogeneration, combined cycle, and IGCC. *Changes in the market and the many advantages of these types of power plants enhance their ability to compete and eventually replace older model plants.* It is anticipated that independent power producers and different utilities plan to construct additional cogeneration, combined-cycle, and IGCC plants in the near future due to their short construction lead time, low staffing requirements, as well as, low capital investment, and operating and maintenance costs.

This book covers in detail the economic analysis of modern cogeneration and combined-cycle plants, and all the features that enhance their reliability, maintainability, and all the preventive and predictive maintenance required for these power plants. Moreover, this book will focus on the application of gas turbines in power plants, protective systems, tests, and modern brushless double-feed generators (BDFG;

*A combined-cycle power plant consists of gas turbines operating in conjunction with a steam power plant.

generators that enable power generation over a wide range of rotor speeds and that also eliminate the need for brushgear, ideal for wind turbine applications). This book places a special emphasis on all types of compressors and pumps used in gas turbines, cogeneration, and combined-cycle plants, including centrifugal, axial-flow, and reciprocating compressors and centrifugal and positive displacement pumps.

The failure of equipment and systems causes extended outages for power plants and significant losses. This book covers all the common problems with these power plants equipment, including compressor surge and choking, pump cavitation, and water hammer. In addition, the book also covers thoroughly all the surge protection and choking protection systems of centrifugal and axial-flow compressors, mechanical seals, dry seals, advanced sealing systems, and magnetic bearings. This information is essential to minimize failure in the equipment used in different types of power plants and to achieve reduced capital, operating and maintenance costs, along with increase in efficiency and profitability.

This book illustrates in detail the reasons why cogeneration, combined-cycle plants, and IGCC are the most economic method of power generation. The power generated by these plants is 7 to 10 times cheaper than that generated by green-power (wind and solar) plants. Their capital and operation and maintenance costs are a fraction of nuclear power plants. These plants are also more than twice as efficient as conventional fossil and nuclear plants (i.e., they burn less than half the fuel to generate the same amount of power). These plants are also environmentally friendly. The emission of nitrogen oxides, NO_x (NO and NO_2), is less than 3 ppmv (parts per million in volume). They do not emit any sulfur oxides, SO_x (SO_2 and SO_3), when they burn gaseous fuels. They also do not have the potential safety hazards associated with nuclear power plants. Another important advantage is that these plants can be located close to the power user eliminating the need for long transmission systems, reducing transmission losses, and increasing reliability.

The IGCC plants use a process called gasification to convert low-value fuel such as coal, petroleum coke, orimulsion, biomass, or municipal waste to high-hydrogen gas known as synthetic gas, or syngas. The syngas generated by this process is used as the primary fuel for gas turbines after it has been cleaned. The cleanup process involves normally the removal of sulfur compounds, ammonia, metals alkalytes, ash, and particulates. Methanol, ammonia, fertilizers, and other chemicals can be made from the compounds removed from the syngas. These marketable by-products also improve the economics of the IGCC.

The IGCC plants have achieved exceptional levels in environmental performance, availability, and efficiency at a competitive cost of electricity. The IGCC plant has a similar physical size to a conventional coal-fired power boiler. However, additional space is required for a conventional coal plant for scrubber sludge treatment or ash dewatering that IGCC plants do not require. The SO_x (SO_2 and SO_3), NO_x (NO and NO_2), and particle emissions of an IGCC plant are fractions of those of a conventional coal power plant. Thus, less effort and time are required to obtain local and government environmental permits to build IGCC plants.

Environmental agencies have classified IGCC plants as the best environmental solution to generate power from coal. In an IGCC plant, the harmful pollutants are removed from the synthetic gas; thus cleanup of the exhaust gas from the gas turbine is not necessary. Most of the mercury is removed at a very low cost in IGCC plants because there is no necessity to implement mercury removal systems for the back-end flue gas

of an IGCC gas turbine as activated carbon bed filters the syngas. In addition, most of the mercury is removed by recycled water streams at a minimal cost. The carbon is removed from the syngas in IGCC plants in order to create a high-hydrogen fuel that, as a result, eliminates the carbon dioxide emissions. Carbon is removed from the exhaust gas (after combustion) in conventional boiler plants. This process is about 10 times more expensive due to the larger volume of the post-combustion gas. The reliability, availability, and maintenance (RAM) of an IGCC plant are similar to those of natural gas combined-cycle plants.

The benefits of cogeneration, combined-cycle, and IGCC plants are enormous as they have a 4- to 5-year payback period due to their high rate of return (the rate of return can exceed 35% in some applications—see Chap. 11 for details). The output of an IGCC plant varies from 10 MW to more than 1.5 GW. They produce electricity at a lower cost than conventional solid fuel plants. Finally, these plants can be applied in any new or repowering project.

These plants also have the advantage of long-term fuel price stability and fuel flexibility (they can burn any type of gaseous or liquid fuel, including heavy-liquid fuels such as Type 6). Moreover, these plants have a much higher capacity factor* than any fossil or nuclear power plant.

Finally, there is no question that a shake-up in the electricity market is in the forecast and the competitive edge of cogeneration, combined-cycle, and IGCC plants provides them with a promising future.

This book was written for all technical individuals, including engineers of all disciplines, managers, technicians, operators, and maintenance personnel. All the concepts are presented in a simple and practical manner that allows the reader to understand them without relying on advanced mathematical equations. This book is a must for any individual who has an interest in power generation or the equipment used in power plants, and would like to gain an in-depth understanding of modern types of power plants, their equipment, and their numerous advantages.

I sincerely hope that this book will be as interesting to read as it was for me to write and that it will be a useful reference to anyone interested in power generation.

Philip Kiamleh, M.A.Sc., B.Eng., D.Eng., P.Eng.

*The capacity factor of a power plant is the ratio of the actual output of a power plant over a period of time and its potential output if it had operated at full nameplate capacity the entire time. To calculate the capacity factor, take the total amount of energy the plant produced during a period of time and divide by the amount of energy the plant would have produced at full capacity.

This page intentionally left blank

Acknowledgments

1. Reprinted with permission from Dresser-Rand:
Figs. 13.1–13.6, 13.8, 13.10–13.14, 16.4, 16.6, 16.8, 16.10, 17.2–17.4, 17.7–17.26, 17.29, 17.30, 17.33–17.36, 17.41–17.46, 17.48–17.50, 20.19–20.23
2. Reprinted with permission from Pall Corporation:
Figs. 22.1–22.4
3. Reprinted with permission from Flowserve:
Figs. 31.4, 31.6–31.9, 31.11–31.14, 31.18–31.25, 31.29, 31.35–31.37, 31.39
4. Reprinted with permission from SKF:
Figs. 31.34, 41.3
5. Reprinted with permission from Babcock and Wilcox Company:
Figs. 6.12, 6.14, 6.16
6. Reprinted with permission from Gorman-Rupp Company:
Fig. 34.2
7. Reprinted with permission from General Electric:
Figs. 2.1–2.12, 3.1–3.12, 5.1–5.14, 8.1, 9.1–9.10
8. Reprinted with permission from Anglo Compression, Inc.:
Figs. 17.37, 17.38
9. Reprinted with permission from Aerzen:
Figs. 20.1–20.18
10. Reprinted with permission from Hydraulic Institute, Inc.:
Figs. 39.8, 39.11
11. Reprinted with permission from Teikoku:
Fig. 35.1
12. Reprinted with permission from *Power* magazine:
Figs. 10.1–10.3, 30.2
13. Reprinted with permission from McMahon, R. A., Wang, X., and Abdi-Jalebi, E., *The BDFM as a Generator in Wind Turbines*, Cambridge University, Trumpington Street, Cambridge CB2 1PZ, UK, 2007:
Figs. 12.1–12.9
14. Reprinted with permission from Cucuz, M., *The Economics of Combined Cycle Cogeneration Plants*, Westinghouse Power Manufacturing Division, Hamilton, 1997:
Figs. 11.1–11.6

15. Reprinted with permission from John Crane:
Figs. 28.4, 32.4–32.6, 32.8, 32.9, 32.13, 32.18, 32.20, 32.25, 32.28
16. Reprinted with permission from Aetna Bearing Company:
Fig. 40.10
17. Reprinted with permission from Compressor Control Corp.:
Figs. 27.18–27.20
18. Reprinted with permission from Chevron:
Figs. 42.1–42.11
19. Reprinted with permission from National Resources Canada:
Figs. 15.1–15.19
20. Reprinted with permission from Burckhardt:
Figs. 17.1, 17.6, 17.28
21. Reprinted with permission from Cameron Corporation:
Figs. 16.2, 17.5, 17.27, 17.39, 17.40, 17.43
22. Reprinted with permission from IDEX Chemical, Food & Pharma:
Fig. 34.4
23. Reprinted with permission from National Oilwell Varco:
Fig. 33.8
24. Reprinted with permission from Tubular Exchanger Manufacturers Association:
Fig. 7.12
25. Reprinted with permission from Gardner Denver Nash LLC:
Fig. 14.6
26. Reprinted with permission from Sundstrand Corp.:
Fig. 14.12
27. Reprinted with permission from Revolve Technology, Calgary, Alberta:
Figs. 29.1–29.15
28. Reprinted with permission from Elliott Group, Ebara Corporation:
Figs. 23.2, 25.2–25.8, 28.1, 28.5, 28.7
29. Reprinted with permission from MAN Diesel & Turbo SE:
Figs. 23.5, 23.7, 23.8, 29.16–29.20
30. Reprinted from El-Wakil, M. M., *Power Plant Technology*, McGraw-Hill, New York, 1984, with permission from McGraw-Hill:
Figs. 13.7, 13.9
31. Reprinted with permission from Higgins, L., *Maintenance Engineering Handbook*, 5th ed., McGraw-Hill, New York, 1995, with permission from McGraw-Hill:
Figs. 18.1–18.6
32. Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill:
Figs. 31.1, 31.3, 31.4, 31.15, 31.27, 31.30–31.32, 31.41–31.43, 34.3, 34.7, 34.8, 37.1–37.8
33. Reprinted from Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006, with permission from McGraw-Hill:
Figs. 30.3, 30.4, 39.1–39.6, 39.10, 39.13–39.17

Gas Turbine Applications in Power Stations, Gas Turbine Protective Systems, and Tests

1.1 Introduction

Gas turbines are used to supply power for the following purposes:

- *Peak shaving*: This situation occurs when the load demand is high and the steam power stations are unable to generate additional power.
- Starting a power station when the grid supply is not available.
- Following the tripping of a power station from load when the grid supply is not available. Gas turbines provide power in this situation to provide lubrication and cooling to hot station equipment to prevent damage.

The following are the requirements for the gas turbine plant in power stations:

- Output higher than 10 MW. This is the power required to start a large generator (more than 600 MW)
- High availability (more than 99% in some applications)
- High flexibility
- Minimum maintenance cost

Two or three aeroderivative engines are used for this gas turbine plant. The output of each engine is between 10 and 25 MW. Some applications use gas generators (the portion of the gas turbine which generates the hot gases—see Fig. 1.1) that discharge their gases into a turbine. This turbine is known as “free power turbine.” The generator is coupled directly to this turbine.

Modern gas turbine power stations use a single 70-MW electric generator (Fig. 1.2). This electric generator is coupled to two free power turbines at each end. Each free

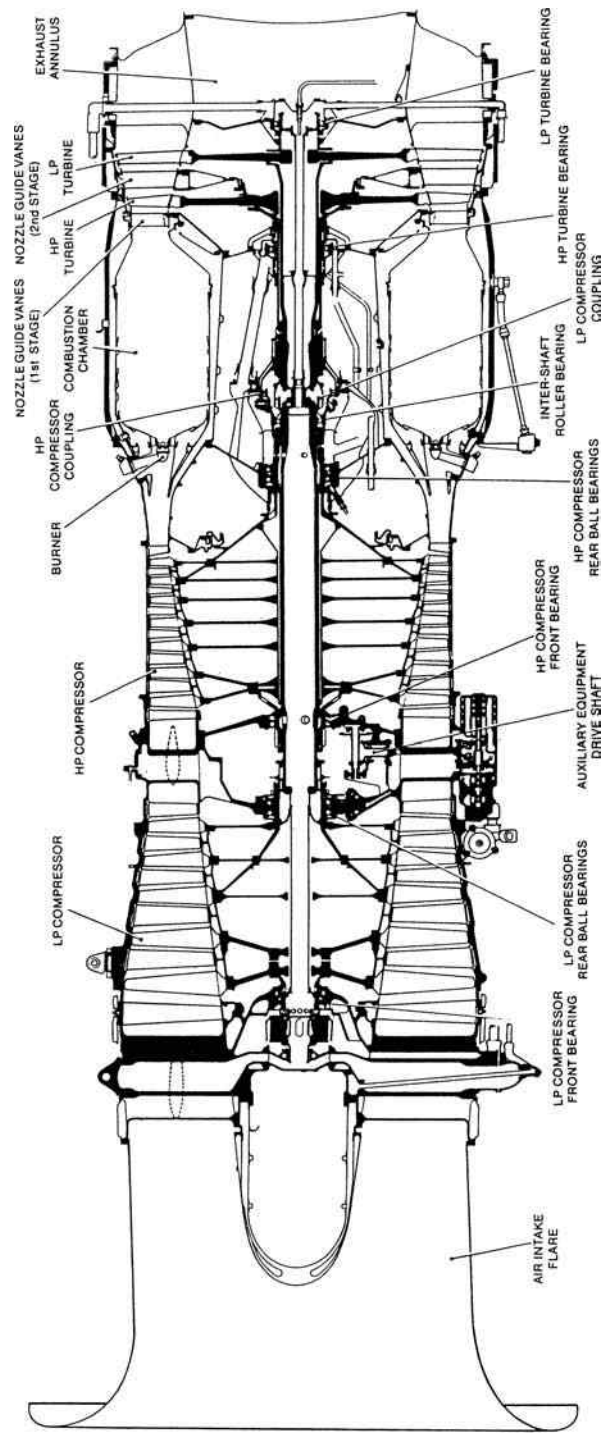


Figure 1.1 The Olympus gas generator.

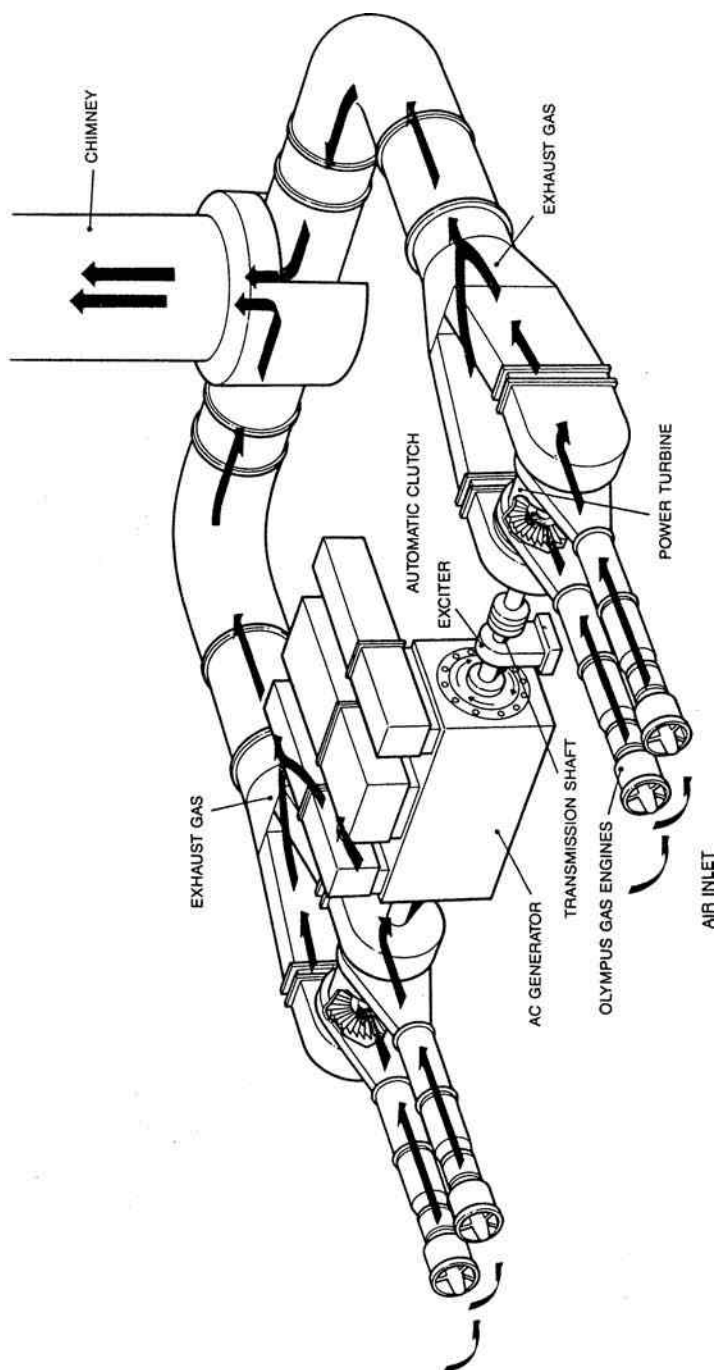


FIGURE 1.2 Current design of gas turbine plant.

power turbine is driven by a gas generator similar to the one shown in Fig. 1.1. This configuration allows the electric generator to operate at part load. This is done by varying the following:

- Number of engines (gas generators) in service
- The gas flow generated by each engine

1.2 Working Cycle

Figure 1.3 illustrates the working cycle of the gas turbine. The gas volume (or velocity) increases in the combustion chambers (known as combustors). However, the combustion occurs at constant pressure (there is a slight decrease in pressure inside the combustors due to friction). This allows for the use of lightweight combustors and low-octane fuels.

1.2.1 Starting

The gas generator is started by a starter motor. This motor rotates the HP compressor. This induces air into the LP compressor. The induced air forces the LP compressor to rotate. The igniters are energized when the HP compressor speed reaches 800 rpm. The fuel boost pump is started 2 seconds later and the fuel servo-valve opens. The compressor must be able to develop sufficient air pressure to provide air for combustion and cooling the combustion chambers and other components at light-up. This occurs at a speed of 1000 rpm. Fuel is admitted at this speed.

The compressor will reach a speed of 1000 to 2000 rpm within 15 to 20 seconds. The fuel is admitted with the igniters energized. All combustion chambers must be lit within 10 to 20 seconds. The engine exhaust temperature increases rapidly at this stage. The exhaust gas temperature must exceed 200°C within 45 seconds from the beginning of the start sequence. Otherwise, the engine will be shut down. This is done to protect the plant from the hazards of unburnt fuel. The gas flow increases with the speed. The unit is synchronized at the operating speed.

The engine exhaust temperature increases significantly during loading. This temperature is between 450 and 490°C at full load. The loading rate is restricted by the rapid rise in this temperature. The time required to reach full load varies from 2 to 5 minutes.

The fuel/air ratio must be accurately controlled over the full range of operation. This is essential to maintain optimum efficiency and acceptable environmental conditions. Microprocessor-based engine management system is used for this purpose. This system is programmed to match the characteristics of fuel delivery systems and variations in compressor performance. Modern control systems also provide the following:

- Plant condition monitoring
- Alarm scanning
- Data processing
- Fault diagnosis

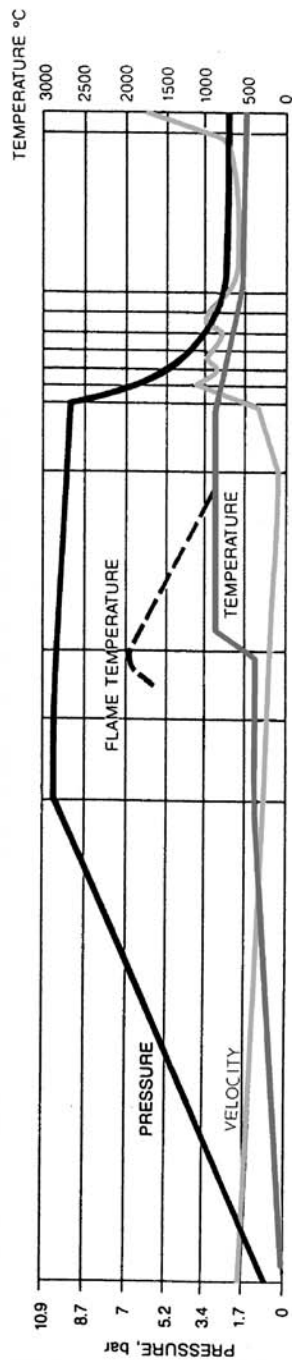
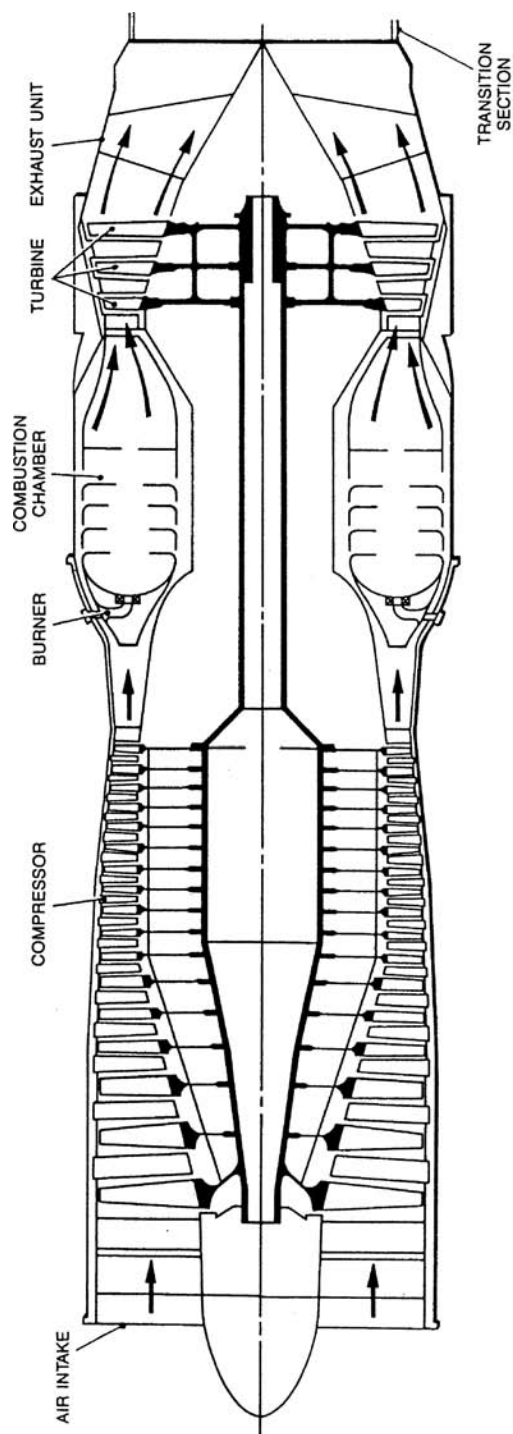


FIGURE 1.3 Working cycle of a gas turbine.

1.2.2 Shutdown

The shutdown process is normally the reverse of the starting process. However, a controlled cooling period is required for the free power turbine in some designs. This period will reduce the thermal shock when the engine is tripped. The hot ducting and stack will generate a natural draught following a trip. This will draw cold air through the engine and power turbine for a period of time. The cold air flow will produce high thermal stresses in the free power turbine. The unit should be operated for 5 minutes at 2 or 3 MW. This will reduce the gas temperatures entering the power turbine to 300 to 325°C. This will reduce the thermal shock on the power turbine significantly. Alternatively, the engine air-intake should be closed following a trip by closing a damper. Steam turbines have a period of shaft barring following shutdown to prevent the thermal stresses.

1.3 Protection

The *high-speed shut-off cock* (HSSOC) is latched open at the start of a run. This spring-operated device admits high-pressure fuel to the combustors. This device is equivalent to the steam turbine emergency stop valve on an aero-engine. The tripping of this device initiates a shutdown under controlled or emergency conditions.

The *low-pressure cock* (LPC) is a second fuel shut-off valve. This device is positioned before the engine fuel pumps and regulator valve. The LPC operates always with the HSSOC. However, the LPC is manually operated for priming the engine fuel system following maintenance.

Gas turbines are housed in acoustic enclosures. These enclosures are strengthened to protect personnel from possible engine explosions. The engines are protected by a fire detection system. Carbon dioxide (CO₂) and halon 1301 have been traditionally used as extinguishing agents for the turbine, auxiliary components, and generator. However, preaction water spray systems have been used for turbine units in buildings. Care must be exercised to prevent water impingement on hot turbine parts. One or more fire detectors must actuate for preaction water spray systems to discharge. Heat must also be present to melt the fusible links in the nozzles. Thus, these systems are less susceptible to inadvertent discharge. The National Fire Protection Association (NFPA) provides standards for the design and installation requirements for fire protection systems.

Gas turbine operators have preferred halon over CO₂ for many years. This is because halon is not an asphyxiant. However, halon breaks down into corrosive and toxic byproducts (hydrogen fluoride, hydrogen bromide, and free bromine) after it has acted under fire or when it is exposed to surface temperatures above 482°C (900°F).

Watermist has proved to be a highly promising agent for the protection of gas turbines. The extinguishing mechanisms of watermist are heat removal, oxygen depletion, and steam expansion. Watermist has the following advantages:

- It is safe for occupied areas.
- Provides better cooling effects than gaseous suppression agents.
- Watermist systems are pre-engineered and approved for a compartment of certain maximum volume (gaseous systems must be engineered for a specific application).
- Watermist systems are not as sensitive to enclosure tightness as gaseous systems.

NFPA 750, Watermist Fire Protection Systems, governs the design, installation, maintenance, and testing of these systems. Several gas turbine installations have used watermist systems successfully. The operation of the fire detectors will trip the unit and close the fire valves. These valves are positioned in the fuel supply line.

The precautions taken during the starting phase of the gas turbine to prevent fuel ignition following a flame-out must continue throughout normal operation. Flame-detection devices (known as flame scanners) have proved to be unreliable at detecting flame-out. The engine exhaust temperature has proved to be the most reliable method for detecting flame-out. The exit-gas temperature is around 250°C for maximum fuel settings. Thus, the engine is assumed to have flame-out when the temperature drops below 200°C. The protection system should trip the engine when the exit-gas temperature drops below 200°C. All engine bearings are sealed with air bled from the compressor. The engine will be tripped if a fault occurs in the sealing system. The engine will also trip on low oil pressure in the bearings and low fuel pressure in the supply line.

1.4 Black Start

Black start is a term used to describe starting a gas turbine from a battery bank when the normal AC electrical supply has been lost. The unit must be able to operate at full-load for sufficient time to reestablish normal AC electrical supplies. The battery bank must have sufficient energy to perform the following functions:

- Operate the starting device (e.g., DC motor) to drive the engine to the self-sustaining speed (this speed at which the turbine can drive the compressor to the operating speed without the aid of the starting device).
- Operate fuel igniters, instruments, and control equipment.
- Operate the turbine-generator lubricating oil pumps and engine fuel boost pumps when fitted.
- 110-V DC battery is normally used for the starting device, control, and instrumentation. A 28-V tapping from this battery is provided for the igniters. The fuel pumps and lubricating oil pumps are supplied from a 240-V DC station battery.

The *Black start* is initiated in the following cases:

- Loss of busbar voltage
- Drop in frequency below a predetermined value

The engine will not be able to synchronize with the busbar when the busbar voltage is lost (the circuit-breaker will not be able to close when the synchronous speed is reached). Thus, the synchronizing interlocks must be overridden if it is essential to close the circuit-breaker automatically. This is the case in nuclear power plants.

The controls are performed manually in fossil-fuel stations when the gas turbine reaches synchronous speed. Station instructions are followed at this stage to restore electrical power. These instructions are consistent with the capacity of the gas turbines

and the anticipated requirements of the plant. The following are the highest priority power supplies of these instructions:

- Restoring battery-charger
- Replenishment of gas turbine fuel stocks
- Turbine-generator lubricating oil pumps
- Generator hydrogen seal oil pump
- Turbine turning gear (barring motor)
- Boiler circulation pumps
- Boiler forced-draft and induced-draft fans (to remove explosive gases from the boilers)

1.5 Routine Tests

Most gas turbine engines operating in power stations are designed to operate at least 2000 service hours between major overhauls. The operating regime in power stations is different from the aviation industry. The following are the main differences:

- The gas turbines operate continuously at ground level in power stations. The air density is higher at high altitudes. This increases the stress on the engine components when operating at full power. The life of these components will be reduced due to this increase in stress. Most aeroderivative engines are derated when operating at ground level. This is done to reduce the stress on these components. This derating is between 20 and 25% of the engine rated capacity.
- The number of engine starts is higher at ground-level operation.

The engine condition is reflected in the following two areas:

1. Bearing vibration levels
2. Debris found in the bearing lubricating oil scavenge filters

The following inspections should be performed at 500-hour intervals:

- Air-intake areas
- Compressor blade conditions
- Fuel drain systems

The protection systems should be tested routinely at intervals ranging from 1 to 6 months. The state of the compressor blades is determined by the following two conditions:

1. Level of impurities (sand, dust, organic materials, etc.) in the ambient air around the engine
2. Filter effectiveness

The compressor-blade cleaning should be performed before significant degradation. The following procedure is recommended for compressor-blade cleaning:

- Inject an approved solvent into the compressor intake while the engine is running at cranking speed. This will loosen the dirt and grease stuck to the compressor blades.
- Allow the solvent to soak for a few hours.
- Spray demineralized water at cranking speed to flush the compressor.
- Run the engine at idle speed for a short period of time. This will dry the engine.

Reliability of these units is very important. The reliability target for some stations is 99%. This indicates that the gas turbine system (2 or 3 gas turbines) must be available to supply power to the station more than 99% of the time. Each gas turbine engine can deliver the station power requirement. The redundancy in the system is known as 2 or 3 $\times$ 100%. Each unit is started weekly. This is done if the engine did not operate during this period.

1.6 Bibliography

British Electricity International, *Modern Power Station Practice—Station Operation and Maintenance*, Vol. G, 3rd ed. Pergamon Press, Oxford, United Kingdom, 1991.

This page intentionally left blank

CHAPTER 2

Steam Turbine Selection for Combined-Cycle Power Systems

2.1 Abstract

A wide variety of steam turbines is available for steam and gas (STAG) combined-cycle plants. Screening tables are presented in this chapter for steam turbine selection. The following application considerations are discussed:

1. Reheat versus non-reheat.
2. Multi-shaft versus single shaft.
3. Axial exhaust versus down exhaust.

2.2 Introduction

Reheat steam cycles are used commonly in STAG combined-cycle plants. These cycles with advanced gas turbine designs are called advanced combined cycles (ACC). Steam cooling in the gas turbine is used in some of these plants. The steam conditions in STAG plants vary up to 12.4 MPa (1800 psig) and 566°C (1050°F).

2.3 Steam Turbine Application to Steam and Gas Plants

2.3.1 Steam and Gas Plants Structure

Table 2.1 lists the gas turbines commonly used in combined-cycle plants. The LM6000 is an aircraft derivative gas turbine. Table 2.2 lists the gross power developed by combined cycles based on the gas turbines listed in Table 2.1. The gas turbine output listed in Table 2.2 differs from the output listed in Table 2.1. This is due to the increase in exhaust pressure drop associated with the heat recovery system generator (HRSG).

Hz	GT Model	Exh Temp °F	Exh Temp °C	Flow K LB/HR	Flow KG/HR	GTG MW	STAG Cycle
60	LM6000	866	463	989.6	44880	40	NRH
60	6B	1002	539	1104.4	500760	39	NRH
60	6FA	1107	597	1591.0	721660	70	RH OR NRH
60	7EA	1002	539	2365.0	1072847	85	NRH
60	7EC	1031	555	2822.0	128006	116	RH OR NRH
60	7FA	1104	596	3509.0	1591680	169	RH OR NRH
50	LM6000	866	463	989.6	44880	40	NRH
50	6B	1002	539	1104.0	500760	39	NRH
50	6FA	1108	598	1587.0	719860	70	RH OR NRH
50	9E	1003	539	3254.0	1476010	123	RH OR NRH
50	9EC	1036	558	4032.0	1828910	169	RH OR NRH
50	9FA	1110	599	5119.0	2322054	240	RH OR NRH

TABLE 2.1 GE Gas Turbine Exhaust Characteristics

2.3.2 Steam Turbine Exhaust Size Selection

Table 2.3 lists the typical last-stage buckets (LSB) for 50 and 60 Hz combined-cycle steam turbines. Several of these buckets are suitable for high back pressure operation.

Figure 2.1 illustrates the effect of steam turbine exhaust area on output for a range of exhaust pressures. These curves indicate that the unit with the largest annulus area have the best performance at low back pressures. The units with smaller exhaust area have better performance at high back pressure. Thus, the selection of steam turbines involves balancing the output against the equipment investment.

2.3.3 Non-Exhaust Cycle-Steam Conditions

The exhaust temperature of gas turbine model 6B, 7EA, and 9E (listed in Table 2.2) is around 538°C (1000°F). This provides a steam temperature around 510°C (950°F). The aero-derivative LM6000 have an exhaust temperature around 484°C (903°F). This provides a steam throttle* temperature around 454°C (850°F).

The throttle pressure is selected based upon the following:

- 1. Size of the steam turbine
- 2. Economic considerations

Higher throttle pressures increase the efficiency of multiple pressure heat recovery steam generator (HRSG) plants.

However, higher pressure has the following effects:

- 1. Decrease in the inlet volumetric flow rate to the steam turbine. This reduces the length of the nozzles (stationary blades) and buckets (moving blades).
- 2. Increase in the stage leakage losses as a fraction of total flow.

*The throttle conditions are the pressure and temperature at the inlet to the turbine.

Frequency (Hz)	GT Model	STAG Model	Steam Cycle	GTG (MW)	STG (MW)	Total (MW)
60	LM6000	260	NRH	75	31	106
60	6B	106B	NRH	38	22	60
60	6B	206B	NRH	76	45	121
60	6B	406B	NRH	152	91	124
60	6FA	106FA	RH	67	40	107
60	6FA	206FA	RH	134	83	217
60	7EA	107EA	NRH	84	46	130
60	7EA	207EA	NRH	166	98	264
60	7EC	107EC	RH	114	66	180
60	7EC	207EC	RH	228	135	363
60	7FA	107FA	RH	166	93	259
60	7FA	207FA	RH	332	190	522
50	LM6000	260	NRH	74	30	104
50	68	106B	NRH	38	22	60
50	6B	206B	NRH	76	45	121
50	6B	406B	NRH	152	90	243
50	6FA	106FA	RH	67	40	107
50	6FA	206FA	RH	134	85	219
50	9E	109E	NRH	123	67	190
50	9E	209E	NRH	245	138	384
50	9EC	109EC	RH	163	96	259
50	9EC	209EC	RH	326	197	523
50	9FA	109FA	RH	240	133	376
50	9FA	209FA	RH	478	280	758

TABLE 2.2 STAG Power Plants—Approximate Output

Thus, the benefits of increased throttle pressure are greater for larger units. Detailed studies of pressure optimization have resulted in the selection of the following pressure:

1. 5.9 MPa (850 psig) for smaller STAG plants with multiple pressure steam cycles.
2. 6.9 MPa (1000 psig) for units in the intermediate range from 40 to 60 MW.
3. 8.6 MPa (1250 psig) for steam turbine ratings greater than 60 MW.

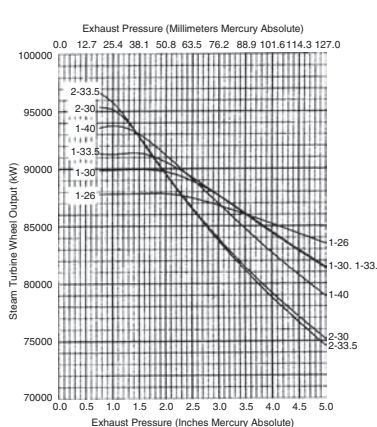
Frequency Hz/rpm	Length		Pitch Diameter		Exhaust Annulus Area for Number of Parallel Flows			
	(in)	(mm)	(in)	(mm)	1 (sq ft)	1 (sq m)	2 (sq ft)	2 (sq m)
60/3600	12.3*	312	52.8	1342	14.2	1.32	–	–
60/3600	14.9	378	58.5	1487	19.0	1.76	–	–
60/3600	17.5	445	54.4	1382	20.8	1.93	–	–
60/3600	13H*	330	68.0	1727	19.3	1.79	–	–
60/3600	20	508	60.0	1524	26.2	2.43	–	–
60/3600	20H*	508	75.0	1905	32.7	3.04	65.4	6.08
60/3600	23	584	65.5	1664	32.9	3.06	65.8	6.11
60/3600	26	660	72.0	1830	41.1	3.82	82.2	7.64
60/3600	30	762	85.0	2160	55.6	5.16	111.2	10.33
60/3600	33.5	851	90.5	2300	66.1	6.14	132.2	12.28
60/3600	40†	1016	100.0	2540	87.3	8.11	174.6	16.22
50/3000	15*	381	64.2	1631	21.0	1.95	–	–
50/3000	17.5	445	70.0	1778	26.9	2.50	–	–
50/3000	22H*	559	88.0	2235	42.2	3.92	84.4	7.84
50/3000	26	660	91.0	2310	51.6	4.79	103.2	9.59
50/3000	33.5	851	99.5	2530	72.7	6.75	145.4	13.51
50/3000	42	1067	110.4	2804	101.2	9.40	202.4	18.80
50/3000	48†	1219	120.0	3048	125.7	11.68	251.4	23.36

*Suitable for extended ranges of exhaust pressure.

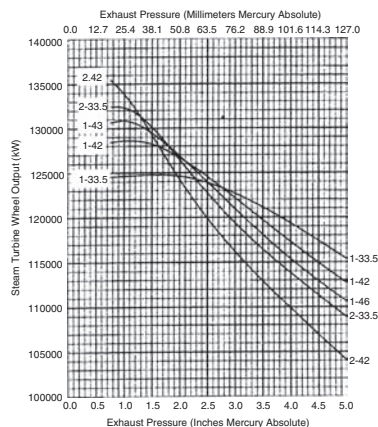
†Application limited to reheat cycles.

TABLE 2.3 Last Stages Available for Combined-Cycle Steam Turbines

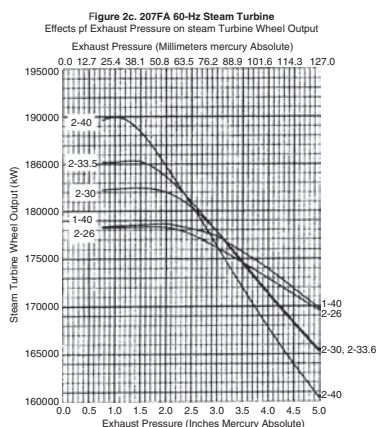
Intermediate pressure (IP) and low-pressure (LP) steam turbines are used in non-reheat cycles. These plants do not normally have extractions for feedwater heating. However, provisions are made within the low-pressure turbine to accommodate feedwater heating extraction(s) if the plant requires this extraction to meet the HRSG/stack minimum temperature requirements. The HRSG stack temperature is normally kept as low as possible. This is done to extract the maximum amount of energy from the gas turbine exhaust. Acid condensation on the stack is a concern in plants using high sulphur gas turbine fuels. LP turbine extraction is used in these applications to heat the feedwater above the acid dew point before supplying the feedwater to the HRSG economizer.



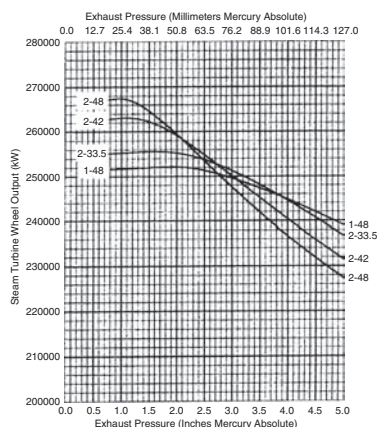
(a) 107FTA 60 Hz steam turbine



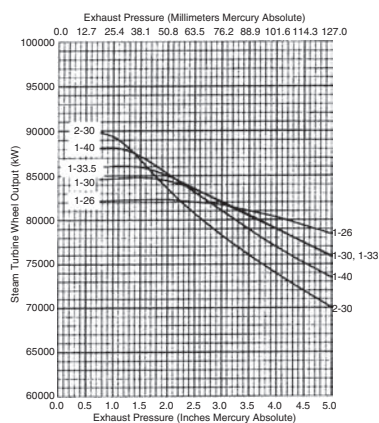
(b) 109FA 50 Hz steam turbine



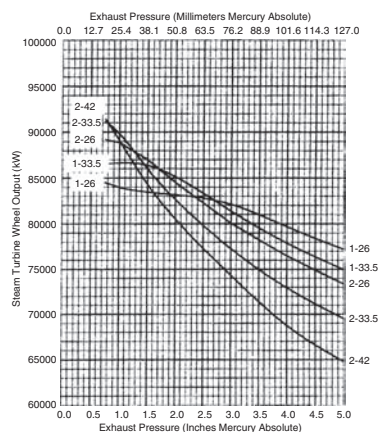
(c) 207FA 60 Hz steam turbine



(d) 207FA 50 Hz steam turbine



(e) 206FA 60 Hz steam turbine



(d) 206FA 50 Hz steam turbine

FIGURE 2.1 Steam turbine wheel output as a function of exhaust pressure and exhaust size, reheat STAG 1400 psig 538°C/1000°F (96 bar 538°C/1000°F) steam conditions. (Source: Reprinted with permission from General Electric.)

2.3.4 Reheat Cycle Steam Condition

The reheat cycle is employed when the gas turbine exhaust temperature exceeds 535°C (995°F). Figure 2.2 illustrates reheat and non-reheat expansions from 10 MPa (1450 psig) and 538°C (1000°F). The steam leaving the high-pressure turbine in the reheat case is returned to the HRSG. This steam is reheated back to the initial temperature.

The reheat cycle improves the performance of the plant due to the following reasons:

1. Heat is added to the steam at a higher average temperature than the non-reheat cycle.
2. Moisture is reduced in the low-pressure section. This reduces the last-stage moisture erosion in the turbine.
3. The efficiency of the plant is increased.
4. The size of the cooling system (condenser, etc.) and the amount of cooling flow are reduced.

Figure 2.3 illustrates the three-pressure reheat combined cycle. The H technology developed by some manufacturers employ a three-pressure reheat steam cycle. The initial steam conditions are either one of the following:

1. 16.5 MPa (2400 psig) and $565^{\circ}\text{C}/565^{\circ}\text{C}$ ($1050^{\circ}\text{F}/1050^{\circ}\text{F}$)
2. 12.4 MPa (1800 psig) and $565^{\circ}\text{C}/565^{\circ}\text{C}$ ($1050^{\circ}\text{F}/1050^{\circ}\text{F}$)

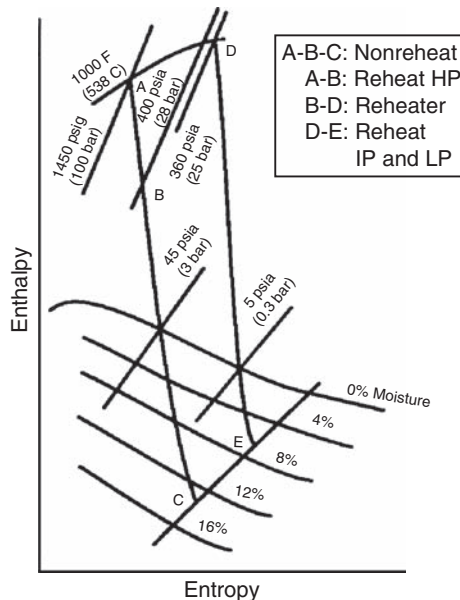


FIGURE 2.2 Comparison of non-reheat and reheat expansions. (Source: Reprinted with permission from General Electric.)

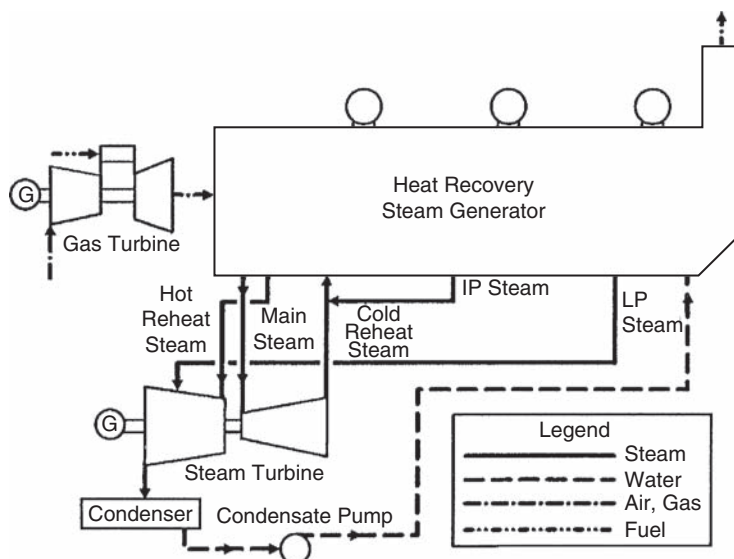


FIGURE 2.3 Three-pressure reheat cycle diagram. (Source: Reprinted with permission from General Electric.)

The higher initial pressure improves the efficiency of the plant. Multi-casing design with a short inner shell over the first few HP stages is used for this cycle.

The 7H and 9H gas turbines employ the advanced technology, closed-circuit steam cooling systems. The gas turbine cooling system is integrated in the following areas as follows:

1. Steam is supplied from the HP steam turbine exhaust and the HRSG IP evaporator (boiler) to the closed-circuit system. This system cools the nozzles and buckets of the first two stages of the turbine. The cooling steam is returned in the hot reheat line to the steam cycle.
2. Air extracted from the compressor discharge of the gas turbine is cooled externally. This air is then used to cool the wheels in the high-pressure stages of the compressor. Water from the discharge of the IP economizer in the HRSG is used to cool the cooling air. This water is used subsequently to heat the natural gas fuel.

2.4 Steam Turbine Product Structure

2.4.1 Performance

The performance of the steam turbines used in the steam and gas systems (STAG) listed in Table 2.2 is presented in Tables 2.4 through 2.8. The steam turbine output is listed for various configurations and exhaust pressures.

The suggested turbine type is given in Tables 2.4 through 2.8 in terms of the following:

- 1. Last-stage bucket size
- 2. Number of low-pressure turbine flows for different condenser (exhaust) pressures

These tables indicate that the exhaust annulus area decreases as the exhaust pressure increases. Two or three steam turbine designs are recommended in most cases for the plants.

The net efficiency for the H combined-cycle plants is 60%. The significant increase in efficiency that these plants have over the F technology is achieved by increasing the pressure ratio and firing temperature. This advantage is achieved while maintaining single-digit NO_x and CO capability.

EXHAUST PRESSURE, INCHES (MM) MERCURY ABSOLUTE									
Hertz	STAG	0.75 (19)		1.5 (38)		2.5 (64)		3.5 (89)	
		Steam Turbine	Wheel Output	Steam Turbine	Wheel Output	Steam Turbine	Wheel Output	Steam Turbine	Wheel Output
60	106B	1 × 26	24.0						
		1 × 23	23.6	1 × 23	22.9				
						1 × 17.5	21.7	1 × 17.5	21.0
60	260	1 × 33.5	33.5						
		1 × 30	33.1						
				1 × 26	31.6				
				1 × 23	31.3	1 × 23	30.1		
						1 × 20	29.9		
						1 × 17.5	29.4	1 × 17.5	28.8
50	106B	1 × 26/91	24.2						
				1 × 17.5	23.1	1 × 17.5	22.1		
				1 × 15	21.9	1 × 15	21.5	1 × 15	21.0
50	260	1 × 33.5	35.0						
		1 × 26/91	34.0	1 × 26/91	32.9				
				1 × 17.5	30.9	1 × 17.5	30.3		
						1 × 15	28.0	1 × 15	27.7

Note: Throttle steam temperature for STAG 260 plants is 446°C (835°F).

TABLE 2.4 STAG Steam Turbine Selection Chart Non-Reheat Steam Turbines Less Than 40 MW 850 psig (58.5 bar) 510°C (950°F)

EXHAUST PRESSURE, INCHES (MM) MERCURY ABSOLUTE									
Hertz	STAG	0.75 (19)		1.5 (38)		2.5 (64)		3.5 (89)	
		Steam Turbine	Wheel Output	Steam Turbine	Wheel Output	Steam Turbine	Wheel Output	Steam Turbine	Wheel Output
60	206B	2 × 30	49.9						
		2 × 26	49.3						
		1 × 33.5	48.5						
				1 × 30	47.0				
				1 × 26	46.3	1 × 26	44.9		
						1 × 23	44.7		
						1 × 20	43.7	1 × 20	43.1
								1 × 17.5	42.4
60	107EA	2 × 30	51.0						
		2 × 26	50.3						
		1 × 33.5	49.4	1 × 30	48.0				
				1 × 26	47.1	1 × 26	45.7		
						1 × 23	45.4	1 × 23	44.2
						1 × 20	44.3	1 × 20	43.7
50	206B	1 × 42	49.8						
		1 × 33.5	48.8	1 × 33.5	47.1				
				1 × 26/91	46.9	1 × 26/91	44.7		
						1 × 17.5	44.2	1 × 17.5	43.4

TABLE 2.5 STAG Steam Turbine Selection Chart Non-Reheat 40 to 60 MW Steam Turbines 1000 psig (69 bar) 510°C (950°F)

2.4.2 Casing Arrangements

Figure 2.4 illustrates the schematics of available STAG steam turbine casing arrangements.

Non-Reheat Multi-Shaft

There is a flexible expansion joints between the turbine exhaust and the condenser in the STAG plant shown in Fig. 2.4a. The HP turbine is fixed to the foundation in this design. Figure 2.5 illustrates a single-casing turbine with an axial exhaust. Figure 2.6 illustrates the double-flow turbine.

EXHAUST PRESSURE, INCHES (MM) MERCURY ABSOLUTE									
Hertz	STAG	0.75 (19)		1.5 (38)		2.5 (64)		3.5 (89)	
		Steam Turbine	Wheel Output	Steam Turbine	Wheel Output	Steam Turbine	Wheel Output	Steam Turbine	Wheel Output
60	406B	2 × 33.5	97.3						
		2 × 30	95.7	2 × 30	94.4				
				2 × 26	93.0	2 × 26	90.0		
						1 × 33.5	89.9		
						1 × 30	89.0	1 × 30	87.0
60	207EA	2 × 133.5	100.3						
		2 × 30	98.5	2 × 30	97.5				
				2 × 26	95.9	2 × 26	92.9		
						1 × 33.5	92.6		
						1 × 30	91.6	1 × 30	89.8
50	406B	2 × 42	94.8						
				2 × 33.5	94.6				
				2 × 26/91	94.2				
				1 × 42	94.1	1 × 42	89.9		
						1 × 33.5	90.3		
								1 × 26/91	87.0
50	109E	1 × 42	72.7						
				1 × 33.5	70.0				
				1 × 26/91	68.2	1 × 26/91	67.0	1 × 26/91	64.7
50	209E	2 × 42	146.2						
				2 × 33.5	141.0				
				2 × 26/91	137.4	2 × 26/91	134.9		
						1 × 42	134.7	1 × 42	130.3
								1 × 33.5	130.5

TABLE 2.6 STAG Steam Turbine Selection Chart Non-Reheat Steam Turbines Greater Than 60 MW
1250 psig (86 bar) 510°C (950°F)

EXHAUST PRESSURE, INCHES (MM) MERCURY ABSOLUTE									
Hertz	STAG	0.75 (19)		1.5 (38)		2.5 (64)		3.5 (89)	
		Steam Turbine	Wheel Output	Steam Turbine	Wheel Output	Steam Turbine	Wheel Output	Steam Turbine	Wheel Output
50	106FA	1 × 42	44.7						
		1 × 33.5	44.5						
		3 × 26/91	43.5	1 × 26/91	42.5				
						1 × 17.5	41.5	1 × 17.5	40.4
50	206FA	2 × 42	91.2						
		2 × 33.5	90.9						
				1 × 33.5	86.1	1 × 33.5	83.1		
						1 × 26/91	82.9	1 × 26/91	80.7
50	109EC	2 × 42	102.3						
		2 × 33.5	101.3						
				1 × 33.5	95.6	1 × 33.5	93.3		
						1 × 26/91	92.2	1 × 26/91	90.3
50	209EC	2 × 48	203.7	2 × 48	197.8				
				2 × 42	197.3				
				2 × 33.5	194.4				
						1 × 48	189.6	1 × 48	183.8
						1 × 42	187.4	1 × 42	183.9
50	109FA	2 × 42	135.5						
		2 × 33.5	132.5	2 × 33.5	130.2				
				1 × 48	129.8				
				1 × 42	124.5	1 × 42	124.5		
						1 × 33.5	124.0	1 × 33.5	121.1
50	209FA	2 × 48	267.4	2 × 48	264.3				
				2 × 42	262.3				
						2 × 33.5	253.9	2 × 33.5	248.1
						1 × 48	251.8	1 × 48	248.0

TABLE 2.7 STAG Steam Turbine Selection Chart 50 Hertz Advanced Combined Cycles 1400 psig (96 bar) 538°C/1000°F

EXHAUST PRESSURE, INCHES (MM) MERCURY ABSOLUTE									
Hertz	STAG	0.75 (19)		1.5 (38)		2.5 (64)		3.5 (89)	
		Steam Turbine	Wheel Output	Steam Turbine	Wheel Output	Steam Turbine	Wheel Output	Steam Turbine	Wheel Output
60	106FA	1 × 33.5	44.5						
		1 × 30	44.0						
				1 × 26	42.5				
				1 × 23	42.0	1 × 23	40.9		
						1 × 20	40.4		
						1 × 17.5	39.8	1 × 17.5	39.3
60	206FA	2 × 33.5	90.8						
		2 × 30	89.8						
		1 × 40	88.1						
				1 × 33.5	86.0				
				1 × 30	84.8	1 × 30	83.4		
						1 × 26	82.3	1 × 26	80.8
60	107EC	2 × 33.5	70.1						
		2 × 30	69.7						
		1 × 40	68.7	1 × 40	66.7				
				1 × 33.5	66.4				
				1 × 30	65.9				
						1 × 26	63.4		
						1 × 23	62.8	1 × 23	61.6
60	207EC	2 × 40	139.7	2 × 40	135.6				
				2 × 33.5	135.0				
				2 × 30	134.0				
				1 × 40	131.7	1 × 40	129.8		
						1 × 33.5	127.8		
						1 × 30	125.9	1 × 30	125.0

TABLE 2.8 STAG Steam Turbine Selection Chart 60 Hertz Advanced Combined Cycles 1400 psig (96 bar) 538°C/1000°F

EXHAUST PRESSURE, INCHES (MM) MERCURY ABSOLUTE									
Hertz	STAG	0.75 (19)		1.5 (38)		2.5 (64)		3.5 (89)	
		Steam Turbine	Wheel Output	Steam Turbine	Wheel Output	Steam Turbine	Wheel Output	Steam Turbine	Wheel Output
60	107FA	2 × 33.5	96.7						
		2 × 30	95.4						
				1 × 40	93.0				
				1 × 33.5	91.4	1 × 33.5	89.3		
						1 × 30	88.9		
								1 × 26	86.0
60	207FA	2 × 40	189.6	2 × 40	188.4				
				2 × 33.5	185.2				
				2 × 30	182.4	2 × 30	180.3		
						1 × 40	178.1	1 × 40	175.7

TABLE 2.8 STAG Steam Turbine Selection Chart 60 Hertz Advanced Combined Cycles 1400 psig (96 bar) 538°C/1000°F (Continued)

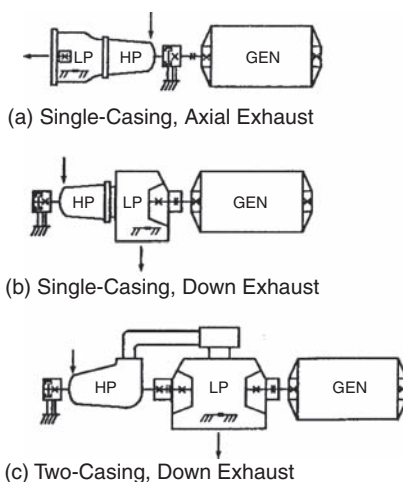


FIGURE 2.4 Non-reheat steam turbine arrangements for multi-shaft STAG: (a) Single-casing, axial exhaust, (b) Single-casing, down exhaust, and (c) Two-casing, down exhaust. (Source: Reprinted with permission from General Electric.)

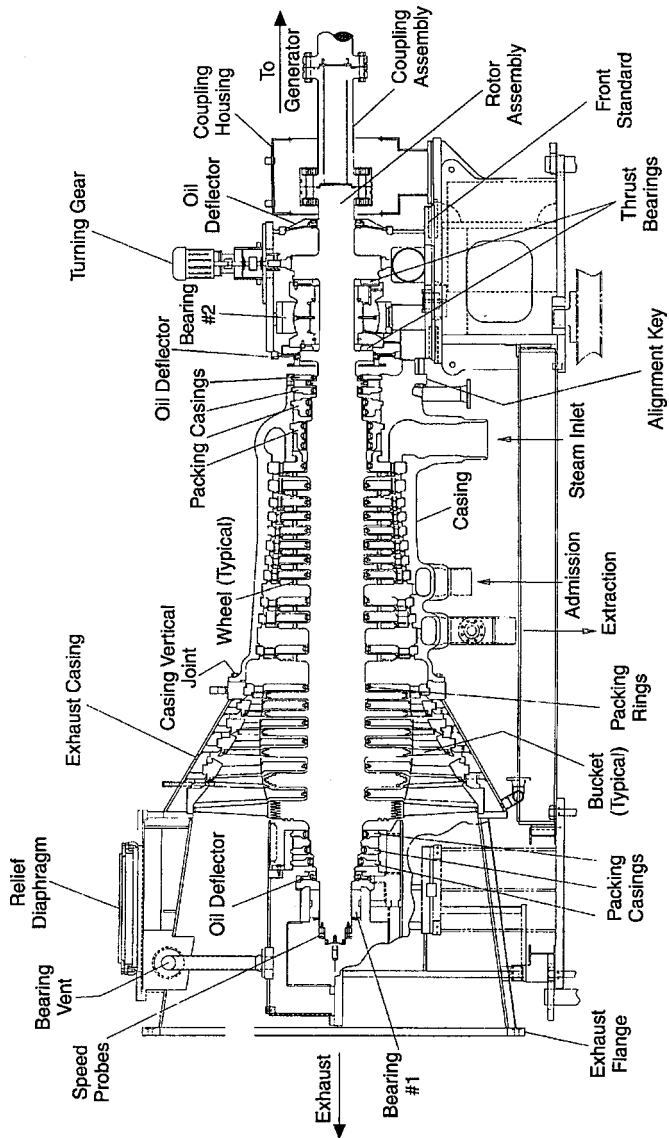


FIGURE 2.5 Non-reheat, single-casing, axial exhaust steam turbine. (Source: Reprinted with permission from General Electric.)

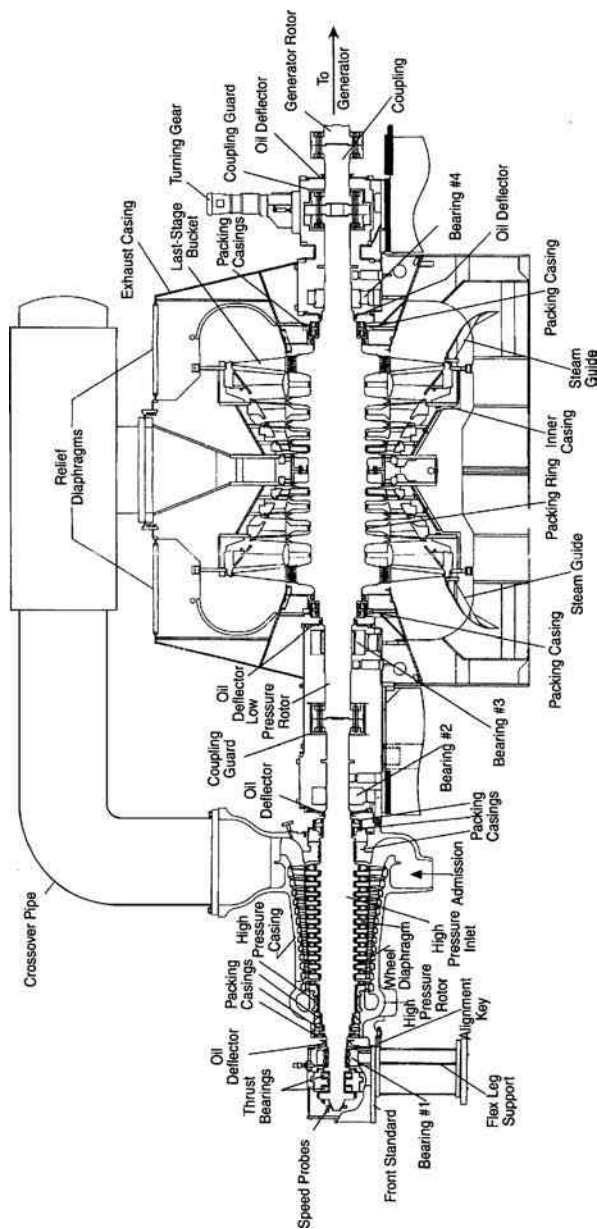


FIGURE 2.6 Non-reheat, double-flow down exhaust unit. (Source: Reprinted with permission from General Electric.)

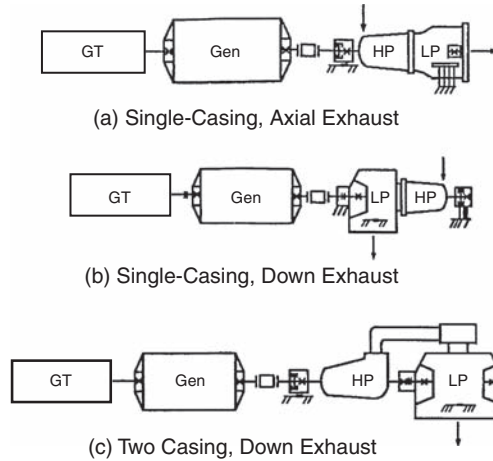


FIGURE 2.7 Non-reheat steam turbine arrangements for single-shaft STAG: (a) Single-casing, axial exhaust, (b) Single-casing, down exhaust, and (c) Two-casing, down exhaust. (Source: Reprinted with permission from General Electric.)

Non-Reheat Single Shaft

Figure 2.7 illustrates non-reheat single-shaft arrangements. Each turbine has the following:

1. Thrust bearing
2. Over-speed protection

A flexible coupling is located between the generator and steam turbine. This coupling accepts limited axial motion.

Reheat Multi-Shaft

Figure 2.8 illustrates the arrangements for multi-shaft reheat STAG plants. There is a separate generator for the steam turbine in this design. Figure 2.9 illustrates the cross-section of the single-casing reheat turbine shown in Fig. 2.8a.

Figure 2.10 illustrates the cross-section of the two-casing reheat turbine shown in Figs. 2.8b and 2.8c. Figure 2.8d illustrates the multi-shaft double-flow exhaust arrangement.

Reheat Single Shaft

Figure 2.11 illustrates the arrangements for single-shaft to reheat STAG plants. These units use a single-thrust bearing. This bearing is located at the inlet to the gas turbine compressor.

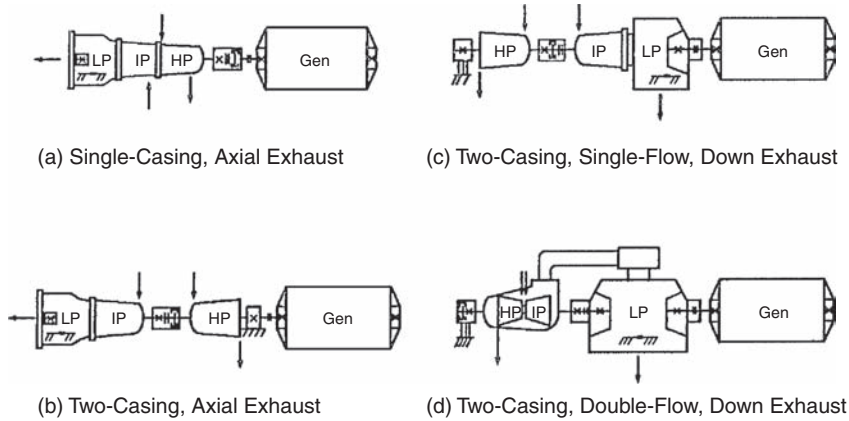


FIGURE 2.8 Reheat steam turbine arrangements for multi-shaft STAG: (a) Single-casing, axial exhaust, (b) Two-casing, axial exhaust, (c) Two-casing, single-flow, down exhaust, and (d) Two-casing, double-flow, down exhaust. (Source: Reprinted with permission from General Electric.)

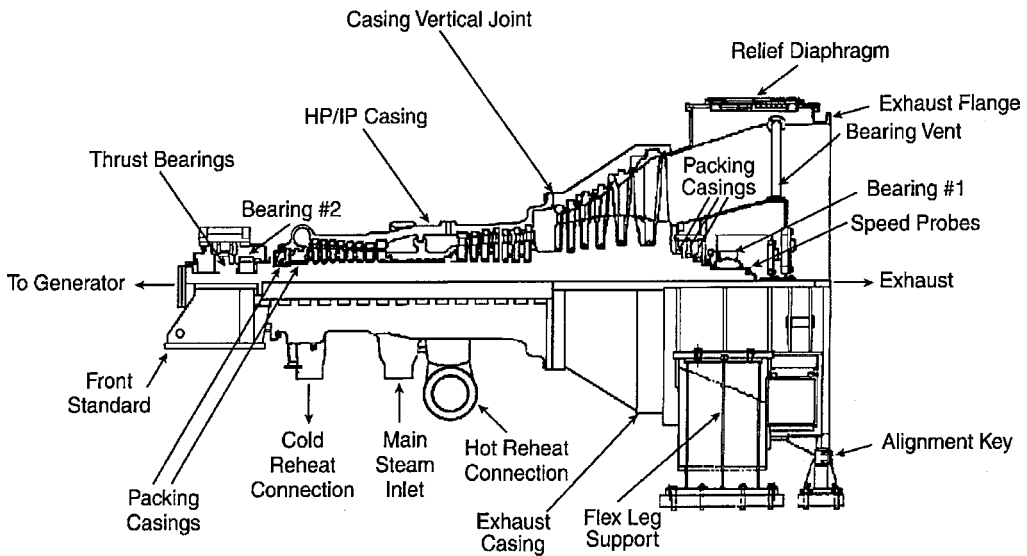


FIGURE 2.9 Single-casing reheat turbine with axial exhaust. (Source: Reprinted with permission from General Electric.)

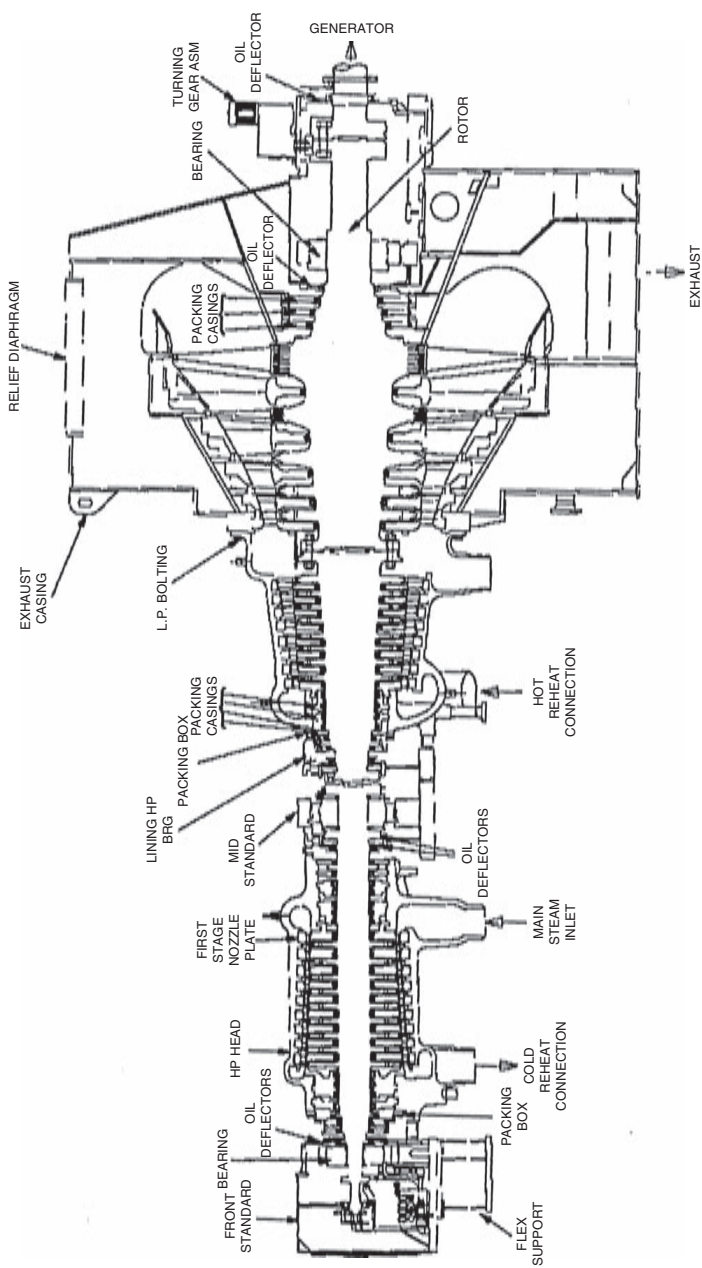


FIGURE 2.10 Two-casing reheat turbine with single-flow down exhaust. (Source: Reprinted with permission from General Electric.)

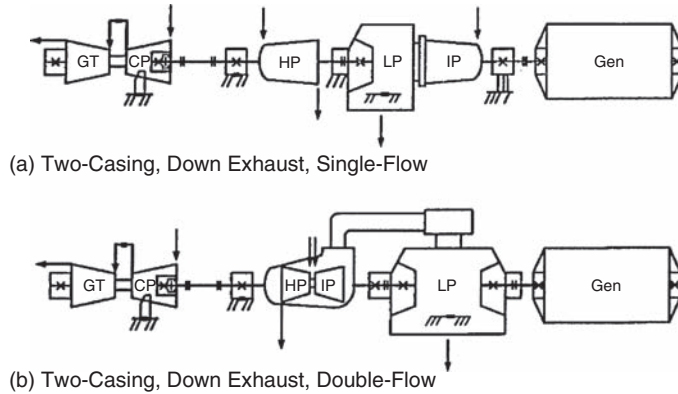


FIGURE 2.11 Reheat steam turbine arrangements for single-shaft STAG: (a) Two-casing, down exhaust, single-flow and (b) Two-casing, down exhaust, double-flow. (Source: Reprinted with permission from General Electric.)

2.4.3 Cogeneration Applications

Figure 2.12 illustrates a steam turbine used in a modern cogeneration STAG application. This unit has multiple inlet control valves and an automatic extraction for cogeneration STAG application.

2.5 Bibliography

Boss, M., "Steam Turbine for STAG Combined Cycle Power Systems," *GE Power Systems*, Schenectady, New York, 1994.

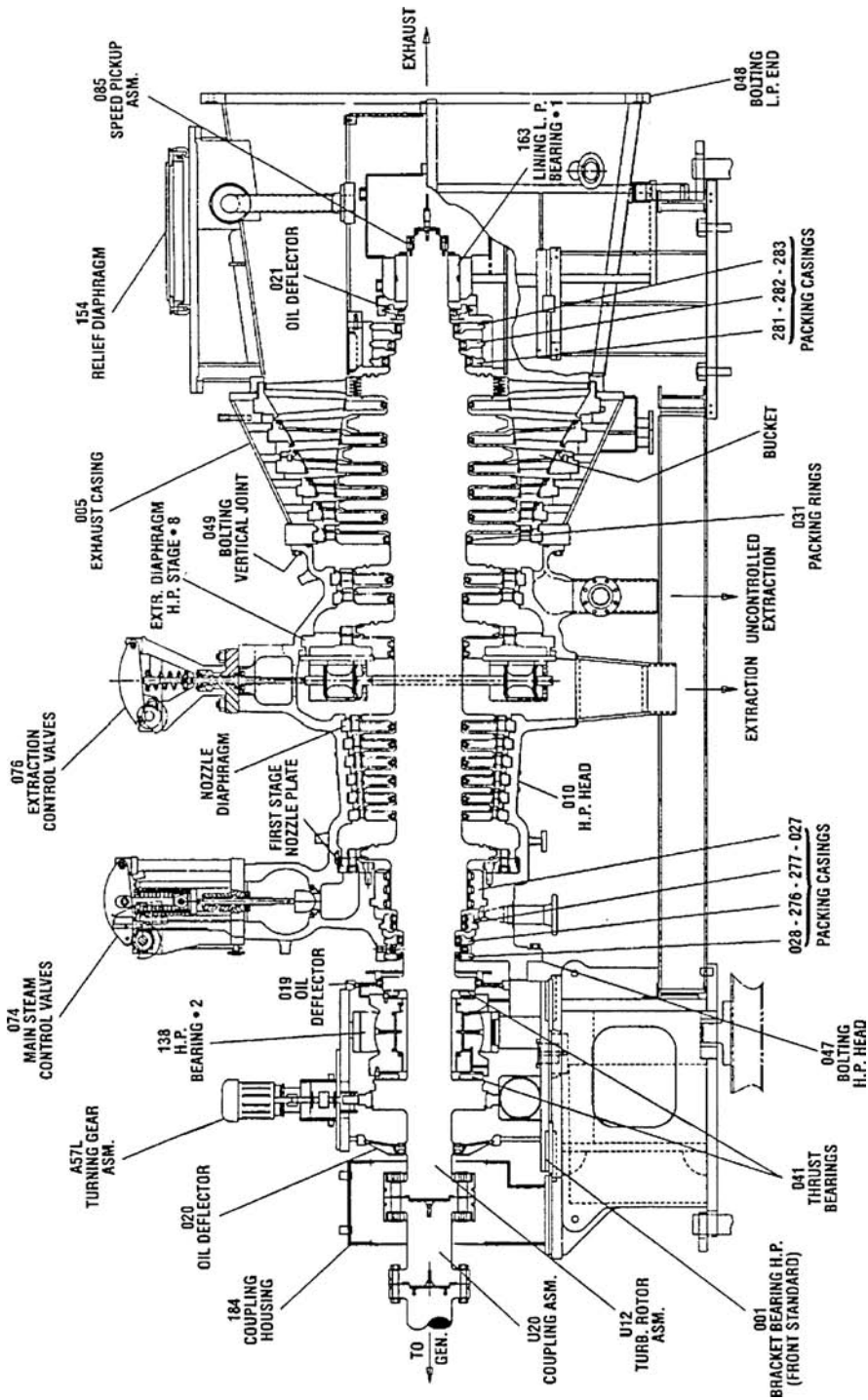


FIGURE 2.12 Packaged cogeneration unit with multiple inlet control valves and automatic extraction. (Source: Reprinted with permission from General Electric.)

Steam Turbine Maintenance

3.1 Life Cycle Operating Cost of a Steam Turbine

The life cycle operating cost of a steam turbine can be divided into the following categories:

1. Replacement parts and maintenance (direct cost)
2. Downtime due to lack of availability (indirect cost)

The indirect cost includes the cost of replacement power which is normally obtained from a more expensive source. Thus, the indirect cost can be significantly higher than the direct cost. Therefore, the implementation of a suitable maintenance strategy for a steam turbine has a significant impact on the post-installation operating costs of a unit.

3.2 Steam Turbine Reliability

The steam turbine reliability can be improved by limiting the starting, loading, and unloading temperature change rates. It is essential to abide by the heating and cooling rates specified by the manufacturer. For example, a limit of 1.56°C (2.8°F) per minute has been imposed by some manufacturers on the heating and cooling rates of steam turbines.

3.3 Boroscopic Inspection

Visual examination of some of the latter stages of low-pressure turbines is permitted without disassembly. Figure 3.1 illustrates a borescope access to the two stages upstream of the last stage. This feature is available on most modern units. It permits the inspection for damage due to moisture erosion, and foreign objects of steam path components, such as diaphragms, nozzles (stationary blades), buckets (moving blades), etc.

3.4 Major Cause of Steam Turbine Repair and Maintenance

The major cause of steam turbine repair and maintenance work is the erosive particles and impurities in the steam that cause deposits and chemical attack on turbine components. Thus, plant engineers and operators play a critical role in minimizing the maintenance and repair required by reducing the oxide formation and exfoliation in the steam-generator components and by maintaining adequate steam chemistry.

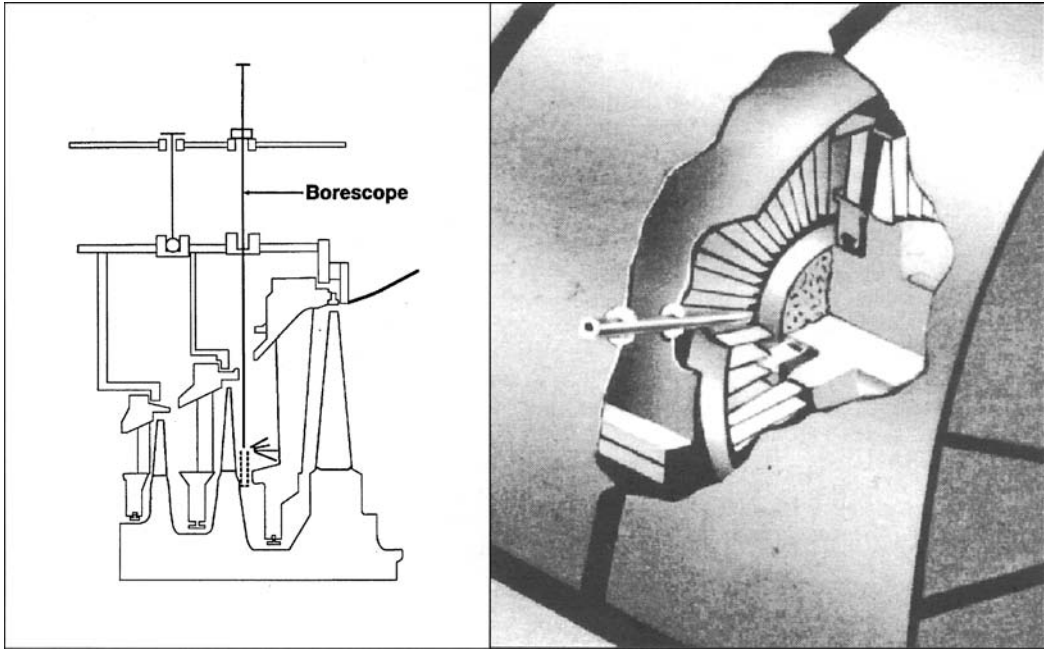


FIGURE 3.1 Borescope access. (Source: Reprinted with permission from General Electric.)

3.5 Maintenance Activities

The following are the major maintenance activities required on a steam turbine:

- Replace damaged buckets.
- Perform weld repairs when cracking or erosion is found on the exhaust hoods and inner casings of low-pressure turbines. These units are normally made of carbon steel alloys.
- Inspect the bearings.
- Machine the bolt holes of the couplings and insert an annular ring in each hole to restore the circular shape of the bolt holes (the circular shape of the bolt holes becomes deformed due to the high forces carried by the bolts).
- Perform nondestructive examination on the rotor body to determine if the rotor can be used for long-term service. Welding repairs should also be performed on the rotor if damage is found.
- Perform weld repairs, if required, on the stationary steam components such as nozzle plates and diaphragms. These components are normally damaged by steam impurities.
- Apply protective hard coatings on the stationary steam components such as nozzle plates and diaphragms to minimize the damage caused by the impurities. These coatings can be retrofitted on existing units. They are also included in most modern units.

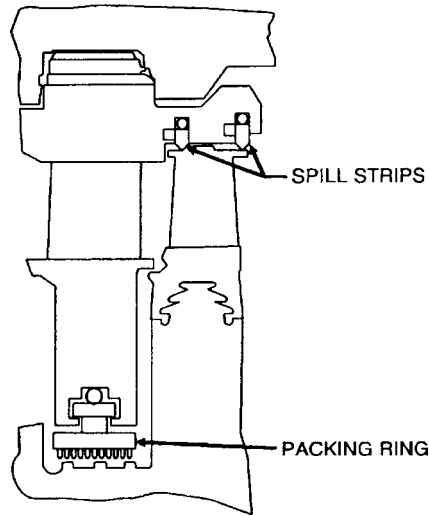


FIGURE 3.2 Removable seals. (Source: Reprinted with permission from General Electric.)

- Inspect and replace, if required, the bucket tip spillstrips and diaphragm shaft packing seals (Fig. 3.2). The wear of these components will increase the steam path efficiency losses due to an increase in steam leakage over the bucket tips and through the nozzle diaphragm bores. The bucket tip spillstrips and diaphragm shaft packing seals are made of materials that minimize wear if radial rubbing occurs. They are normally spring-loaded in the radial direction. They can deflect radially away from the rotor when rubbing occurs. This greatly minimizes the rubbing force and wear acting on the sealing and rotor components.
- Realign the thrust and journal bearings if required. Shims are used to facilitate realignment. These bearings can be removed for maintenance. Figure 3.3 illustrates a thrust bearing. It is designed to position the rotor assembly axially relative to the stationary components. The journal bearings support the weight of the rotor on a hydrodynamic oil film. They also provide dynamic stability and alignment to the rotor.
- Replace the tilt pads in the journal and thrust bearings if required.
- Perform inspection and maintenance of oil heat exchangers, including ultrasonic tests on the shell, eddy current inspection of the tubes, chemical cleaning, tube plugging, etc., as required. Fully redundant heat exchangers can be removed online.
- Install full-flow lube oil filtration system. This feature can be retrofitted to existing units. It is also available on new units. This system ensures that the lube oil supply is clean. It also reduces operational problems and required maintenance significantly. Impurities in the oil can cause serious damage to the bearings, rotor journals, and hydraulic control components. Modern systems allow online filter maintenance. In some cases, filter replacement can be performed while still maintaining continuous full-flow filtration capability.

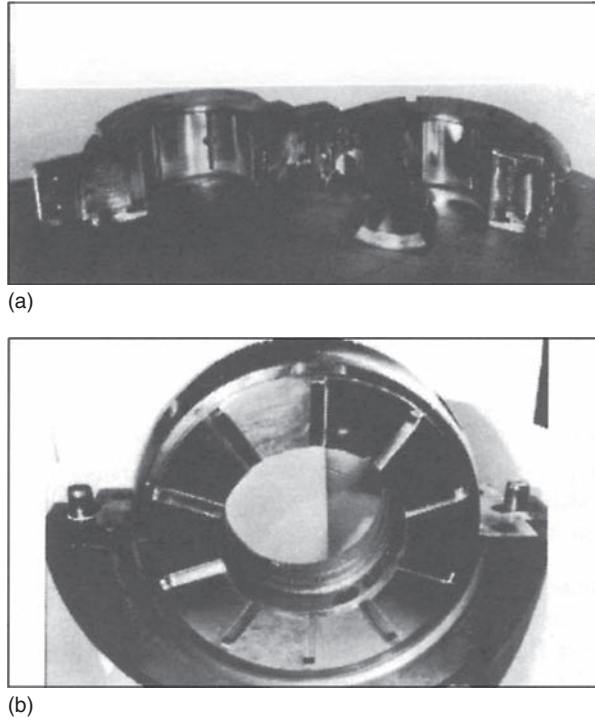


FIGURE 3.3 Tilting-pad journal bearing (a) and thrust-bearing assembly (b). (Source: Reprinted with permission from General Electric.)

- Balance the turbine-generator rotor using the computerized rotor-balancing method. This method provides the balance weights required for a multi-plane balance solutions. The possibility of rotor rubbing and the time required to balance the rotor are greatly reduced by using this method.

3.6 Advanced Design Features for Steam Turbines

The solid particle erosion (SPE) damage is the main cause of degradation in the efficiency of steam turbines. The iron oxide particles entrained with the steam erode the nozzles and buckets of the turbine. These particles are exfoliated from the inside surfaces of steam-generator tubes and the piping of the main and reheat steam.

Research has demonstrated that the SPE damage is caused by high velocity, low-angle impact of particles on the trailing edge of the blade. The first stages of the high-pressure turbine and of the reheat turbine receive normally most of the SPE damage. This damage can be reduced by applying erosion-resistant coating.

Service experience showed that the use of a protective coating alone will reduce the SPE damage for 1 to 2 years only. However, some manufacturers have overcome this limitation by developing a new blade design that reduces the SPE damage significantly. Figure 3.4 illustrates the new blade design. It uses slanted nozzle partitions to reduce the number of particle impacts on the trailing edge surface of the nozzles. The impacts

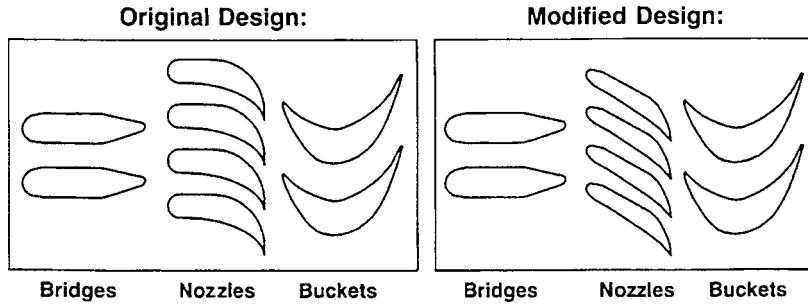


FIGURE 3.4 Modified control stage to minimize SPE damage. (Source: Reprinted with permission from General Electric.)

that occur are at a reduced velocity and very shallow impact angles. The use of this new nozzle shape in combination with a protective diffusion coating increases the service life of the nozzles to three times that of conventional designs. This new design does not reduce the “as new” efficiency of the turbine. It provides significant benefits by maintaining higher long-term sustained efficiency (i.e., lower long-term heat rate). Figure 3.5 illustrates the heat rate benefit of a typical unit.

The SPE-resistant stage is available in most modern units. It can also be retrofitted as well to existing nozzle box designs. The stage efficiency is brought back to “as new” level when the new SPE design is retrofitted. The new SPE-resistant nozzle design was installed on two supercritical double-reheat units. These units had well-established histories of severe SPE damage. The new design was inspected after 3 years in service. It was found to be virtually erosion-free. The old design had experienced severe SPE damage within 1½ years of service. The utility also achieved an improvement of 0.4% in the heat rate with the new design after the first 2 years of service.

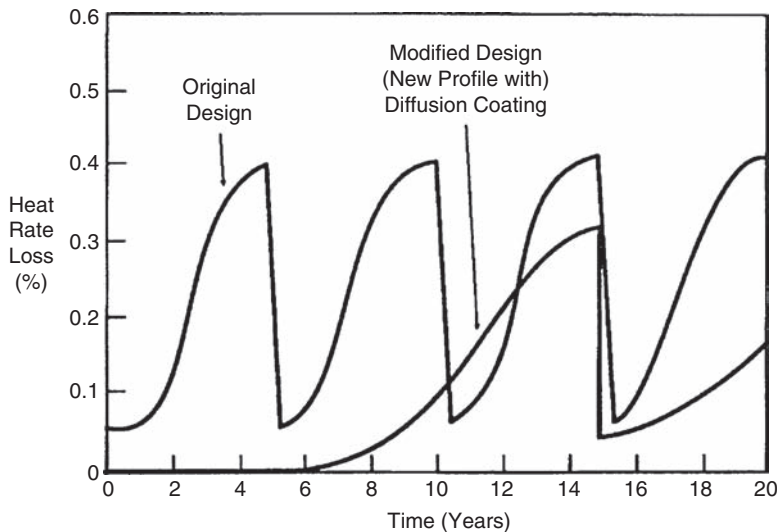


FIGURE 3.5 Control stage heat rate loss due to severe SPE damage. (Source: Reprinted with permission from General Electric.)

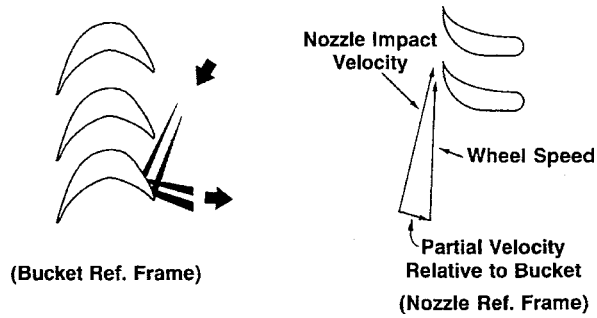


FIGURE 3.6 First reheat stage suction surface damage caused by particle rebounding. (Source: Reprinted with permission from General Electric.)

Research has showed that the cause of erosion in the first reheat stage nozzles and buckets is a complex entrapment phenomenon of particles that result in the multiple rebounding of captured particles between the nozzles and buckets. Figure 3.6 illustrates that the erosion in the first reheat stage nozzles is caused by the rebounding of particles from the leading edges of the buckets to the nozzles. The high velocity of the particles acquired by their impact with the buckets becomes the impact velocity of the particles on the nozzles. Figure 3.7 illustrates the severe SPE damage experienced by the nozzle surfaces of a reheat diaphragm.

Research showed that the reheat erosion mechanism can be reduced significantly by increasing the edge-to-edge axial clearance between the nozzles and buckets. Some manufacturers have also added a Diamond Tuff HVOF (high-velocity oxygen fuel) erosion-resistant coating to the nozzle trailing edge suction surface and the diaphragm outer sidewall.

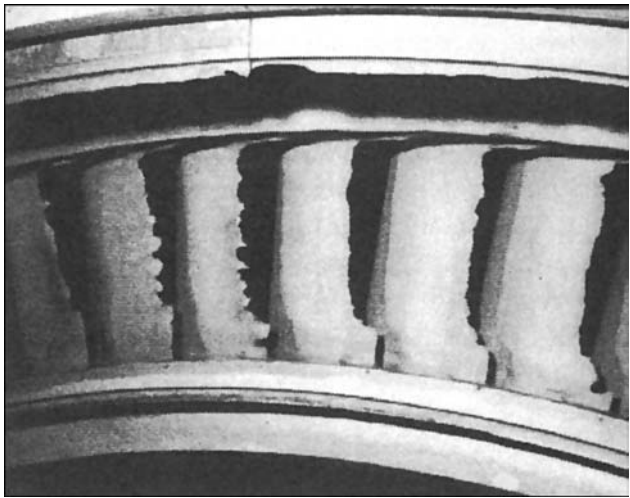


FIGURE 3.7 Severe erosion on suction side of first reheat stage diaphragm. (Source: Reprinted with permission from General Electric.)



FIGURE 3.8 Automatic plasma-spray process coats reheat diaphragm. (Source: Reprinted with permission from General Electric.)

Figure 3.8 illustrates the application of this coating by a robot to a reheat diaphragm. Figure 3.9 illustrates the improvement in heat rate resulting from the usage of new SPE-resistant design. These features are available in some new designs. They can also be retrofitted to most existing reheat diaphragms. Figure 3.10 illustrates an SPE-resistant diaphragm that was in service for 3½ years in the first reheat stage of supercritical reheat turbine. Figure 3.7 illustrates the condition of a conventional first reheat stage diaphragm from the same turbine after being in service for the same period.

Rubbing erodes the labyrinth seal packings. This erosion is significant during start-up. Excessive clearance caused by rubbing during the start-up phase results in a decrease in the efficiency of the unit. Some manufacturers provide positive-pressure clearance packing. This design provides a larger clearance during the start-up phase and reduced clearance when the unit is synchronized. Thus, it provides optimum

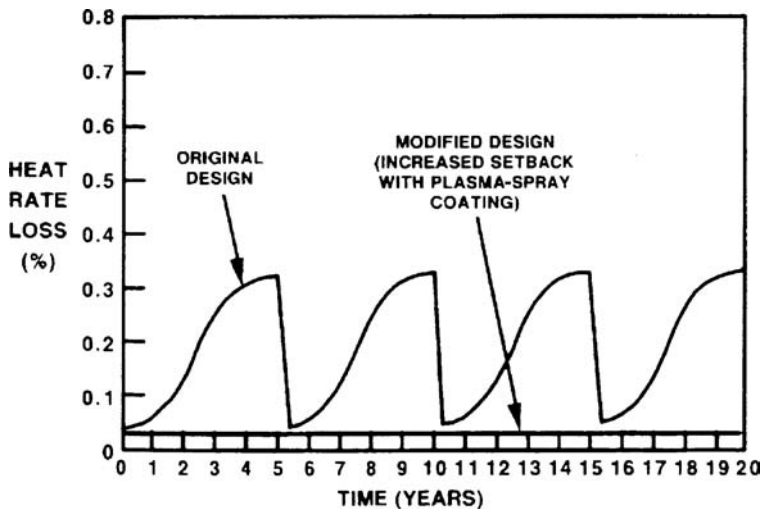


FIGURE 3.9 First reheat stage heat rate loss due to severe SPE damage. (Source: Reprinted with permission from General Electric.)

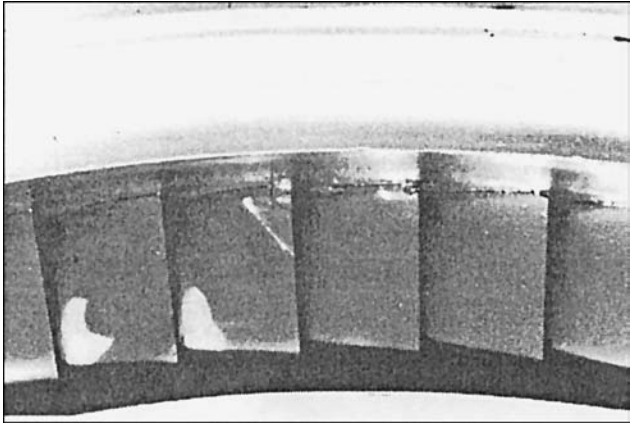


FIGURE 3.10 SPE-resistant diaphragm after 3½ years of service in the first reheat stage of a supercritical reheat turbine. (Source: Reprinted with permission from General Electric.)

sealing when the unit is loaded while minimizing rubbing during the start-up phase. Figure 3.11 illustrates a positive-pressure variable clearance packing. High-pressure steam is applied on the back of the packing ring to reduce the clearance of the packing teeth. Figure 3.12 illustrates the improvement in heat rate associated with the use of a positive-pressure packing (the standard packing is replaced every 5 years).

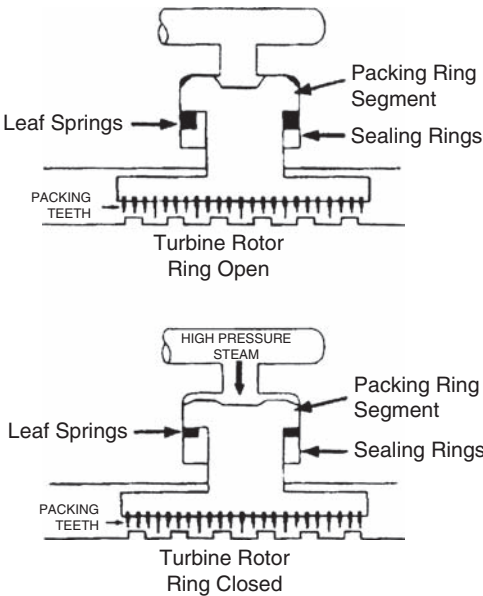


FIGURE 3.11 A positive-pressure packing. (Source: Reprinted with permission from General Electric.)

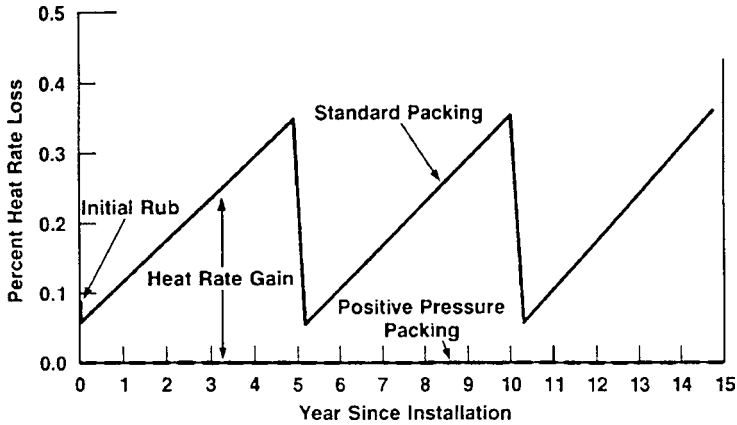


FIGURE 3.12 Heat rate improvement for positive-pressure packing in 500 MW units. (Source: Reprinted with permission from General Electric.)

3.7 Bibliography

- Bievenue, R. T., Ruegger, W. A., and Stoll, H. G., "Features Enhancing Reliability and Maintainability of GE Steam Turbines," GER-3741B, 38th GE Turbine State-of-the-Art Technology Seminar, Schenectady, New York, 1994.
- Cofer, J. I., Koenders, S., and Summer, W. J., "Advances in Steampath Technology," GER-3713C, 38th GE Turbine State-of-the-Art Technology Seminar, Schenectady, New York, 1994.

This page intentionally left blank

Frequently Asked Questions About Turbine-Generator Balancing, Vibration Analysis, and Maintenance

4.1 Balancing

1. What is the difference between high-speed balancing and low-speed balancing?

Low-speed balancing of a turbine rotor is performed at 300 to 600 rpm. High-speed balancing is performed at the rated speed (e.g., 3600 rpm). High-speed balancing is performed at the manufacturer before shipping the turbine to the user. Ideally, If the repair and refurbishment of a turbine rotor is performed at the manufacturer's facility, low and high-speed balancing are performed. This involves corrections to balance the rotor in the low-speed range, at the critical speeds, and at high-speed (also known as trim balancing). However, if the repair and refurbishment of the turbine rotor is performed at a different facility, only low-speed balancing is normally performed due to lack of capability to perform a high-speed balancing of the rotor.

It is essential to perform a moment-weighing of the blades before re-blading (changing of the blades). The weight and weight-profile of the blades located at the opposite ends of the rotor must be identical to prevent balancing problems.

2. Which rotor is not available for low-speed balancing? And why?

The generator rotor should not be balanced at low speed. This is because the low-speed balancing of this rotor does not confirm its integrity. If the generator rotor is relatively new, it should be balanced at 120% of the rated speed while the windings are hot. However, if the generator rotor is relatively old (more than 15 years), it should be balanced at 115% of the rated speed while the windings are hot.

3. Some turbine-generator rotors have high vibration even though each turbine rotor and the generator rotor have been balanced properly. What is the cause of this high vibration and solution for this problem?

Even if all the turbine rotors are balanced at low and high speed and the generator rotor has been balanced at the required over speed, high vibration can occur when all the turbine rotors and the generator rotor are coupled. This vibration is caused normally by misalignment of the rotor shafts or eccentric bearings. This problem can be rectified by aligning the rotor shafts properly and eliminating the eccentricity in the bearings.

4. What is static balancing?

Static balancing is normally performed on small rotors. In this process, the shaft is held on two knife edges to minimize the friction. The rotor will then settle at a specific plane. The heaviest part of the rotor will be located at the bottom of this plane. The position of this plane is marked. Trial weights are added on the opposite side of the heaviest part in order to balance the rotor. The rotor is allowed to settle again. If it settles at the same plane as before, the trial weights are changed. This process is continued until the rotor settles at a different plane.

4.2 Vibration Analysis—Cam Bell Diagram

1. What is the Cam Bell diagram?

The Cam Bell diagram is a graphical representation of the vibration profile at the critical speeds of a turbine-generator. It is also known as the critical speed map. Finite element method is applied to the Cam Bell formula to calculate the critical speeds. Several parameters for the turbine generator such as shaft length, shaft diameter, stiffness of bearing, and bearing foundation, must be provided to perform this calculation. The Cam Bell formula is very useful because it can also be used to determine the most effective location (plane) for balancing the turbine generator. There are points along the turbine generator where balancing is not effective. These points are called “noodle points.” The Cam Bell formula is useful in determining these noodle points as well. However, balancing of the turbine generator may require deviations from the ideal planes recommended by Cam Bell formula due to practical reasons. For example, if the ideal plane to balance the turbine generator is in the generator rotor and the generator cannot be opened for some reason. Balancing can be performed successfully by adding weights to the third low-pressure turbine rotor (LP3).

2. What is the difference between critical speed map and Cam Bell diagram?
The critical speed map is the Cam Bell diagram.

4.3 Turbine-Generator Maintenance

1. If a scratch or a crack is found in a turbine rotor or journal (the portion of the shaft located inside the bearing), how can it be repaired?

If a scratch less than 2 thou (0.002 in or 0.005 cm) is found in a turbine rotor or journal, apply emery cloth to blend it off. If the scratch (crack) is deeper, it should

be ground or machined. Non-destructive examination (NDE) involving magnetic particles inspection (MPI) should be performed. The crack should not be filled with any material to prevent damaging the bearings. The white metal of the bearings located around the turbine rotor should normally be replaced.

2. What is the standard for the replacement of the turbine blades? How can the lifetime of the turbine blades be estimated?

The following are some of the factors that affect the lifetime of the turbine blades:

- a. The steam particle erosion (SPE) is the main cause of blade damage (please refer to Chap. 3 "Steam Turbine Maintenance"). Improving the chemistry of the demineralized water will extend the lifetime of the blades.
- b. Applying a Diamond Tuff HVOF (high-velocity oxygen fuel) erosion resistant coating to the blades will extend their lifetime. Please refer to Chap. 3 "Steam Turbine Maintenance."
- c. Selecting a steam turbine having the advanced design features explained in Chap. 3 "Steam Turbine Maintenance" will extend the lifetime of the blades.

The increase in the discharge temperature of the turbine indicates deterioration of the blade condition. The decrease in the power output of the turbine could also be caused by deterioration of the blade condition. Thus, there are no hard rules to replace the blades. If the turbine efficiency drops by 10% in 3 years, the blades will have to be replaced. However, if adequate precautions are taken, the turbine blades should last about 8 years. A major inspection and overhaul is required after 8 years of service. Some of the blades may have to be replaced following the inspection.

3. If shaft bending is found after runout checks, how can it be corrected?

Bending in the turbine rotor can sometimes be corrected by running the turbine generator at low speed (200–300 rpm) for a period of time (2–6 hours). If the problem persists, detailed vibration analysis should be performed before adding weights to the rotor. Please refer to the chapter "Vibration Analysis" of the *Electrical Equipment Handbook* and "Generator Components, Auxiliaries, and Excitation" of the *Power Generation Handbook*.

4. What is electric runout?

The vibration instrumentation installed in the white metal of the bearings measure the displacement of the shaft. If there is runout in the shaft, the distance between the vibration instrument and the shaft will change as the shaft rotates. In some cases, the shaft is magnetized. This generates a signal in the vibration instrument similar to shaft runout even if the shaft does not have any runout. This phenomenon is called electric runout. The vibration signals generated by electrical runout can be significant if the shaft is highly magnetized. The vibration signals caused by electrical runout can be eliminated by periodically demagnetizing the shaft. Equipment known as "degauzing" is used to demagnetize the shaft at different areas to eliminate electrical runout.

5. What is the limit of rotor runout?

The allowable limit for the rotor runout is 1 thou (0.001 in or 0.0025 cm). If runout is found in the turbine generator, it must be corrected before operating the machine.

6. What are the limits of a rotor-shaft vibration?

It is recommended to adopt the following limits suggested by the International Standard Organization (ISO) for turbine-generator vibration:

- a.* At rated speed, the vibration amplitude shall not exceed $50\text{ }\mu\text{m}$ (0.002 in) peak to peak in both the vertical and transverse directions.
- b.* Over the remainder of the speed range, including passage through critical speeds, the vibration amplitude shall not exceed $125\text{ }\mu\text{m}$ (0.005 in) peak to peak, in both the vertical and transverse directions.

CHAPTER 5

Features Enhancing the Reliability and Maintainability of Steam Turbines

5.1 Steam Turbine Design Philosophy

The design objective for steam turbines is the minimization of product life cycle cost. This term is defined as the present value of all costs incurred by the turbine owner over the useful life of the unit. The life cycle cost is a function of the following parameters:

1. Capital cost
2. Installation cost
3. Efficiency
4. Reliability
5. Maintenance cost

The reliability and maintenance costs are among the most significant operating costs. These costs can be divided into the following categories:

1. *Direct* costs associated with the replacement parts and maintenance services of the turbine over its useful life.
2. *Indirect* costs associated with the lack of availability of the equipment (equipment downtime). These costs include the cost of replacement power required to compensate for the units unavailability. The replacement power is normally provided from a much more expensive source when the unit is out of service. The indirect costs can have a more significant impact on the overall life cycle cost than the direct costs. Thus, the reliability, availability, and the maintainability of the steam turbine play an essential role in minimizing the life cycle costs.

5.2 Measures of Reliability, Availability, and Maintainability

The service performance of a steam turbine generator unit is measured by the following criteria:

1. Reliability
2. Availability
3. Maintainability

Reliability is a measurement of freedom from forced outages that prevent unit operation due to equipment or system failure. It is measured traditionally by its reciprocal or “unreliability.” This measurement is known as the *forced outage rate*. It is the percentage of time that a turbine generator cannot operate when called upon. The forced outage rate is calculated as follows:

$$\text{FOR} = \text{FOH} / (\text{FOH} + \text{SH})$$

where FOR = unit forced outage rate

FOH = hours during a year a unit is unavailable due to a forced outage

SH = unit service hours during a year

The unit forced outage rate is a function of the following variables:

1. Mean time between failure (MTBF)
2. Mean time to repair (MTTR)

The MTBF is a measure of how often a unit fails. It depends mainly on the inherent design. The MTTR is a measure of the time required to repair a unit once it fails. It is a measure of the following parameters:

1. Parts availability
2. Maintainability of the design
3. Maintenance effort applied following the failure

Availability is defined as the percentage of time that a turbine generator is capable of providing power within a given period of time. This measurement is a function of the following parameters:

1. Reliability of the unit
2. Downtime required to perform scheduled maintenance

Availability is measured traditionally by the *availability factor*. This parameter is calculated as follows:

$$\text{AF} = (\text{PH} - \text{FOH} - \text{MOH} - \text{POH}) / \text{PH}$$

where AF = unit availability factor

FOH = unit forced outage hours during the period

MOH = unit maintenance outage hours during the period

POH = unit planned outage hours during the period

PH = period hours (typically 8760 hours per period)

The time required to perform scheduled maintenance has a strong effect on the availability. This time depends on the following parameters:

1. Manufacturer's maintenance recommendations
2. Degree of predictability of outage workscope before opening the turbine
3. Spare parts availability
4. Intensity of maintenance effort applied during the outage
5. Inherent maintainability of the turbine design

Design maintainability can be defined as the ease with which a unit can be disassembled, inspected, repaired, and reassembled for continued reliable and efficient operation. Maintainability has a strong impact on the unit's availability because it affects the MTTR associated with forced outages and the time required to perform maintenance activities during planned outages.

5.3 Design Attributes Enhancing Reliability

The post-installation operating costs associated with equipment reliability have a strong impact on the steam turbine life cycle cost. Most modern manufacturers place a strong emphasis on equipment reliability. This emphasis manifests itself in the overall design approach, as well as in the development of features that maintain high equipment reliability.

5.3.1 Overall Mechanical Design Approach

The mechanical design approach consists of comparing the in-service stresses and strains of turbine components with the corresponding material strengths or strain capabilities. Most modern steam turbine manufacturers use a statistical or probabilistic approach for establishing design basis. This is a more realistic approach than simply comparing a single value of stress to a single value of strength (conventional approach). The margin of safety was determined based on the difference between the two values. The probabilistic approach is significantly better than the conventional approach. This is because there will always remain a considerable amount of uncertainty in calculating or measuring stresses or strains in service under different operating conditions. Figure 5.1 illustrates the statistical comparison between stress and strength.

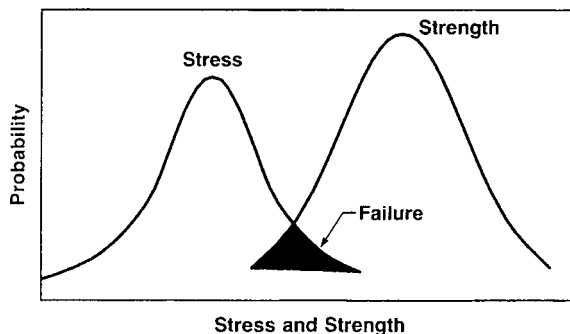


FIGURE 5.1 Probabilistic comparison between stress and strength. (Source: Reprinted with permission from General Electric.)

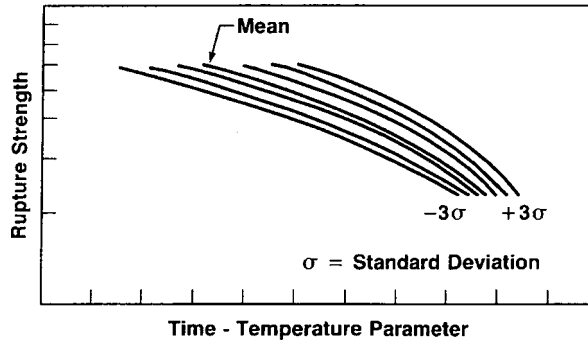


FIGURE 5.2 Illustrative statistical distribution of rotor-material rupture strength. (Source: Reprinted with permission from General Electric.)

The “permissible” probability of failure, or failure rate, is also shown in Fig. 5.1. A statistical evaluation is also performed on all material strengths. For example, Fig. 5.2 illustrates the rupture strength distribution of a rotor material.

5.3.2 Modern Steam Turbine Design Features

The following design features increase the reliability of all steam turbine generators regardless of size or application.

Impulse Wheel-and-Diaphragm Construction

Impulse stage design with wheel-and-diaphragm construction provides a rugged and reliable design due to the following reasons:

1. The pressure drop occurs across the stationary, rather than moving parts.
2. The impulse stage design requires fewer stages than the corresponding reaction stage design. The additional space available in the impulse stage design permits the installation of a sturdy diaphragm design.

These features reduce the failure rate in the moving and stationary parts of the steam turbine.

Turbine-Rotor Design

The rate at which the turbine generator load can be changed is limited by the thermal stresses that occur in high-temperature rotors. The highest thermal stresses occur at the rotor surface. These stresses depend heavily on the following parameters:

1. Diameter of the rotor body
2. Corresponding stress concentration factors

The wheel-and-diaphragm design has the following advantages over the reaction drum-rotor construction:

1. The diameter of the rotor body in the wheel-and-diaphragm design is much smaller than the corresponding rotor-body diameter in the reaction drum-rotor construction.

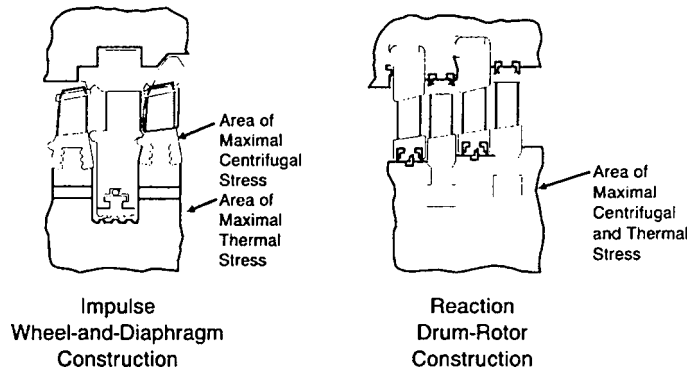


FIGURE 5.3 Wheel-and-diaphragm design separates regions of maximum thermal and centrifugal stress. (Source: Reprinted with permission from General Electric.)

2. The axial distance between stages in the wheel-and-diaphragm design is larger than the corresponding distance in the reaction drum-rotor construction. This permits generous fillet radii at the intersection of the rotor body and the side of the wheels (Fig. 5.3) in the wheel-and-diaphragm design. This feature results in low stress concentration factors at the point of maximum thermal stress. In contrast, the reaction drum-rotor design has relatively higher stress concentration factors due to the intricate geometry required in the blade attachment area. Thus, the thermal stresses in the wheel-and-diaphragm construction are significantly lower than the corresponding stresses in the reaction drum-rotor construction. Therefore, the wheel-and-diaphragm design has a much higher capability for load cycling than the reaction drum-rotor design.
3. The region where the maximum rotor thermal stress occurs in the wheel-and-diaphragm design is separated from the bucket dovetail region where the maximum centrifugal stress occurs. However, in the drum-rotor construction, these regions occur at the same location (Fig. 5.3). Thus, an impulse wheel-and-diaphragm turbine rotor will normally have a longer, trouble-free operating life than a comparable reaction turbine rotor if the two rotors are being subjected to the same operating conditions.

Interstage Sealing Components

The impulse wheel-and-diaphragm design has another advantage. It is the ability to use spring-backed packing for sealing between turbine stages. These packings are supported in the diaphragms. They can move radially if contacted by the rotor during operating transients. In contrast, the reaction designs have dense stage spacing characteristic. This necessitates the use of fixed interstage packings. Thus, “hard rubs” occur during operating transients that could lead to forced outages due to high vibration.

Bearings

Tilting-pad journal and thrust bearings have the following advantages when used with modern steam turbines:

1. They provide excellent rotor damping to prevent erratic vibration patterns.
2. They prevent shaft instability, or whip.
3. They can tolerate a higher level of rotor misalignment than conventional bearings.

These bearings increase the reliability of the turbine generator due to the following reasons:

1. They make the turbine less susceptible to trips
2. They reduce the bearing failure rate.

Auxiliary Systems

Manufacturers of modern steam turbines build redundancy in the critical components of the auxiliary systems to increase the reliability. The following are examples:

- *Lube Oil Pumps:* All lube oil pumps have a backup pump driven by an AC motor. An emergency pump driven by a DC motor is used when all the station AC power is lost to provide power to the bearings during turbine rundown.
- *Hydraulic Oil Pumps:* Two 100% capacity pumps are used in all oil hydraulic systems. These pumps are normally driven by AC motors. An automatic bumpless transfer will occur if one pump fails during operation to avoid a turbine trip.
- *Filters:* Full-flow lube oil filters are used with modern steam turbines. They filter all the oil supplied to the bearings. The filters can be changed online. An inline filter is used with each hydraulic oil pump. It cleans the hydraulic oil.
- *Coolers:* Two 100% shell-and-tube heat exchangers are used for all-utility turbine-lubricating oil systems. These heat exchangers are installed in parallel. They incorporate a transfer valve system that allows online switching from one heat exchanger to the other.
- *Solenoid Valves:* The steam turbine stop valves are provided with solenoid test valves to enable online (while the unit is operating) stroke testing. Redundant servo valve coils can be provided.

Controls and Instrumentation

A triple modular redundant (TMR) digital control system is used by most manufacturers of modern steam turbines. These systems provide complete redundancy from sensor to the servo coil of the valve actuator. This reduces the number of spurious trips caused by the failure of the sensor or control processor. These control systems incorporate additional self-diagnostic features that warn operators of problems before they affect the unit's performance. All faulty control system components can be replaced online without requiring a unit shutdown.

The following sensors are provided for all modern steam turbines:

1. Proximity probe-type vibration sensors are installed near each radial bearing.
2. Bearing metal temperature sensors are installed in each radial and thrust bearing.

These sensors provide early warning of impending problems to prevent costly forced outages.

Continuously Coupled Last-Stage Turbine Buckets

Some last-stage buckets (moving blades) of modern steam turbines are continuously coupled. This design includes a nub-and-sleeve mid-vane connection. Each bucket in this design is loosely coupled at the mid-vane area and the tip to its two adjoining buckets (Fig. 5.4). The damping of random and deterministic excitation forces increases significantly due to these couplings when compared with free-standing designs. The increase in damping has been larger at low load conditions where bucket flutter and buffeting have caused problems in free-standing designs. Figure 5.5 illustrates the significant decrease in bucket buffeting response due to the continuously coupled design. This feature has prevented all bucket failures during the last two decades.

Special Features of Industrial Turbines

Some modern steam turbines are based on a modular or “building block” structure. This feature allows the manufacturer to optimize a turbine configuration for specific operating conditions. This is done by implementing the following:

1. Selecting and integrating pre-engineered and field-proven components from an array of alternative offerings.
2. Designing a custom steampath that satisfies the specific requirements of the application.

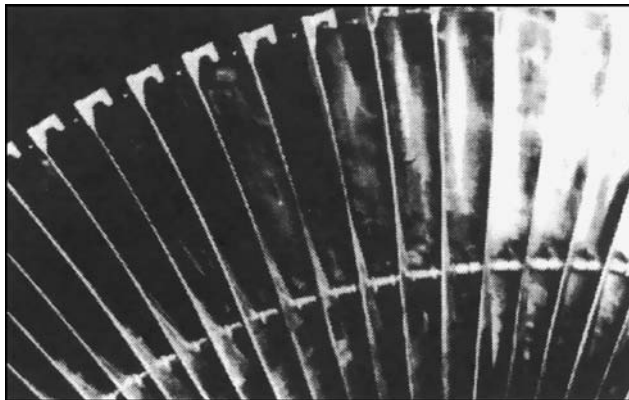


FIGURE 5.4 Continuously coupled last-stage. (Source: Reprinted with permission from General Electric.)

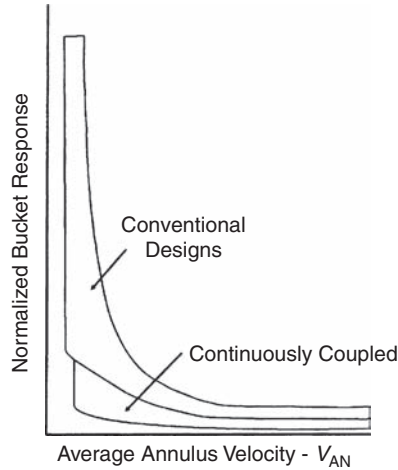


FIGURE 5.5 Maximum bucket buffeting response for conventional and continuously coupled designs. (Source: Reprinted with permission from General Electric.)

The component modules that make up the “building blocks” of the product line include the following:

1. Bearing standards
2. Inlet sections
3. Valve gear
4. Extraction and exhaust sections

The turbine reliability is increased by selecting components from an array of field-proven designs.

The most reliable design for steam generators in the 20–120 MW range have the following features:

1. They utilize single casing construction.
2. The turbine drives the generator directly without the use of a reduction gear.

The reasons for the increase in reliability in these designs compared with the alternative designs that utilize two turbine casings and a reduction gear are

1. The use of lower number of component parts.
2. They have a lower risk of misalignment and accidental overspeed.

Some modern steam turbines use a unique feature to accommodate the thermal expansion of the turbine. It consists of supporting the end of the turbine which is not fixed to the foundation by multiple vertical flex legs (Fig. 5.6). This allows the turbine to expand and contract freely without using sliding surfaces. This feature eliminates the possibility of sliding grease plates hanging up and preventing the turbine thermal expansion.

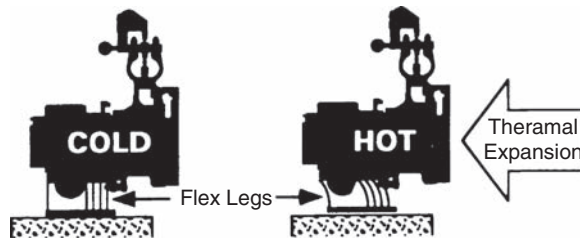


FIGURE 5.6 Down-exhaust steam turbine with front standard mounted on flex legs (movement exaggerated). (Source: Reprinted with permission from General Electric.)

5.4 Design Attributes Enhancing Maintainability

The turbine maintenance activities should be considered in the initial design stages of all the turbine components and systems. However, some features that would enhance the maintainability could affect the operational reliability and efficiency of the machine adversely. For example, the use of fewer bolts would simplify the assembly and disassembly of the turbine. However, this design would probably result in steam leakage through the bolted flanges. Thus, the final design must be an optimization of all aspects, including

1. Reliability of the machine (life) for its intended usage
2. Long-term efficiency
3. Ease (cost/time) of maintenance

5.4.1 Maintainability Features

The following features enhance the maintainability of the steam turbine. They are used by some of the modern steam turbine manufacturers.

Turbine Shells

All turbine shells (Fig. 5.7) have a horizontal joint with bolted flanges. This feature provides quick access to the steampath components. The upper-half of the outer shell can be removed by disassembling only flanged pipe connections. The shell supports shims, and axial-thrust shims are located for easy removal. Most of the internal supervisory instrumentation (e.g., thermocouples) are mounted in the upper-half shells of the turbines. This feature facilitates the replacement or repair of these instrumentation if necessary. Jacking provisions are conveniently located in the turbine shells. Hydraulic jacks or jack screws can easily be used to separate the upper- and lower-half shells. Jacking provisions are also used to remove the axial thrust shim.

The shells of high-temperature turbines are made from cast low-alloy steels. This material has good long-term properties such as ductility and rupture strength. It also permits the repair of in-service cracking by welding.

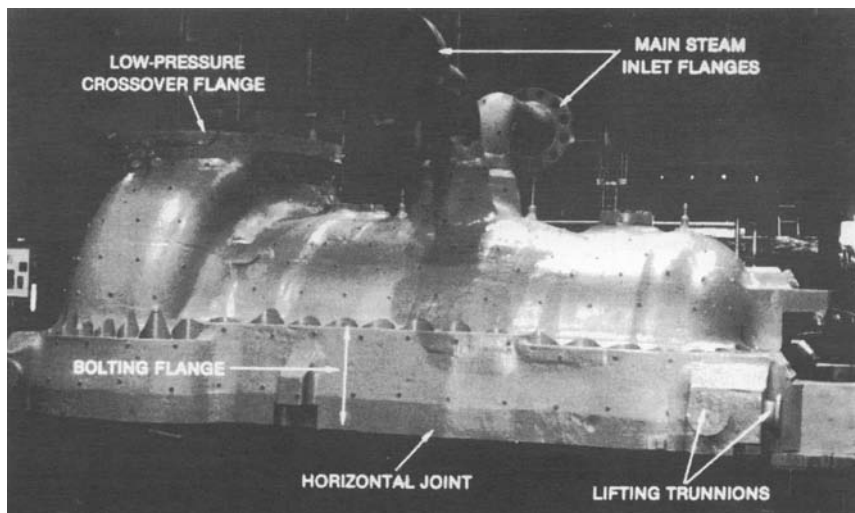


FIGURE 5.7 Typical opposed-flow, high-pressure, and reheat turbine upper outer shell. (Source: Reprinted with permission from General Electric.)

Low-Pressure Turbine Exhaust Hoods and Inner Casings

The following are traditional features conducive to maintenance in low-pressure turbine exhaust hoods (Fig. 5.8):

1. All the major bolting in the hoods is easily accessible.
2. Atmospheric relief diaphragms on condensing steam turbines are readily accessible to facilitate maintenance activities.

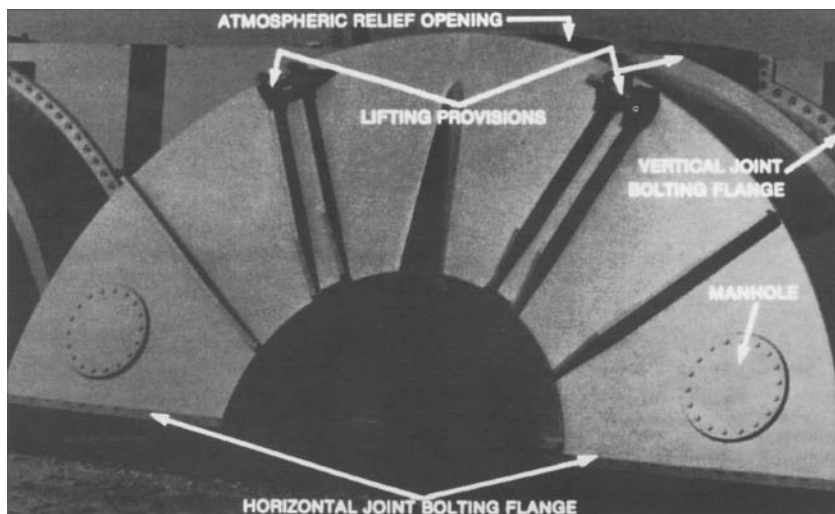


FIGURE 5.8 Low-pressure turbine exhaust hood (upper). (Source: Reprinted with permission from General Electric.)

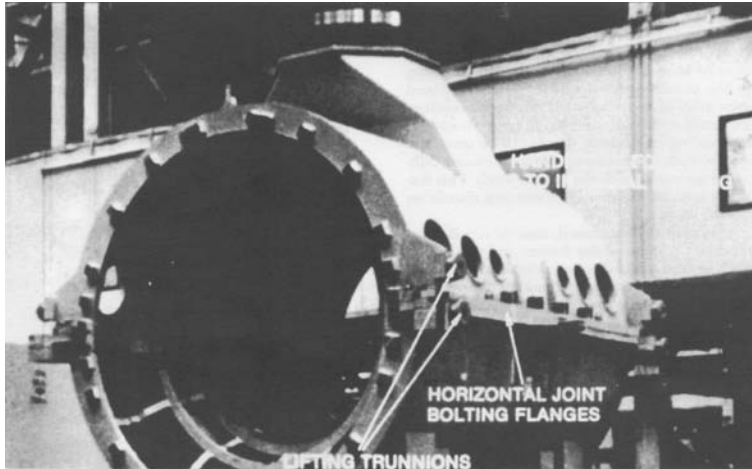


FIGURE 5.9 Low-pressure turbine inner casing. (Source: Reprinted with permission from General Electric.)

3. The hoods of large steam turbines have manholes that permit inspection of the last-stage buckets and structural members.
4. Horizontal joint flanges are designed with easily accessible bolts for the bolted inner casings of steam turbines (Fig. 5.9).
5. The low-pressure turbine exhaust hoods and inner casings are made from low carbon steel. This facilitates the weld repair in case there is structural damage to the components such as cracking and erosion.
6. Borescope access ports are designed to permit visual examination of the latter stages of low-pressure turbines without disassembly (Fig. 5.10). This feature is useful for checking for damage such as moisture erosion, and foreign object damage to steampath components (diaphragm partitions, buckets, etc.).

Rotors

The major causes of steam turbine repair and maintenance work are the erosive particles and impurities in the steam. These erosive particles and impurities in the steam cause deposits and chemical attack on turbine components. Thus, the maintenance and repair work of steam turbines can be reduced by implementing the following:

1. Control of oxide formation and exfoliation in the steam generators
2. Steam chemistry control

Rigid bolted couplings are used to connect most steam turbine rotors to each other and to the generator rotor. Some of these couplings are designed to use “fitted” (small clearance) studs. This feature prevents coupling slippage when extreme abnormal torsional loading is applied to the rotor. Special studs having expandable sleeves have been used on couplings requiring a “fitted” stud arrangement. This design facilitates the removal and reassembly of coupling studs. The use of these special studs eliminates

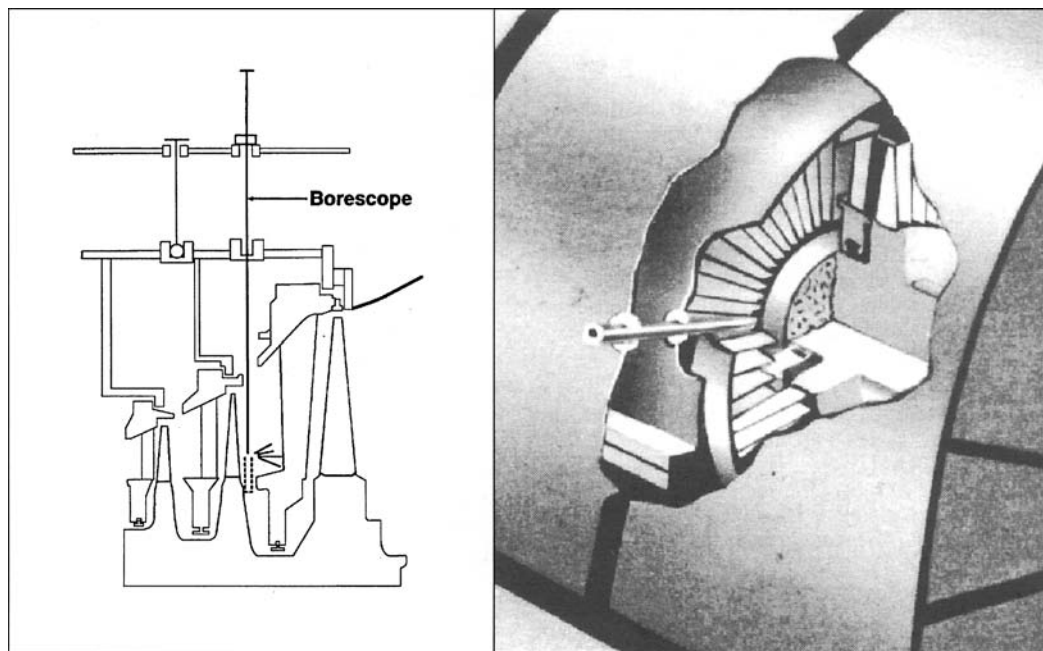


FIGURE 5.10 Borescope access. (Source: Reprinted with permission from General Electric.)

the maintenance difficulties associated with the “fitted” studs. This feature can be retrofitted on existing units.

Nozzle Boxes and Diaphragms

The nozzle boxes and diaphragms (Fig. 5.11) are designed to withstand the harsh steam environment. However, the steam impurities (solid particle and chemical) are the major cause of deterioration in these stationary steampath components. They can be repaired in most cases by weld repairs. Removable alignment and support shims are used with the diaphragms to facilitate their assembly and alignment.

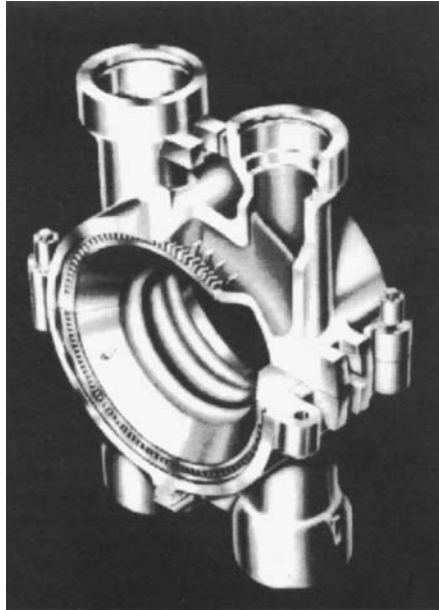
Some manufacturers have developed a good understanding of the mechanics of solid particle erosion (SPE). Methods were developed to reduce the potential to SPE damage to steampath components. For example, protective hard coatings and application procedures are used to increase the overall resistance to steampath SPE damage. These features are available for new turbines. They can also be retrofitted into most existing units. The SPE packages include the following:

1. Stationary turbine parts such as nozzle partitions
2. Rotating parts such as the buckets

Steam Path Sealing Features

The efficiency of the turbine drops due to steam leakage over the bucket tips and through the nozzle diaphragm boxes. These losses can be reduced by installing bucket

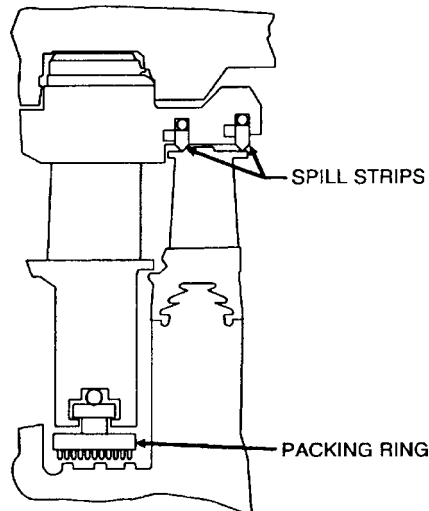
FIGURE 5.11 Fossil unit nozzle box. (Source: Reprinted with permission from General Electric.)



tip spillstrips and diaphragm shaft packing seals that minimize the radial clearances (Fig. 5.12). The tip spillstrips and diaphragm shaft packing seals can easily be removed when repair or replacement is required. The materials used for these component meet the following requirements:

1. Compatibility with the operating environment at each specific stage of the turbine (e.g., steam temperature, pressure, moisture content, etc.).
2. Compatibility with rotor-component materials to minimize heat input and wear if radial rubbing occurs.

FIGURE 5.12 Removable seals. (Source: Reprinted with permission from General Electric.)



The spillstrips and shaft packing are normally spring-loaded in the radial direction. This allows them to deflect radially away from the rotor if rubbing occurs. This feature has the following advantages:

1. Reduction of the radial rubbing force acting on the sealing and rotor components.
2. Minimization of wear and heat generation.

Some manufacturers developed a positive pressure variable clearance packing arrangement to further reduce the probability of radial rubbing on steam shaft packing rings. This arrangement provides the following:

1. Larger clearance during startup.
2. Normal clearance during normal operating conditions.

This design provides the following advantages:

1. Significant reduction in startup operational problems such as rotor vibration.
2. Reduction in the drop in turbine efficiency due to wear of turbine clearances.
3. Reduction in maintenance costs for repair or replacement of damaged packing.

This feature is available as an option on some modern new turbines and as a retrofit to some existing designs.

Primary Steam Valves

The vertical valve stem arrangement facilitates the maintenance of primary steam valves (main stop valves, and control valves).^{*} This is because it allows a straight vertical crane to lift the valve covers and internal parts.

The normal maintenance work on these valves does not involve disturbing the hydraulic actuating system. Some primary steam valves have removable seats. This facilitates the repair work required. Valve stems should be made from wear-resistant and oxidation-resistant materials to improve the reliability and reduce the maintenance work.

Bearings and Lubrication System

The turbine journal bearings provide the following functions:

1. Support the weight of the rotor on a hydrodynamic oil film.
2. Provide dynamic stability to the rotor system.

The function of the single thrust bearing is to axially position the rotor assembly relative to the stationary components (Fig. 5.13).

^{*}These valves are known also as emergency stop valves and governing valves, respectively.

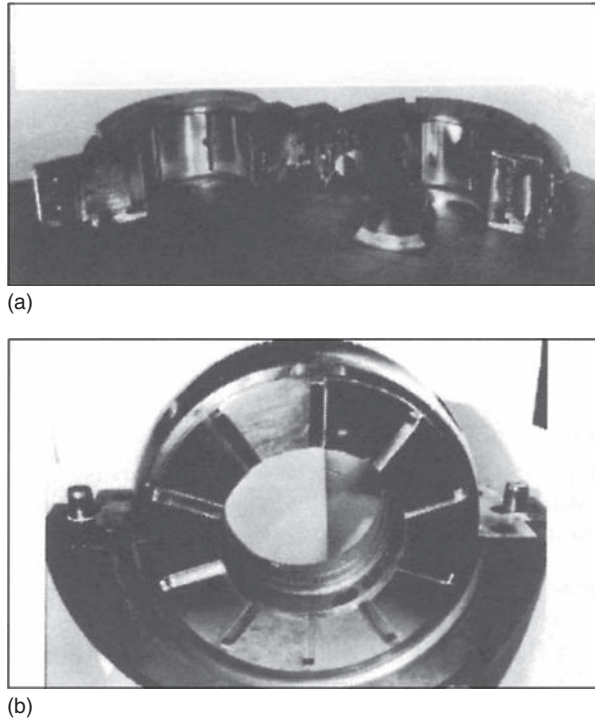


FIGURE 5.13 Tilting-pad journal-bearing (top) and thrust-bearing assembly (bottom).
(Source: Reprinted with permission from General Electric.)

The journal bearings determine the alignment of the rotor system. Realignment of one or more bearings is occasionally necessary to restore the recommended alignment. Support and alignment shims are used to facilitate realignment. The two types of journal bearings are

1. Elliptical fixed geometry
2. Tilt pad

These bearings can be readily removed for maintenance. The tilt pads can easily be removed for repair or replacement if they are damaged.

A jib crane arrangement can be used to facilitate the removal and reassembly of journal bearings that are located under the steam crossover pipe. This optional feature is available on new units. It can also be a retrofit on existing turbines. The following are the benefits of the jib crane arrangement:

1. Reduction in bearing inspection time.
2. Minimization of the potential for bearing damage during handling.
3. The main crane can be used for other purposes.

A booster pump is used with some lubricating systems. It should be flanged-mounted to facilitate its removal for maintenance. Some lube piping system designs do

not have pipe sleeves. This minimizes the locations where foreign material can be entrapped. A redundant full-capacity oil cooler should be provided. This permits the removal of the out-of-service cooler for maintenance with the unit online. A vertical crane lift is used to remove the cooler. Some turbine units have a full-flow lube oil filtration system. This is an optional feature for new units. It can also be retrofitted into existing units. The advantage of this feature is that it ensures the lube oil supply is clean. This reduces significantly the operational problems and maintenance required on units that experienced problems associated with dirty oil (e.g., damage to bearings and mechanical-hydraulic control components). Some systems allow the filter maintenance to be performed online, that is, the filter can be replaced while still maintaining continuous full-flow filtration capability.

Bolting

The bolting materials vary from low-alloy steel to advanced nickel-based alloys for high-temperature applications. Tapered threads are used in highly stressed bolts and studs to obtain a more uniform load distribution over the engaged threads. This feature avoids cracking at the first or second engaged thread. This is the location of high-load concentration in non-tapered bolts. In some applications, the turbine shell bolts are prestretched by heating to obtain the required bolting force.

Turbine-Generator Control and Supervisory Systems

The reliability and maintainability of the turbine-generator control and supervisory systems must be high. This is essential to allow these systems to perform critical functions that include accurate control of speed, load, and extraction steam flows. Some manufacturers have developed electrohydraulic control (EHC) systems that provide reliable protective action while minimizing spurious trips. The following are some of the significant features that enhance the maintainability of EHC systems:

1. Redundant power supplies in the EHC cabinets. This feature permits the maintenance of the power supplies online.
2. Redundant pumps, coolers, and filters in the hydraulic fluid of the governing system. This feature permits the maintenance of these components online.
3. Improved diagnostic testing that can identify the faulty card in any of the triple-redundant controllers.
4. Online replacement capability of the triple modular redundant circuit boards.
5. Employing fully digital equipment to eliminate the drift problems caused by analog controls.

5.4.2 Maintenance Recommendations

The optimum maintenance plan varies from a station to another depending on the variations in fuel costs, labor costs, and cost of shutdown. However, it is highly recommended to follow the maintenance plan provided in the manufacturer's maintenance manual. This plan is normally based on decades of manufacturer's experience. It provides the best overall results in terms of reliability, availability, efficiency, and operating and maintenance costs of the station.

5.5 Cost/Benefit Analysis of High Reliability, Availability, and Maintenance Performance

High reliability, availability, and maintenance (RAM) performance results in a significant increase in profits. Thus, it is essential to select a steam turbine-generator manufacturer that has an established track record of high RAM performance. The following example illustrates the significant increase in profit due to an improvement in forced outage rate.

5.5.1 Reliability, Availability, and Maintainability Value Calculation

Calculate the value of a 1% improvement in forced outage rate if the value of a forced outage per day is \$3,000,000.

Value of a 1% improvement in forced outage rate = $(\$3,000,000/\text{day}) \times (0.01 \text{ improvement}) \times (365 \text{ days/year}) = \$10,950,000$.

Therefore, a slight increase in reliability and availability of a steam turbine generator will result in a significant increase in profits.

5.6 Conclusion

The enhancement of reliability, availability, and maintainability of a steam turbine generator will minimize the total life cycle cost to the owner. Thus, it is essential to select a manufacturer of a steam turbine generator that has the highest reliability in industry.

5.7 Bibliography

Bievenue, R. T., Ruegger, W. A., and Stoll, H. G., "Features Enhancing Reliability and Maintainability of GE Steam Turbines," GER-3741B, 38th GE Turbine State-of-the-Art Technology Seminar, Schenectady, New York, 1994.

This page intentionally left blank

Steam Generators

6.1 Introduction

Fossil and nuclear fuel power generating plants use steam generators. Modern steam generators produce superheated steam at 16.5 to 24 MPa (2400–3500 psia). The *steam generator* involves a combination of the following components:

1. Economizer
2. Boiler
3. Superheater
4. Reheater
5. Air preheater

The steam generator auxiliaries include the following components:

1. Stokers
2. Pulverizers
3. Burners
4. Fans
5. Emission control equipment
6. Stack
7. Ash-handling equipment

The boiler converts the saturated water to saturated steam. Steam generators are classified as either one of the following types:

1. Utility
2. Industrial steam generators

Utility steam generators are used in power generating plants. Modern utility steam generators can either be one of the following kinds:

1. Supercritical water-tube drum type
2. Supercritical once-through type

The latter kind operates at a pressure higher than 24 MPa (3500 psig). The steam critical pressure is 22.1 MPa (3206.18 psig). The subcritical drum type steam generator operates at either of the following:

1. 13 MPa (1900 psig)
2. 18 MPa (2600 psig)

The modern utility steam generators operate at 18 MPa (2600 psig). They generate superheated steam at 540°C (1000°F). They normally have one or two stages of reheat. These units burn pulverized coal or oil. Natural gas is used in some units. However, the use of this fuel has dropped recently. This is due to its high cost. However, natural gas is a relatively pollution-free fuel. The steam flow from modern utility steam generators is 125 to 1250 kg/s (1–10 million lb_m/h). These units are used in power generating plants rated at 125 to 1300 MW.

Industrial steam generators include water-tube pulverized-coal units. The steam generators burn the following:

1. Stoker (lump) coal
2. Oil
3. Natural gas
4. Municipal refuse
5. Process wastes or by-products

Some of these units are used for heat recovery. They recover heat from industrial processes. Industrial steam generators produce normally saturated steam. They do not produce superheated steam. The operating pressure of these units varies up to 10.5 MPa (1500 psig). Their steam flow varies up to 125 kg/s (1 million lb_m/h).

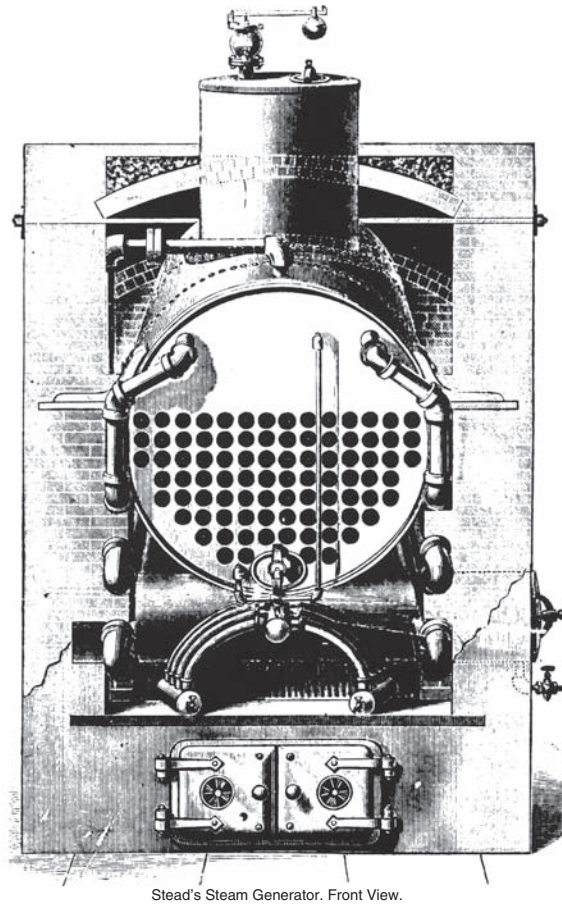
6.2 The Fire-Tube Boiler

Fire-tube boilers have been used to generate steam since the late 18th century. They are normally used in industrial plants to generate steam below 1.8 MPa (250 psig). Their capacity is usually limited to 6.3 kg/s (50,000 lb/h). The fire-tube boiler is a form of the *shell-type boiler*. The shell-type boiler is a vessel that contains water. Hot gases provide heat to a portion of the shell. The *electric boiler* is a form of the shell boiler. Heat is generated in this boiler by electrodes immersed in the water. The *accumulator* is an evolved form of the shell-type boiler. Steam passes through the tubes of this boiler. The shell of these boilers is not exposed to heat. Hot gases flow through the tubes of a fire-tube boiler (Figs. 6.1 and 6.2). The efficiency of this boiler can reach 70%.

The following are the types of fire-tube boilers:

1. The fire box
2. The scotch marine

The furnace on firebox and the fire tubes are located inside the shell in a *firebox boiler*. Figure 6.3 illustrates a *scotch marine boiler*. Combustion in this boiler is occurring inside one or more cylinders. These cylinders are normally located near the bottom of



Stead's Steam Generator. Front View.

FIGURE 6.1 The stead fire-tube steam generator.

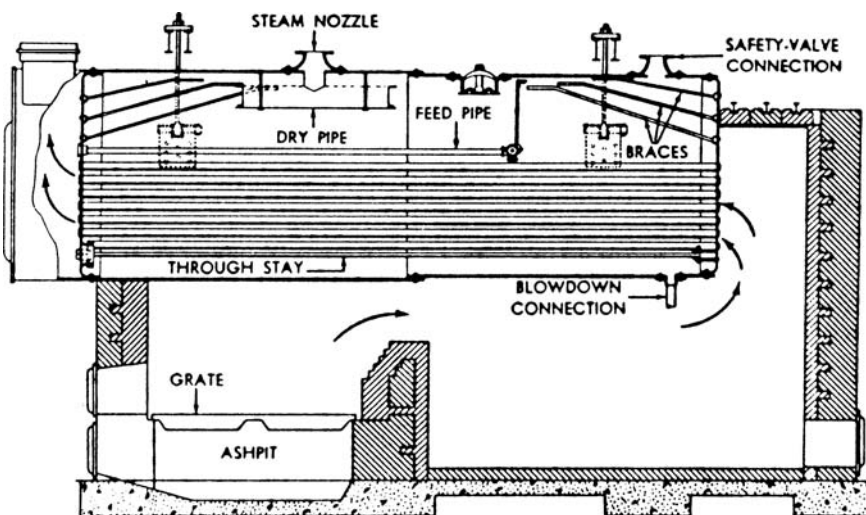


FIGURE 6.2 Schematic of an early horizontal-tube fire-tube boiler.

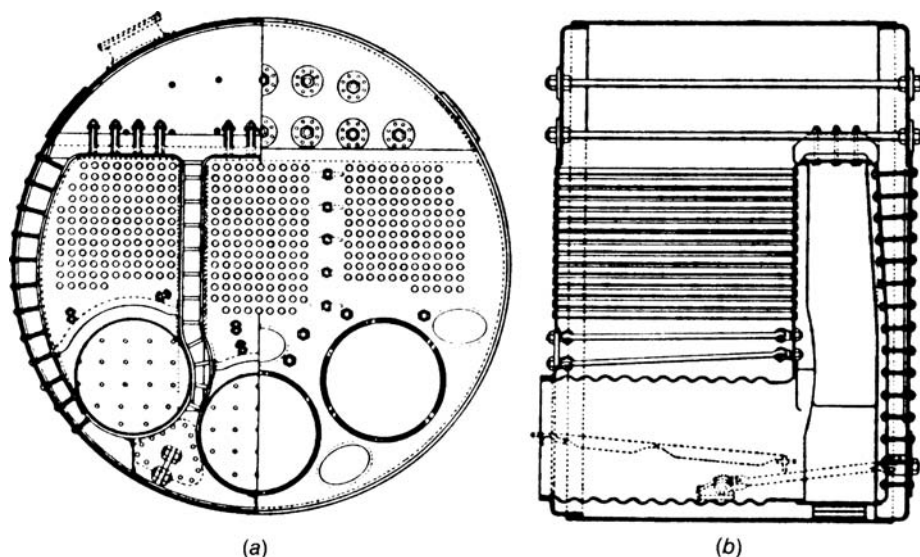


FIGURE 6.3 Schematic of an early scotch marine boiler.

the main shell, inside the boiler. The gases leave the cylinders and return through the fire tubes. The gases are then discharged from the stack.

Fire-tube boilers are capable of generating saturated steam. They are limited to small capacities and low steam pressure.

6.3 The Water-Tube Boiler

The water-tube boiler is the forerunner of the modern steam generator. This design was developed by Babcock and Wilcox in 1867. The early water-tube boilers have similar appearance to the fire-tube boiler. However, the higher-pressure water and steam are inside the tubes in a water-tube boiler. The combustion gases are outside the tubes in this design.

6.3.1 The Straight-Tube Boiler

Figure 6.4 illustrates a *straight-tube boiler*. This design was the first water-tube boiler. The tubes are 3 to 4 in OD, inclined at about 15° and staggered with 7- to 8-in spacings. The tubes are connected to two vertical headers. The first header is the *downcomer*, or *downtake*. This header supplies saturated water to the tubes. Partial boiling occurs in the tubes. The second header is a *riser*, or *uptake*. This header receives the water-steam mixture. Natural circulation occurs clockwise because the water density in the downcomer is larger than the two-phase density in the riser. The two-phase mixture enters an upper drum. This drum is either arranged parallel to the tubes (the *longitudinal drum*, Fig. 6.4a) or perpendicular to them (the *cross drum*, Fig. 6.4b). These drums supplied saturated steam to the superheater. This flow was supplied through a steam separator within the

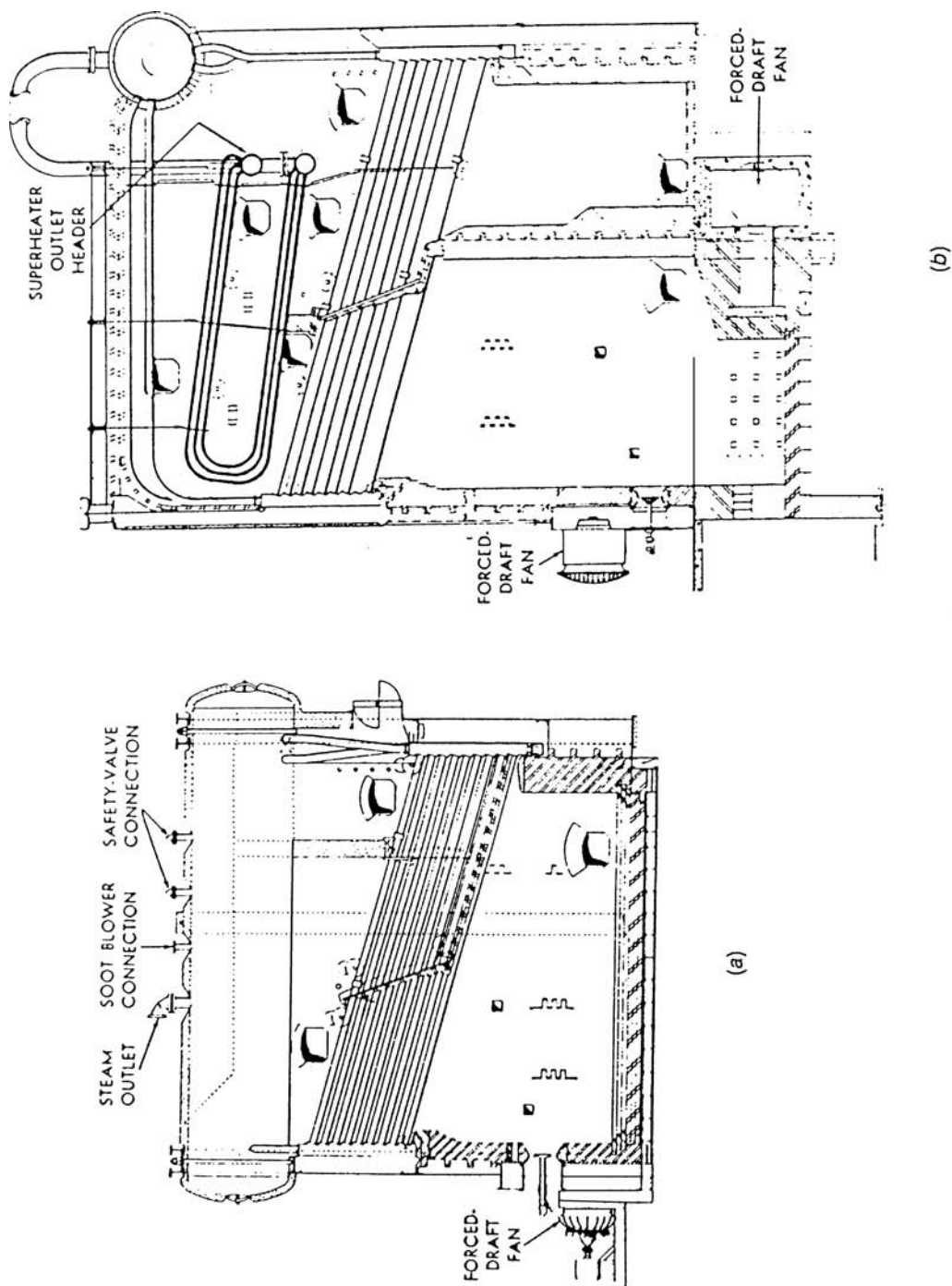


FIGURE 6.4 Early water-tube boilers: (a) longitudinal and (b) cross-drum.

drum. The *mud drum* is connected to the lower end of the downcomer. This drum collected sediments from the water.

Longitudinal-drum boilers were limited to 1.2 to 2.3 MPa (175–340 psia), and steam flow of 0.63 to 10 kg/s (5000–80,000 lb/h). Cross-drum boilers were limited to 1.2 to 10 MPa (175–1465 psig) and steam flow of 0.63 to 63 kg/s (5000–500,000 lb/h).

6.3.2 The Bent-Tube Boiler

The *bent-tube boiler* used bent tubes between several drums and headers. Figure 6.5 illustrates the four-drum Sterling boiler. The three top drums contain a two-phase mixture. The drum at the bottom was filled with water. This drum was called the mud drum. The furnace on this boiler is located at the bottom right. The combustion gases flowed through the first bank of tubes, the superheater, the remaining banks of tubes, and the economizer. The feedwater flowed from the economizer to the rear steam drum. The water then flowed through the downcomer tubes to the lower drum and then through the riser tubes to the center and front drums. The steam and water regions of the upper three drums were interconnected at the top and bottom. The OD of the tubes

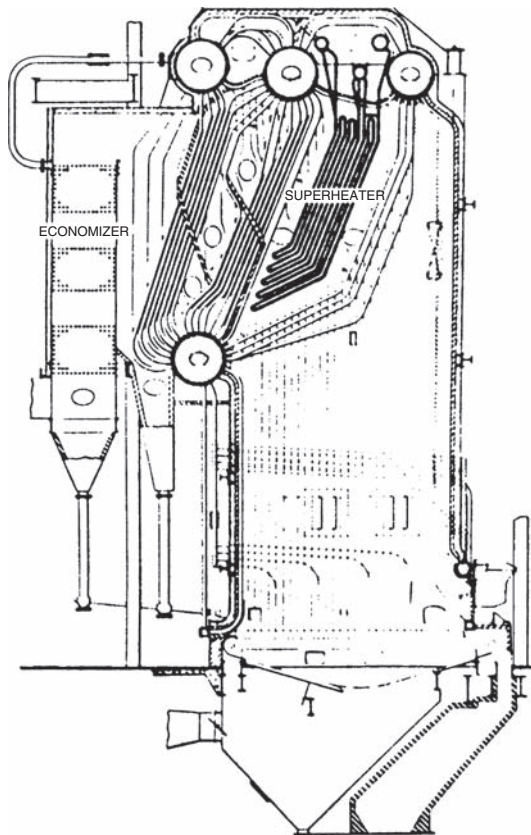


FIGURE 6.5 A four-drum early sterling boiler.

was 7.64 to 8.8 cm (3–3.5 in). These tubes were spaced 12.7 to 17.8 cm (5–7 in) centers. Recent designs of this boiler involved cooled furnace walls. The cooling of these walls was done by lining their interior with tubes carrying water (Fig. 6.5). This design has the following advantages:

1. Increase the heat-absorbing surface
2. Protection of the refractory lining of the walls from excessive temperature

This resulted in higher rates of combustion and increase in the steam-flow rates. The following are the Sterling boiler capabilities:

1. Handling rapidly varying loads
2. Operating in plants having water quality below standards
3. Adapting to various fuels

This boiler was used in stationary and marine applications.

6.4 The Water-Tube Boiler: Recent Developments

The water-cooled furnace walls design is called *water walls*. This design has led to the integration of furnace, economizer, boiler, superheater, reheater, and air preheater in this steam generator. Water cooling is also used for the following components:

1. Superheater
2. Boiler
3. Economizer
4. Screens, dividing walls, etc.

Figure 6.6 illustrates a modern steam generator. Water flows through insulated downcomer from the steam drum to a header. The downcomer is located outside the furnace. The header connects the water tubes. These tubes line the furnace walls. They act as risers. The water flows through the downcomer and water tubes by natural circulation.

The flue gases preheat the atmospheric air at the discharge of the forced-draft (FD) fan (to about 150°C, 300°F) before they are discharged to atmosphere. The temperature in the furnace can exceed 1650°C (3000°F). The *induced-draft (ID) fan* draws the flue gases from the steam generator. The discharge temperature (150°C, 300°F) of the flue gases represents an availability loss to the plant. However, this is deemed acceptable for the following reasons:

1. To keep the gas temperature above the dew point of the water vapor in the gases. This is required to prevent condensation of the water vapor on the stack. The condensation of this water vapor generates acids that can corrode the metal of the stack.
2. The buoyancy of the flue gases must be adequate to allow them to rise in a high plume above the stack. This is required for proper atmospheric dispersion.

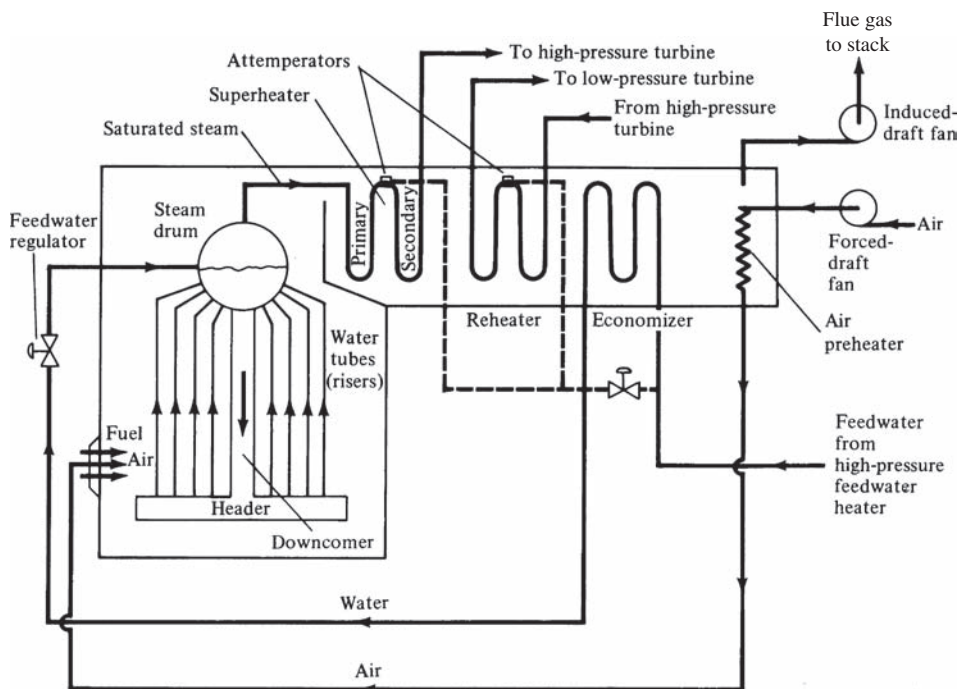
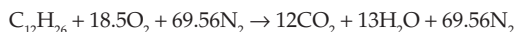


FIGURE 6.6 Schematic flow diagram of a modern steam generator.

Example 6.1 A steam generator burns fuel oil with 20% excess air. The fuel oil may be represented by $C_{12}H_{26}$. Determine the minimum stack temperature needed to avoid condensation. Assume the flue gas pressure is leaving the air preheater at 45 psia.

Solution Recalling that there are 3.76 mol N_2 /mol O_2 in atmospheric air, the combustion equation for stoichiometric (chemically correct mixture) is



With 20% excess air



The partial pressure of any component in a gas mixture is equal to the total pressure times the mole fraction of that component. Thus partial pressure of H_2O in products is

$$45 \times \frac{13}{12 + 13 + 3.7 + 83.472} = 5.215 \text{ psia}$$

This corresponds to a saturation temperature of 164°F. The average temperature of the gases is kept much higher to avoid local cool spots that might cause condensation.

6.4.1 The Boiler Walls

Figure 6.7 illustrates the following types of water tubes:

1. Tangent *bare* tubes
2. *Embedded* in the refractory

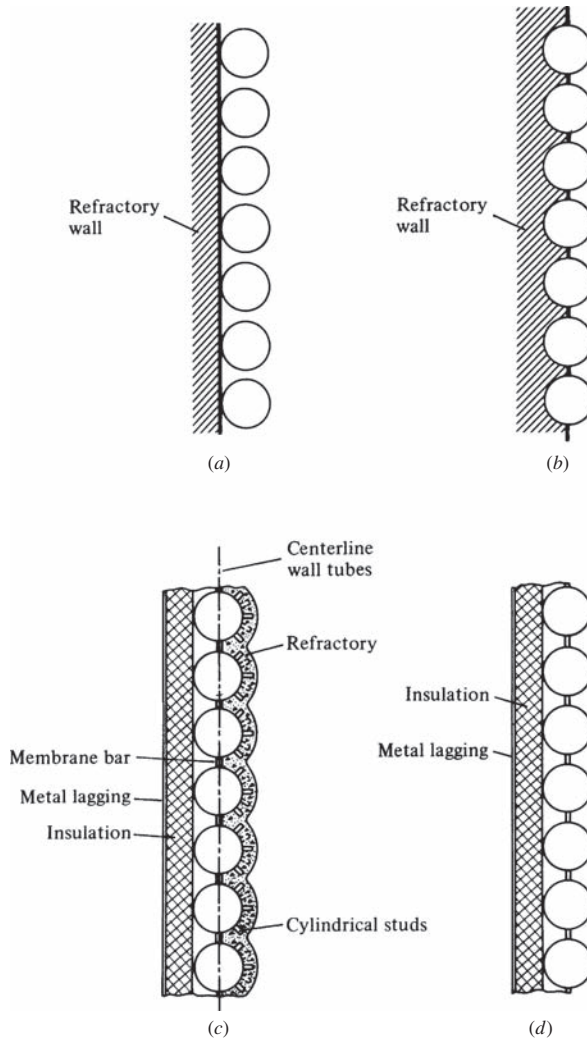


FIGURE 6.7 Plan views of different types of water tubes.

3. *Studded tubes*

4. *Membrane tubes*

The membrane tubes are connected by bars or membranes welded to the tubes at their center lines. The membranes have the following functions:

1. They act as fins to increase the heat transfer.
2. Provide a rigid construction to the furnace.

Additional casings are not required inside the steam generator in this design.

6.4.2 The Radiant Boiler

The combustion gases provide heat to the water walls by radiation and convection. The heat is transferred mainly by radiation in a *radiant boiler*. Combustion gases may be *luminous* or *nonluminous*. The luminous ones emit all wavelengths. Thus, they emit strong visible radiation if there are particulates such as soot particles during combustion. Coal and oil produce luminous combustion gases. Nonluminous combustion gases burn without particulates. Gaseous fuels produce nonluminous combustion gases. There are no truly nonluminous combustion gases. This is because heavier gases in the combustion products such as CO_2 , H_2O , SO_2 , ammonia, and sulphur dioxide are selective radiators. These gases emit and absorb radiation in certain wavelengths. Most of this radiation is outside the visible range. The portion of radiation within the visible range is small. However, it gives the combustion a green-blue appearance.

Radiant energy emitted by combustion gases depends on the following:

1. Gas temperature (to the fourth power)
2. Partial pressure of the individual constituent radiating gases
3. Shape and size of the gases
4. Proximity of the gases to the absorbing body
5. Temperature of the absorbing body (to the fourth power)

Heat is transferred to the water walls by conduction through the membranes and tube walls. This heat is then transferred to the two-phase mixture inside the tubes by forced-convection nucleate boiling.

Radiant boilers are used in power stations to use coal or lignite for pulverized or cyclone furnace applications, oil, or natural gas. They normally generate steam at 540°C (1000°F), capacities up to 1260 kg/s ($10 \times 10^6 \text{ lb/h}$) and pressures of 12.5 to 17 MPa (1800–2500 psig).

6.5 Water Circulation

Water circulates from the steam drum to a bottom header through downcomer pipes. The water then flows upward through water tubes back to the steam drum. These tubes act as risers. The water boils partially in the water tubes. Full boiling to dry steam in the tubes (e.g., 100% quality) is avoided. This is because it would lead to tube burnout or failure. This would occur due to the dry out conditions caused by departure from nucleate boiling (DNB). The saturated water in the downcomers is denser than the two-phase mixture in the risers. Natural circulation flow depends on the difference between these two densities and the elevation of the drum above the bottom headers. The majority of large steam-generator boilers have sufficient natural circulation driving force. These units are called *natural-circulation boilers*. Some boilers pump the single-phase flow. These units are called *controlled- or forced-circulation boilers*. Forced circulation is favored over natural circulation for pressures higher than 16 MPa (2300 psig). The difference between the water and steam density decreases rapidly at these pressures (this difference reaches zero at the critical point, 22.1 MPa, 3208 psig). This reduced the natural circulation flow. A safety margin was required to prevent tube failure and burnout due to DNB, that is, reaching the critical heat flux (CHF). Low flow in a tube is also a major concern in water-cooled nuclear-reactor design. This safety margin was supplied by a pump.

FIGURE 6.8 A natural circulation loop.

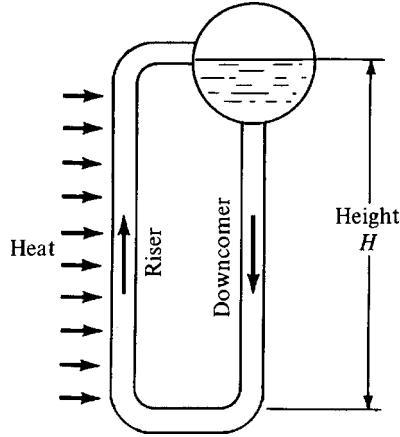


Figure 6.8 illustrates a natural circulation loop. The *driving pressure* caused by natural circulation is given by:

$$\Delta p_d = (\rho_{dc} - \bar{\rho}_r) H \frac{g}{g_c}$$

where Δp_d = driving pressure, lb_f/ft^2 or N/m^2 (Pa)

ρ_{dc} = density of water in the downcomer, nearly saturated at the system pressure, lb_m/ft^3 or kg/m^3

$\bar{\rho}_r$ = average density of steam-water mixture in the riser, lb_m/ft^3 or kg/m^3

H = height of drum-water level above bottom header, ft or m

g = gravitational acceleration ft/s^2 or m/s^2

g_c = conversion factor $32.2 \text{ lb}_m \cdot \text{ft}/(\text{lb}_f \cdot \text{s}^2)$ or $1 \text{ kg} \cdot \text{m}/(\text{N} \cdot \text{s}^2)$

The ρ_r is a function of the *void fraction* in the riser. The void fraction, α of a two-phase mixture is a volumetric quality given by:

$$\alpha = \frac{\text{volume of vapor}}{\text{volume of vapor} + \text{liquid}}$$

The mass quality is x . The following expression relates α and x :

$$\alpha = \frac{1}{1 + [(1 - x)/x]\psi}$$

$$x = \frac{1}{1 + [(1 - \alpha)/\alpha] \frac{1}{\psi}}$$

$$\psi = \frac{v_f}{v_g} S$$

where v_f and v_g are the specific volumes of the saturated liquid and vapor at system pressure, respectively. S is the *slip ratio* of the two phases. The vapor moves faster than the liquid. S is given by:

$$S = \frac{\bar{V}_{s,g}}{\bar{V}_{s,f}}$$

where $\bar{V}_{s,g}$ and $\bar{V}_{s,f}$ are the average vapor and average liquid velocities at any cross-section on the riser. S was found experimentally to be between 1 and 10. S approaches 1 at high pressures. S has normally a constant value along the system.

6.6 The Steam Drum

All modern steam generators except the once-through types have a steam drum. The saturated steam is separated from the boiling water in the steam drum. The drum may also be used for the following purposes:

1. Chemical water treatment
2. Blowdown to reduce the amount of solids in the water

The drum must also accommodate the changes in the level of water. These changes occur due to load variations. The steam generator control system must perform the following functions:

1. Prevent low-level of water in the steam generator. This is required to prevent uncovering a portion of the tubes. Cracks could occur in these tubes due to thermal stresses (the steam is a poor coolant compared with water for the tubes).
2. Prevent carryover of water toward the superheater. This would cause deposit build-up of entrained solids on the superheater tubes. The temperature of these tubes will increase significantly. This will result in their distortion or burnout. Carryover of solids with the steam can also cause deposit build-up on the turbine blades. Silica deposits are the most troublesome. This is because they are not easily removed by water washing.

The main function of the steam-drum is separation of the steam from the boiling water. *Gravity separation* (Fig. 6.9a) is the simplest method. The steam separates naturally from the water surface if its velocity is below 0.9 m/s (3 ft/s). The water droplets and the solids they carry (carryover) will not get entrained with the steam in this case. Gravity separation is used normally in low-pressure applications. This is because the difference between the density of the water and steam drops as the pressure increases. This minimizes gravity separation.

Mechanical separation assists or supplements gravity separation in modern high-capacity, high-pressure boilers. This process is performed in two steps: primary and secondary. *Primary separation* involves the following:

1. Removal of the water from the steam
2. Prevention of steam carryunder with the recirculating water to downcomers and risers

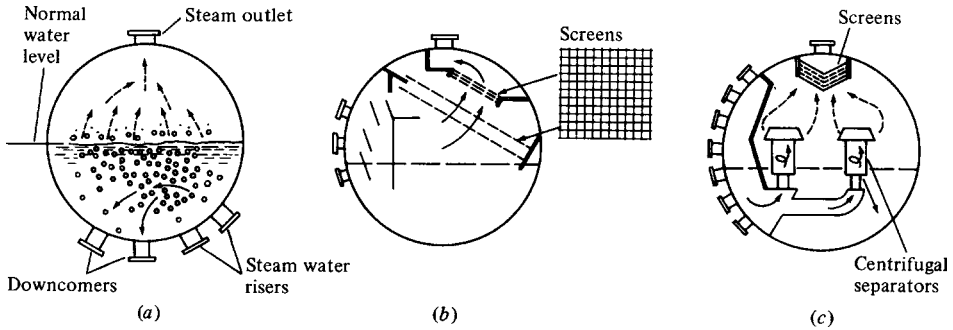


FIGURE 6.9 Steam drum separation: (a) gravity, (b) mechanical primary (baffles) and secondary (screen), (c) centrifugal.

Primary separation is accomplished by installing baffles, screens, bent, or corrugated plates in the drums (Fig. 6.9b). *Secondary separation* is also known as *steam scrubbing* or *drying*. This process involves the removal of remaining mist or fine droplets and the solids carried in them from the steam. This process generates pure, or “dry and saturated” steam. Secondary separation is performed by centrifugal separators.

Baffle plates change the direction of the steam flow (Fig. 6.9b). They act as primary separators. They assist gravity separation. The baffles act as impact plate. The water drains off these plates. The screens are made from wire mesh. They intercept fine droplets. The accumulating droplets fall back on the water. Bent or corrugated plates maximize gravity separation.

Centrifugal separators (Fig. 6.9c) are used at high pressures. This is due to the drop in the density differential between water and steam. The centrifugal forces generated by these separators are much larger than gravity forces. These devices are also known as *cyclone* or *turbo separators*. The mixture of steam and water enters the separator. Guide vanes impart a spinning motion to the mixture inside the separator. The heavier water droplets impinge on the separator wall. They are discharged through an outer concentric cylinder below the water surface. Screens are installed under the drum exit. They dry the steam further.

The steam drums used in utilities vary in length up to more than 30 m (100 ft), in diameter up to more than 5 m (17 ft), and in mass up to a few hundred tons. They have more than 30 outlet nozzles and many risers and downcomer nozzles. Large steam drums are normally constructed from cylindrical sections. These sections are called *courses*. The courses are welded together. Two hemispherical heads are welded at the ends.

6.7 Superheaters and Reheaters

The OD of the tubes used in superheaters and reheaters of utility steam generators is 5 to 7.6 cm (2–3 in). The OD of steam generators used in marine services is 2.5 to 3.8 cm (1–1.5 in). Fins are avoided on the outside surface of these tubes. This is because of the following reasons:

1. Increase in thermal stress
2. Increase the difficulty of cleaning the outside surface of these tubes

These tubes are made from carbon steel if the temperature is below 454°C (850°F). The tubes of modern superheaters and reheaters are made from a special high-strength alloy if the temperature is around 538°C (1000°F). This material is chosen due to its strength and corrosion-resistant properties.

6.7.1 Convection Superheater

The heat is transferred mainly between the combustion gases and the tubes of early superheater designs by convection. Therefore, superheaters became known as *convection superheaters*. The fuels and airflow increase when the demand for steam increases. Thus, the combustion-gas flow increases. This increases the convective heat-transfer coefficients inside and outside the tubes. The overall heat-transfer coefficient between the gas and steam increases faster than the increase in the mass-flow rate of the steam (the combustion temperature does not change significantly when the load changes). Thus, the temperature of the steam increases with the load (Fig. 6.10).

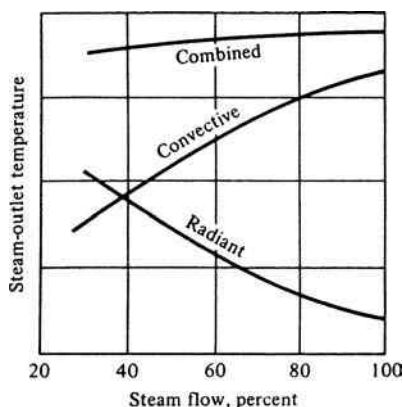
6.7.2 Radiant Superheater

The design of superheaters improved to increase the heat absorption. The following features were implemented:

1. The superheater tubes were placed in view of the combustion flames (near higher temperature).
2. The steam-flow velocity was increased. This was done to increase the overall heat-transfer coefficients.
3. The tube metallurgy was improved to withstand higher temperature.

Most of the heat transfer between the hot gases and flame, and the two outer walls is accomplished by radiation in this design. This type has become known as a *radiant superheater*. The heat transfer by radiation is proportional to $T_f^4 - T_w^4$, where, T_f is the flame absolute temperature, and T_w is the wall absolute temperature. Since T_f is much higher than T_w , this heat transfer is mainly proportional to T_f^4 . Since T_f is not dependent

FIGURE 6.10 Exit-temperature response of convective, radiant, and combined (in-series) superheaters.



on the load, the heat transfer per unit mass flow of steam drops when the steam flow increases. Thus, the exit steam temperature drops when the steam flow increases due to an increase in load (Fig. 6.10). Convection superheaters experience the opposite effect.

Reheaters and superheaters have similar design considerations. However, the following are the differences between them:

1. The overall temperatures in reheaters are lower than superheaters. However, the steam outlet temperatures from both of them are about the same.
2. The pressures in the reheater are around 20 to 25% of those in the superheaters.

Therefore, a lower-grade steel alloy is used in reheaters because the stresses are lower. Low-temperature steam generators use convection superheaters only. High-temperature applications employ radiant and convection superheaters and reheaters. Radiant units consist of flat panels or platen sections with spacing of several feet. This is done to permit radiation through. Sections having narrower spacing are mounted downstream of the flat panels. These sections permit radiation and convection. There are three kinds of these sections: *pendant*, *inverted*, and *horizontal*.

The tubes of *pendant-type* superheaters and reheaters are hung from above (Fig. 6.11a). The main disadvantage for this design is flow blockage by condensed steam after a shutdown. The water that accumulates in the bottom should be purged before starting the unit. Figure 6.11b illustrates an *inverted-type* unit. The tubes of this design are supported from below. This design has the advantage of proper drainage of the condensed steam. Figure 6.11c illustrates a *horizontal-type* unit. The tubes of this design are supported horizontally in a vertical gas duct. The duct is parallel to the furnace. The tubes do not view the flame directly. Thus, this is a convection-type unit. This design has proper drainage. Figure 6.12 illustrates a typical arrangement of superheaters, reheaters, economizers, and air preheater in a drum-type steam generator.

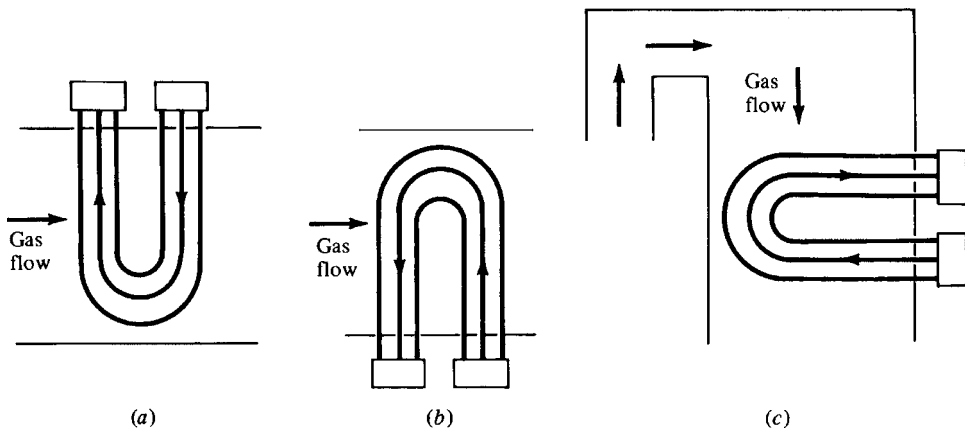


FIGURE 6.11 Schematic diagram showing (a) pendant, (b) inverted, and (c) horizontal superheaters and reheaters.

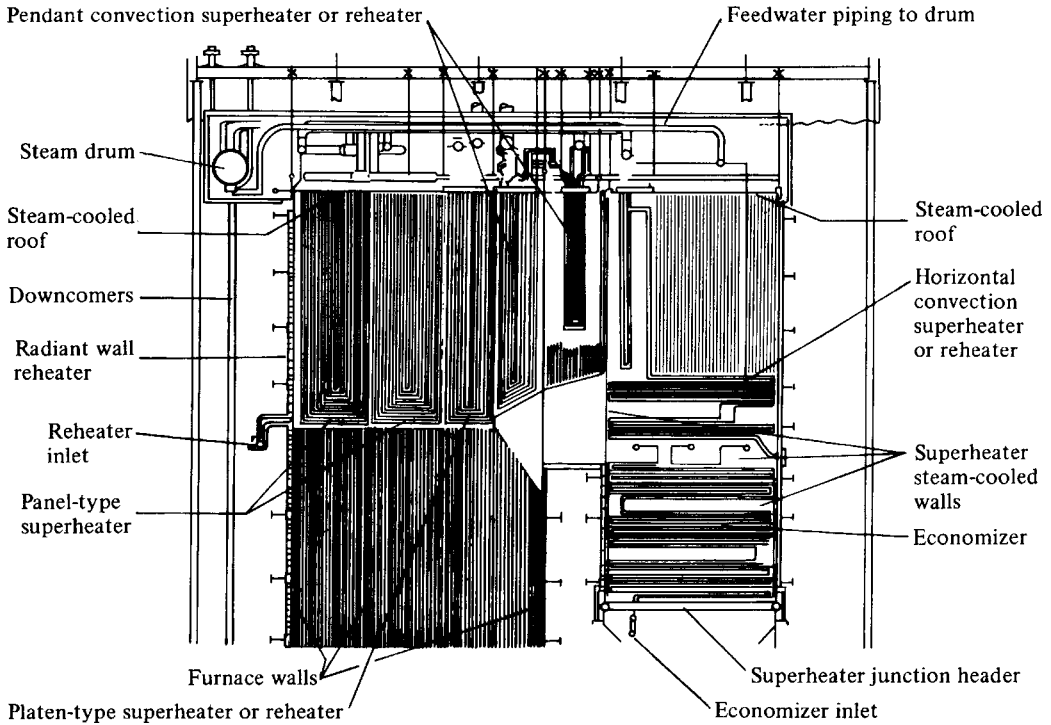


FIGURE 6.12 Superheater, reheater, economizer, and air preheater arrangements in a drum-type steam generator with cyclone furnace. (Source: Reprinted with permission from Babcock and Wilcox Company.)

6.8 Once-Through Boilers

The *once-through* steam generator is also known as *forced-circulation*, *Benson*, or *universal pressure* boiler. The feedwater flows through the economizer, furnace walls, and superheater sections in a once-through steam generator (Fig. 6.13b). This is in contrast to drum-type steam generators (Fig. 6.13a). The once-through steam generator does not employ a steam drum to separate saturated steam from boiling water. Water circulation does not also occur in a once-through steam generator. This design (Fig. 6.14) requires very high purity feedwater.

The once-through steam generator is used for all temperatures and pressures. However, it is economically suitable for large sizes and pressures in the high subcritical and supercritical range (> 22.1 MPa, 3208 psia for steam). In fact, this is the only design suited for supercritical-pressure applications. This is because separation between saturated water and saturated steam in a drum is not possible or necessary (the latent heat of vaporization at and beyond the critical pressure is equal to zero. There is no difference between the liquid and vapour phase at this stage). This design is economical for steam pressure of 13.8 to 27.6 MPa (2000–4000 psig) and flow of 3.8 to 1260 kg/s (30,000–1,000,000 lb_m/h).

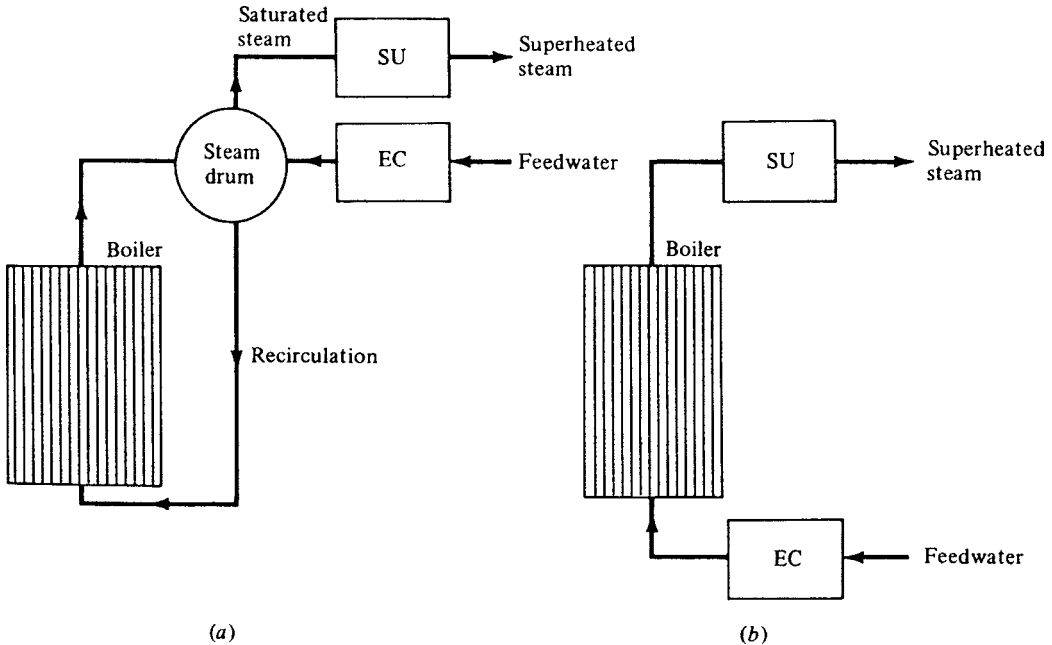


FIGURE 6.13 Schematic flow diagrams of (a) drum type and (b) once-through steam generators. SU = superheater, EC = economizer.

Sulzer Brothers, Ltd., of Switzerland developed the once-through steam generator in the late 1920s. The initial design was for steam pressure from 8.4 to 16.7 MPa (1200–2400 psig). However, many supercritical-pressure once-through steam generators were built with double reheat during the 1940s and 1950s. Some of these units had advanced steam pressures of 31 to 34.5 MPa (4500–5000 psig) and steam temperatures of 620 to 650°C (1150–1200°F). Figure 6.15 illustrates the variation of heat rate with steam temperature.

The capital cost for a supercritical steam generator is comparable to a drum-type subcritical unit of the same capacity. However, the efficiency of the plant having the once-through steam generator is higher.

6.9 Economizers

The *economizer* (EC) is a heat exchanger. The gases leaving the last superheater or reheater enter the economizer. These gases increase the temperature of the water leaving the highest-pressure feedwater heater to saturation. The steam leaves the economizer at 370 to 540°C (700–1000°F).

The gases are kept above the dew point in modern economizers. This is accomplished by increasing the temperature of the feedwater entering the economizer. This reduces the rate of corrosion and fouling. The feedwater is allowed to boil at the outlet

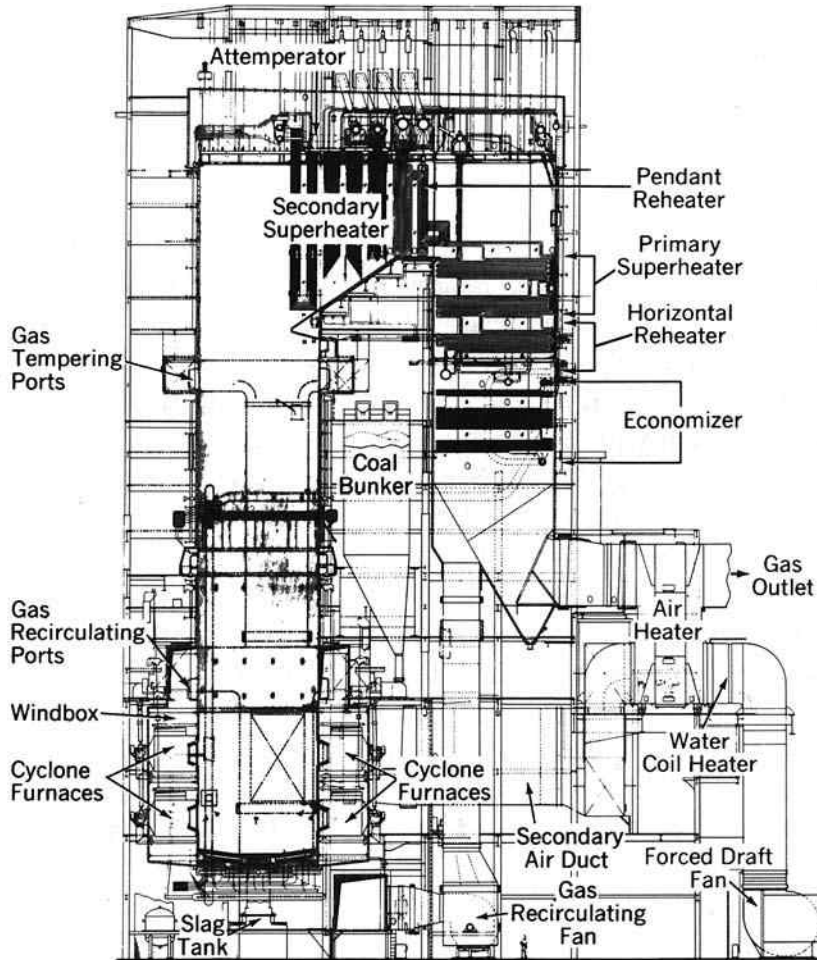


FIGURE 6.14 A once-through steam generator with pulverized coal furnace. (Source: Reprinted with permission from Babcock and Wilcox Company.)

of modern economizers. A quality of 20% is achieved in some designs. The OD of the economizer's tubes is 4.5 to 7 cm (1.75–2.75 in).

6.10 Air Preheaters

Air preheaters are also known as air heaters. This equipment recovers some of the heat remaining in the flue gases. They receive gases at 315 to 427°C (600–800°F). The temperature of these gases drops to 135 to 176°C (273–350°F) at the outlet of the air preheaters. This temperature range was chosen for the following reasons:

1. To avoid condensation on the stack
2. To allow adequate dispersion in the atmosphere

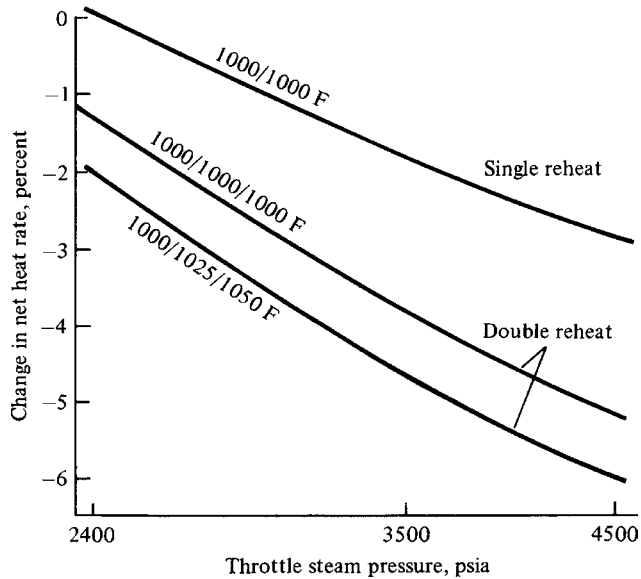


FIGURE 6.15 Effect of steam conditions on heat rate.

The air is heated from the outlet temperature of the forced-draft fan (near atmospheric) to 260 to 350°C (500–650°F). This preheating of the air reduces the fuel consumption. The increase in plant efficiency (or reduction in fuel consumption) is almost proportional to the increase in air temperature in the preheater. The following are typical increases in efficiency:

1. 4% for 93°C (200°F) increase in air temperature
2. 11% for 260°C (500°F) increase in air temperature

Preheated air is required for the operation of pulverized-coal furnaces. Air in the 150 to 315°C (300–600°F) range is needed to dry a fuel. Air also carries the fuel to the furnace. Europeans developed air preheaters like economizers. The first unit built in the United States was a flat-plate heat exchanger. Adjacent steel plates formed alternate air and gas passages in this design. However, modern air preheaters use high air and gas pressures. These units use tubular and regenerative designs. These features are required to withstand the higher pressures. The following are the general types of air preheaters:

1. Recuperative
2. Regenerative

The heat is transferred directly from the hot gases to the air in *recuperative air preheaters*. This is done across the heat exchange surface. Figure 6.16 illustrates a vertical preheater. The tubes are rolled (expanded) into the top and bottom tube sheets.

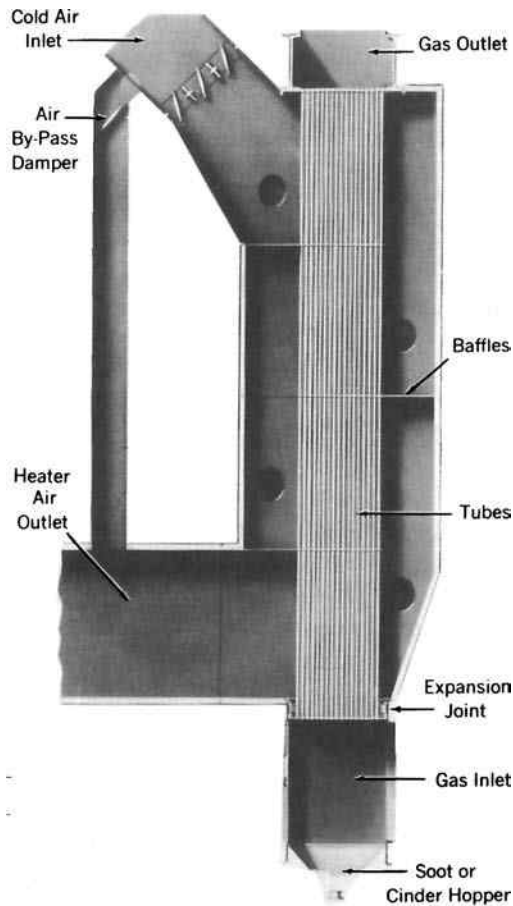


FIGURE 6.16 A tubular counterflow air preheater. Air bypass is used to control metal temperatures at air inlet end. (Source: Reprinted with permission from Babcock and Wilcox Company.)

A bellows-type expansion unit provides thermal expansion. The soot and dust that deposit inside the tubes are collected in a hopper. The tube sizes vary from 3.8 to 10.2 cm (1.5–4 in) in OD.

The heat is transferred from the hot flue gases to a heat-storage medium, then to the air in *regenerative air preheaters*. The most common is the *Ljungstrom* preheater. This is a rotary preheater developed in Europe in 1920. It consists of a rotor driven by an electric motor through a gearbox. The rotor rotates at 1 to 3 r/min. The rotor has between 12 and 24 sectors (Fig. 6.17). The sectors are filled with a heating service made of steel sheets. The sheets constitute the heat storage medium of the preheater. A stationary seal covers two opposite sectors. The hot gases flow in half of the remaining sectors, the air flow in the other half. The rotating sectors are heated by the hot gas. They store the heat. They release this heat to the air when they enter the air zone.

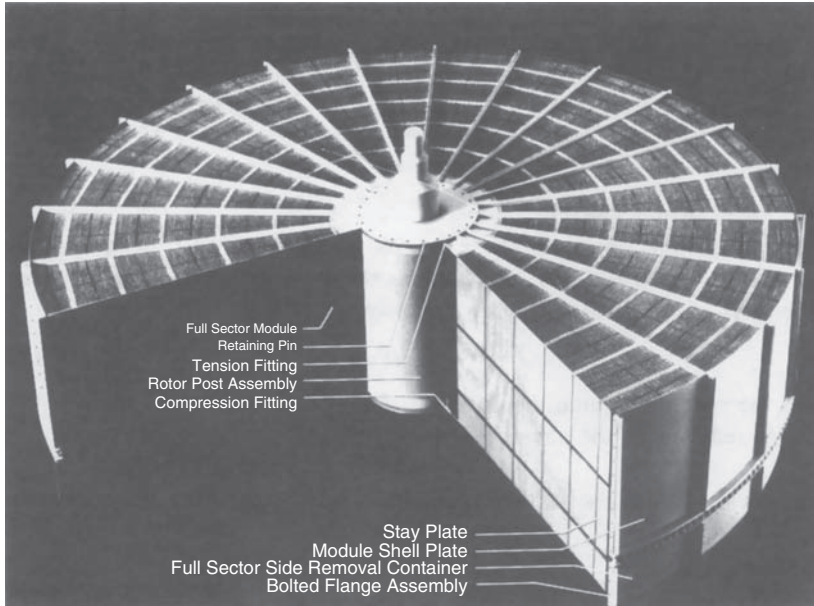


FIGURE 6.17 A vertical shaft Ljungstrom air preheater. (Courtesy of Combustion Engineering, Inc.)

6.11 Fans

Steam generators use *forced-draft* (FD) and *induced-draft* (ID) fans.

Forced-draft fans are located at the air entrance to the air preheater. *Induced-draft fans* are placed in the gas stream between the air preheater and the stack. They discharge the gas at atmospheric pressure. They also place the entire system under negative pressure. Their power requirement is greater than forced-draft fans. This is because they handle hot gases (the power requirement increases with the gas temperature). They also must cope with corrosive combustion products and ash. They are rarely used alone.

The FD fans push atmospheric air through the air preheater into the furnace. The ID fans pull the combustion gases from the furnace through the superheater, economizer, and air preheater. They push these gases into the stack. They are built into the stack base in some applications. The stack adds a natural driving pressure of its own (refer to Sec. 6.13). This is due to its height. The furnace operates with *balanced draft* in these applications. This means that it operates at atmospheric pressure approximately. Actually, the furnace operates at a slightly negative pressure. This is done to ensure inward leakages.

Pressurized firing steam generators are used with low-ash fuels such as gas and oil. Coal-fired generators are normally balanced-draft firing.

Electric power stations use the largest fans. The capacities of these fans exceed $700 \text{ m}^3/\text{s}$ ($1.5 \text{ million ft}^3/\text{min}$) and 0.15 bar (2.2 psi or 60 in of water). Centrifugal and axial fans are used in these applications. Some axial fans have variable-pitch moving blades. These fans maintain higher efficiency over a wide range of loads than constant-speed centrifugal fans. However, they have higher capital cost.

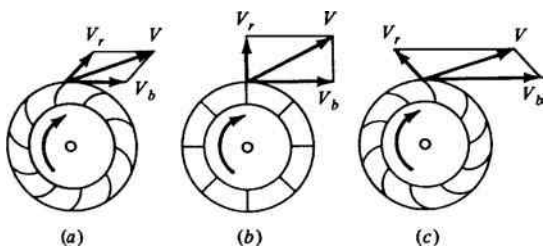


FIGURE 6.18 Centrifugal blading: (a) forward (b) flat, and (c) backward-curved. Vector diagrams show blade-tip velocity V_b , air velocity relative to blade V_r , and absolute velocity of air leaving blade V . V is drawn the same in all cases.

FD fans are normally centrifugal with backward-curved blading (Fig. 6.18). ID fans have flat or forward-curved blading.

The gas flow through the fan can be considered incompressible. This is because the differential pressure (ΔP) across the fan is small. Therefore, $v = \text{constant}$. Thus,

$$|w_{sf}| = \frac{v \Delta P}{\eta_f} \quad \text{ft} \cdot \text{lb}_f / \text{lb}_m \text{ or J/kg}$$

The power would be given by

$$\dot{W}_{sf} = \frac{\dot{m} v \Delta P}{\eta_f} \quad \text{ft} \cdot \text{lb}_f / \text{s or W}$$

where v = specific volume of air or gas, obtained from the perfect gas equations (1–30), $\text{ft}^3 / \text{lb}_m$ or m^3 / kg

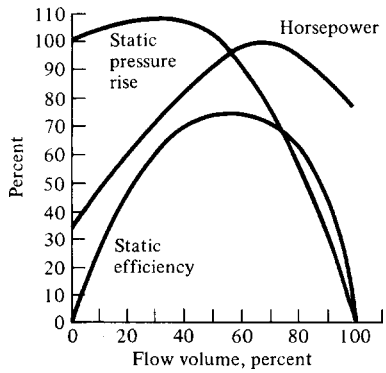
ΔP = pressure rise across fan, $\text{lb}_f / \text{ft}^2$ or N / m^2 (Pa)

η_f = fan efficiency, dimensionless

$\dot{m}$ = mass-flow rate of air or gas, lb_m / s or kg / s

Figure 6.19 illustrates the typical characteristics of a centrifugal fan with backward-curved centrifugal blades. The *static pressure* is defined as the pressure exerted on a wall

FIGURE 6.19 Typical constant-speed characteristics of a centrifugal fan with backward-curved blades.



by an adjacent fluid that is at rest. A moving fluid has also a *velocity pressure*. The *total pressure* is the sum of the static pressure and velocity pressure. The static and velocity pressures are obtained from Bernoulli's equation as follows:

$$Pv + \frac{V_s^2}{2g_c} = \text{constant}$$

where each term has units of energy. Multiplying by the density $\rho = 1/v$ gives

$$P + \rho \frac{V_s^2}{2g_c} = \text{constant}$$

where each term has units of pressure, P is the static pressure, and $\rho V_s^2/2g_c$ is the velocity (or kinetic) pressure. Dividing this equation by the weight density, $\rho g/g_c$ gives

$$h + \frac{V_s^2}{2g} = \text{constant}$$

The h is called *static head*. $V_s^2/2g$ is called the *velocity head*.

6.11.1 Fan Control

The following methods are used to control the fan output:

1. Damper control
2. Variable-speed control

Damper control is less expensive than variable-speed control. It consists of a damper mechanism and an AC motor. Variable-speed control has the following advantages:

1. Lower power consumption than damper control
2. Higher efficiency

Figure 6.20 illustrates the effect of speed on the pressure and power of a centrifugal fan. The main disadvantage of the variable-speed control is that it has higher capital cost than damper control.

The following fans are also used in power plants:

1. Primary-air fan
2. Gas-recirculation fan

Primary-air fans provide air to dry and carry pulverized coal to the furnace or a storage bunker. Gas-recirculation fans are used to control the steam temperature. They recirculate gas from a point between the economizer and air preheater back to the bottom of the furnace.

Fans are a major source of noise. They are normally housed in thick masonry acoustical enclosures. They are also equipped with inlet silencers (FD fans) in some applications.

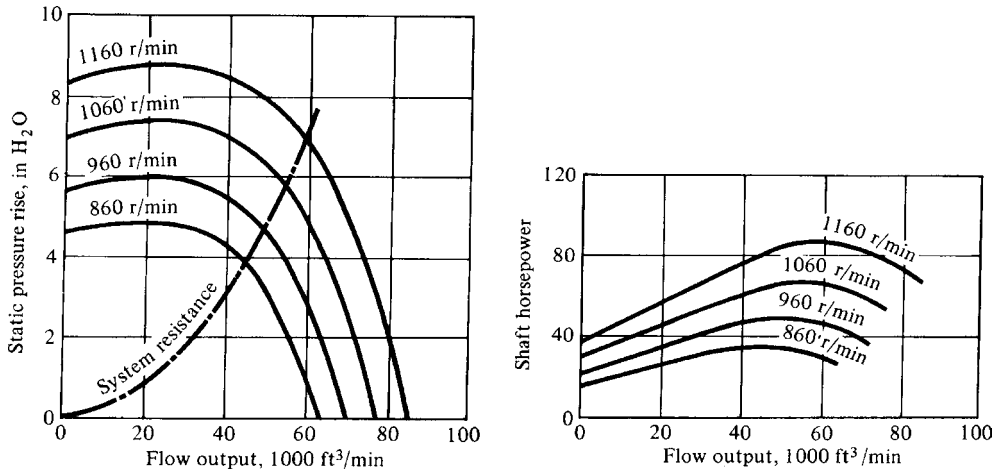


FIGURE 6.20 Typical centrifugal fan performance curves showing effect of varying speed, r/min.

6.11.2 The Stack

Stacks have the following functions:

1. They assist the fans by providing a driving pressure for the gases.
2. They help disperse the gas effluent into the atmosphere.

Driving Pressure

The *driving pressure* Δp_d , in Pa or lb_f/ft^2 , supplied by a stack is given by the following equation:

$$\Delta p_d = (\rho_a - \bar{\rho}_s) H \frac{g}{g_c}$$

where ρ_a = atmospheric air density, lb_m/ft^3 or kg/m^3

$\bar{\rho}_s$ = average stack gas density, lb_m/ft^3 or kg/m^3

H = height of the stack, ft or m

Since both air and gas obey the perfect-gas law, this equation becomes:

$$\Delta p_d = \left(\frac{P_a}{R_a T_a} - \frac{P_s}{R_s \bar{T}_s} \right) H \frac{g}{g_c}$$

where P_a and P_s = absolute pressures of atmospheric air (barometric pressure) and stack gas, respectively, lb_f/ft^2 or N/m^2

R_a and R_s = gas constants for air and stack gas, respectively, $\text{ft} \cdot \text{lb}/(\text{lb}_m \cdot ^\circ\text{R})$ or $\text{J}/(\text{kg} \cdot \text{K})$

T_a and $\bar{T}_s$ = air and average stack gas temperatures respectively, $^\circ\text{R}$ or K

	Altitude, ft									
P_a :	0	1000	2000	3000	4000	5000	6000	7000	8000	10,000
inHg	29.92	28.86	27.82	26.82	25.84	24.90	23.98	23.09	22.22	20.58
psia	14.70	14.17	13.66	13.17	12.69	12.23	11.78	11.34	10.91	10.11
bar	1.013	0.977	0.942	0.908	0.875	0.843	0.812	0.782	0.752	0.697

TABLE 6.1 The Variation of Barometric Pressure with Altitude

The following parameters are approximately the same:

1. P_a and P_s
2. R_a and R_s

Thus,

$$\Delta P_d = \frac{P_a}{R_a} \left(\frac{1}{T_a} - \frac{1}{T_s} \right) H \frac{g}{g_c}$$

P_a varies with weather conditions. It also varies with altitude (Table 6.1). The average temperature of gases in the stack can be approximated by the following:

$$\bar{T}_s = \frac{T_o + T_H}{2}$$

where T_o and T_H are the inlet and exit temperature of the stack, respectively in K or R.

Stacks generate a pressure drop due to the following:

1. Wall friction.
2. Head equivalent to the kinetic energy of the gases leaving the stack.

The latter is normally a few times larger than the former. The total represents only a few percent of the driving pressure.

Dispersion

Dispersion is the second function of the stack. The discharge velocity from the stack generates a *plume* rise C above the stack (Fig. 6.21). The *plume height*, ΔH , is the height of

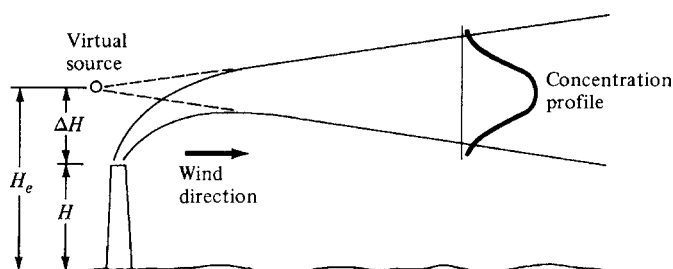


FIGURE 6.21 Dispersion model from a stack of height H and plume ΔH .

a virtual point above the stack. This point is obtained by extending the line of dispersion backward.

Thus, the *effective stack height* H_e is given by:

$$H_e = H + \Delta H$$

6.12 Steam Generator Control

The steam generator control systems include the following:

6.12.1 Feedwater and Drum-Level Control

Feedwater (and therefore steam) flow is controlled for the following reasons:

1. Meet the load demand of the turbine.
2. Maintain the level of water in the steam drum within narrow limits.

Figure 6.22 illustrates a three-element automatic control system. The drum level is one element. The error generated by the drum-level sensor is between the actual drum level and its set point. During periods of high-steam consumption and low-feedwater supply, this parameter acts on the controller to increase the opening of the feedwater valve to meet the steam-flow demand. This action could be slow to correct the error. Thus, it is supplemented by the sensors for feedwater and steam flow. The difference between the signals from these two sensors anticipates the changes in the drum level. It sends a signal to the controller to correct the error.

6.12.2 Steam-Pressure Control

Figure 6.23 illustrates a steam-pressure control system. This system is sometimes called “boiler master.” It maintains the steam pressure by varying fuel and combustion air-flow. These flows will increase when the pressure drops. The steam-pressure sensor controls the fuel- and air-flow controls (pulverized-coal power drive and the forced-draft fan). The trimming signal from the steam-flow sensor and air-flow sensor maintains the proper fuel-air ratio.

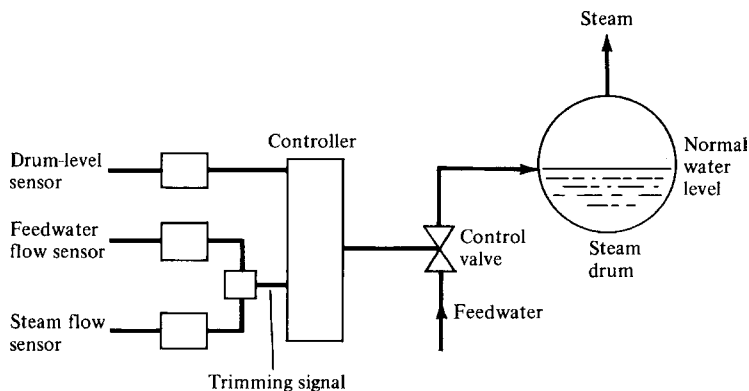


FIGURE 6.22 Schematic of a three-element feedwater control system.

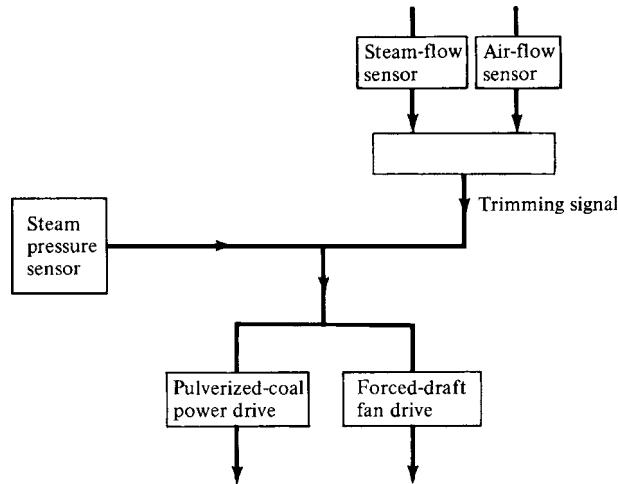


FIGURE 6.23 Schematic of a steam-pressure control system.

6.12.3 Steam-Temperature Control

The temperature of the steam leaving the steam generator should be controlled within a narrow range. The build-up of slag or ash on the heat transfer surfaces will vary the steam temperature. However, the main variations in steam temperature occur due to modifications in load.

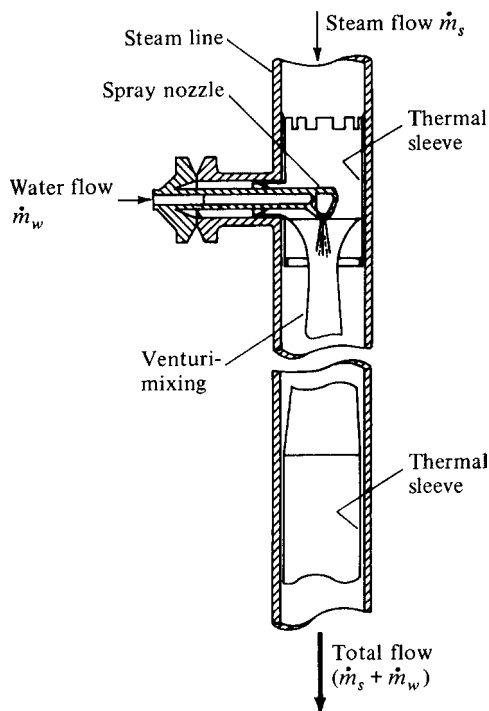
The plant thermal efficiency drops due to a reduction in steam temperature. However, the increase in steam temperature above the design temperature may cause overheating and failure in the following components:

1. Superheater tubes
2. Reheater tubes
3. Turbine blades

The steam temperature is normally adjusted by *attenuation* (reduction in steam temperature). *Surface attenuator* removes the heat from the steam in a heat exchanger. The *shell type* is a form of surface attenuator. A portion of the steam is diverted by a control valve from a point between the primary and secondary superheaters in this method. The steam enters a shell-and-tube heat exchanger containing boiler water. The steam releases some of its heat to the water. It then remixes with the primary steam in the secondary superheater. The steam temperature is controlled by controlling the amount of diverted steam. The *drum type* is another version of surface attenuator. The heat is transferred from the steam to the boiling water in the main steam drum in this design.

The second method of attenuation relies on a device called the *spray, or direct contact, attenuator* (Fig. 6.24). This device sprays low-temperature water from the boiler or economizer exit into a line between the primary and secondary superheaters (Fig. 6.6). This reduces the steam temperature. The water is sprayed into the throat of a mixing venturi. The venturi and a thermal sleeve protect the steam pipe from thermal shock

FIGURE 6.24 A Spray attemperator for steam temperature control.



caused by unvaporized water droplets that reach the pipe. The steam temperature is controlled by varying the amount of water to obtain a flat picture profile beyond point a (Fig. 6.25). The following energy balance is used to determine the flow of water:

$$\dot{m}_s h_{s1} + \dot{m}_w h_w = (\dot{m}_s + \dot{m}_w) h_{s2}$$

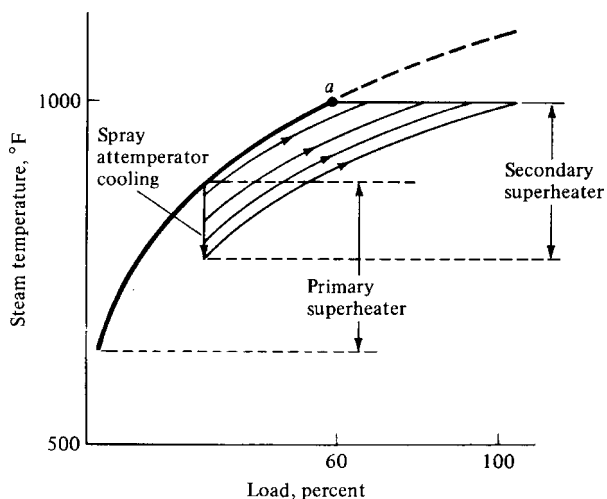
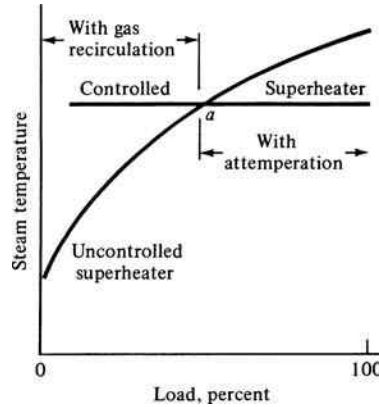


FIGURE 6.25 Attemperation with oversized primary and secondary superheaters.

FIGURE 6.26 Gas recirculation and attemperation in series.



where $\dot{m}_s$ and $\dot{m}_w$ = mass-flow rates of steam and water, respectively, lb_m/h or kg/s
 h_s and h_w = specific enthalpies of steam and water, respectively, Btu/lb_m or J/kg
 subscripts 1 and 2 refer to steam inlet and exit, respectively

Example 6.2 Steam enters a spray attemperator at 2500 psia and 950°F. The spray water comes from the boiler drum, which operates at 2600 psia. Calculate the mass of spray water that must be added per unit mass of steam to reduce its temperature to 900°F.

Solution From the steam tables: $h_{s1} = 1423.1$, $h_{s2} = 1386.7$, $h_w = h_f$ at 2600 psia = 744.47, all in Btu/lb_m. Using equation above,

$$1423.1 + \frac{\dot{m}_w}{\dot{m}_s} \times 744.47 = \left(1 + \frac{\dot{m}_w}{\dot{m}_s}\right) \times 1386.7$$

Therefore,

$$\frac{\dot{m}_w}{\dot{m}_s} = 0.0567$$

The attemperator should be located between the primary and secondary superheaters. Attemperation is used with gas recirculation in some applications (Fig. 6.26).

Separately fired superheater is used in some applications. This superheater has its own burner, fans, combustion chamber, controls, etc. This superheater may serve more than one steam drum. The rate of firing is varied to maintain a flat steam temperature-load curve. This system is normally used in the chemical process industry.

6.13 Bibliography

El-Wakil, M. M., *Power Plant Technology*, McGraw-Hill, New York, 1984.

This page intentionally left blank

CHAPTER 7

Boilers (Steam Generators), Heat Exchangers, and Condensers

7.1 Heat Transfer

The heat-transfer modes are the following:

- *Conduction*: transfers heat by molecular interaction through bodies in contact.
- *Radiation*: transfers heat between bodies by energy propagating electromagnetic waves.
- *Convection*: transfers heat by conduction and radiation and a concurrent mixing of mass flow.
- *Natural circulation (thermosyphoning, natural convection)*: removes heat by natural, or free, convection as temperature change (i.e., density change).
- *Forced convection*: the flow is mechanically induced by a pump or a fan to remove heat.

7.1.1 Steady-State Conduction

The basic conduction equation of Fourier applied to a plane wall (Fig. 7.1a) is

$$q = -kA \frac{dt}{dl}$$

where q = heat transfer, Btu per h

k = thermal conductivity, Btu per sq ft · hr · °F · ft

A = area normal to heat flow, sq ft

dt/dl = temperature gradient, °F per ft (t decreases as l increases, and so minus sign gives plus value for q)

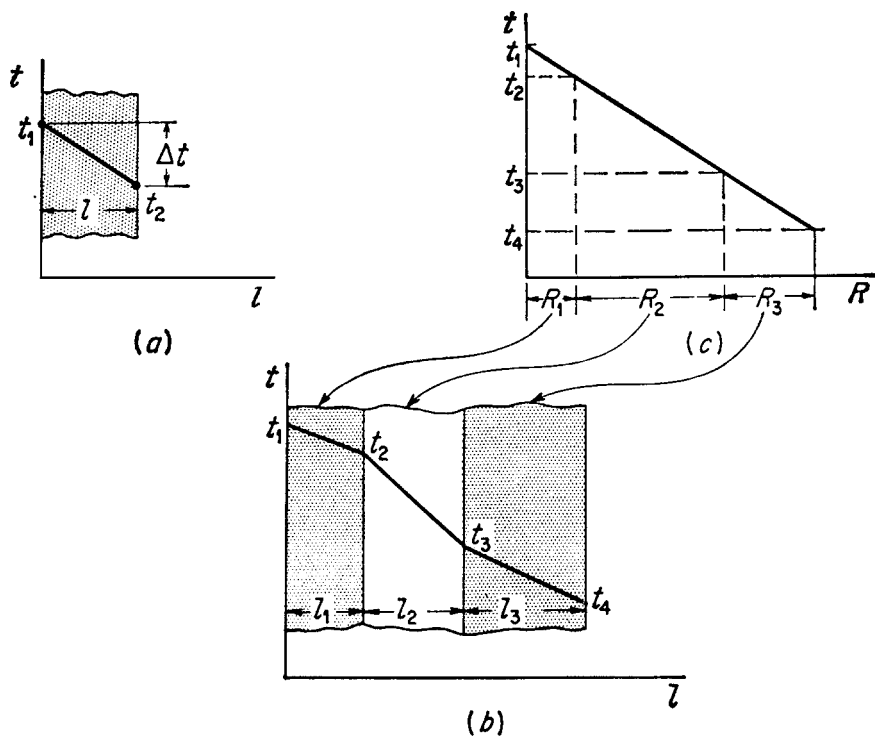


FIGURE 7.1 (a) Temperature gradient with length for single wall; t_1 and t_2 are at wall surfaces, walls are homogeneous, specific heats and conductivities of material constant throughout. (b) Temperature gradient for series conduction; t_2 and t_3 depend on conductivities and wall thickness. (c) Temperature-resistance diagram applied to series conduction; note straight-line characteristic and consequent path temperatures when resistances are known.

When dt/dl is a constant negative value as in Fig. 7.1a

$$q = kA \frac{\Delta t}{l}$$

which can be written as

$$\frac{q}{A} = \frac{\Delta t}{l/k} = \frac{\Delta t}{R} \quad (7.1)$$

where q/A = thermal flow, Btu per h · sq ft

$R = l/k$ = thermal resistance, sq ft · h · °F per Btu

l = wall thickness, ft

For composite walls or thermal resistances in series as in Fig. 7.1b.

$$\begin{aligned} \frac{q}{A} &= \frac{k_1(t_1 - t_2)}{l_1} = \frac{k_2(t_2 - t_3)}{l_2} = \frac{k_3(t_3 - t_4)}{l_3} \\ &= \frac{t_1 - t_4}{l_1/k_1 + l_2/k_2 + l_3/k_3} = \frac{t_1 - t_4}{\Sigma R} = \alpha(t_1 - t_4) \end{aligned} \quad (7.2)$$

where α is the coefficient of heat transfer for the composite wall. Value of α may be computed from individual wall thicknesses and conductivities; specific values will be found in references. Equations (7.1) and (7.2) may be used for curved walls with large radii.

Figure 7.1c shows a resistance diagram for series conduction flow; note the straight-line character.

7.2 Thermal Conductivities

Thermal conductivities are properties of materials and vary widely for different substances—high for metals and low for insulating substances. For any material k depends on the temperature.

7.2.1 Conduction Through Cylindrical Walls

Industrial equipment mostly uses tubes as heat-transfer surfaces. Figure 7.2 shows a cylindrical section for heat transfer.

$$q = -kA \frac{dt}{dr}$$

$$dt = \frac{-qdr}{2\pi Lkr}$$

For constant k by integration

$$q = \frac{2\pi Lk}{\ln(r_2/r_1)} (t_1 - t_2)$$

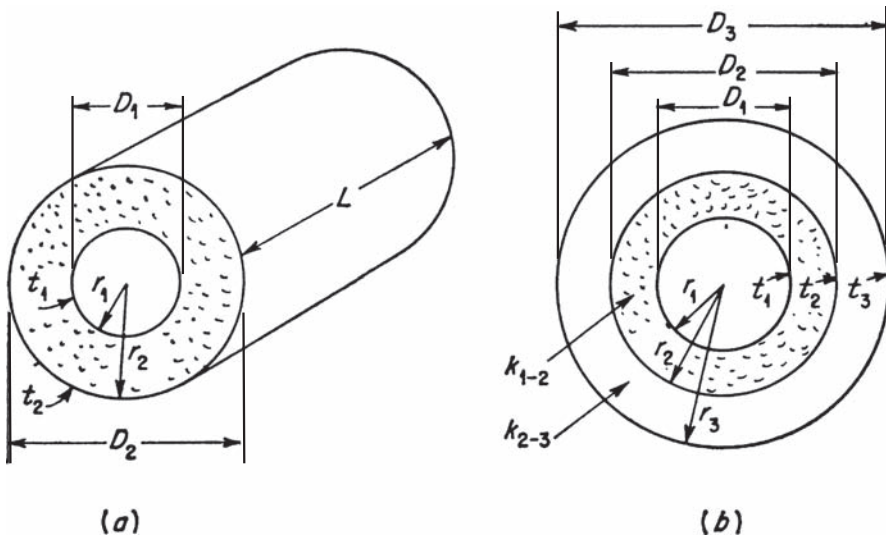


FIGURE 7.2 Cylindrical sections for heat transfer; the heat-flow path is of constant length but increasing area in direction of flow. (a) Single wall, (b) multiple walls.

The thermal current is q/A and may be referred to either the inside or the outside tube area. Based on the inside area,

$$\frac{q}{A_1} = \frac{q}{2\pi r_1 L} = \frac{k}{r_1 \ln(r_2/r_1)} \Delta t$$

For multiple walls (Fig. 7.2b) the unit heat flow is

$$\frac{q}{A_1} = \frac{t_1 - t_3}{\Sigma R} = \frac{t_1 - t_3}{r_1 \left[\frac{\ln(r_2/r_1)}{k_{1-2}} + \frac{\ln(r_3/r_2)}{k_{2-3}} \right]}$$

where k_{1-2} and k_{2-3} are thermal conductivities in Btu per square foot per hour per degree Fahrenheit per foot for the successive wall materials.

7.3 Combination Heat-Transfer Effects

In power plants, heat flows by conduction through refractory furnace walls, boiler and waterwall tubes, feedwater heater tubes, condenser tubes, etc. In all this, equipment conduction is only part of the total effect. Figure 7.3 shows a frequent

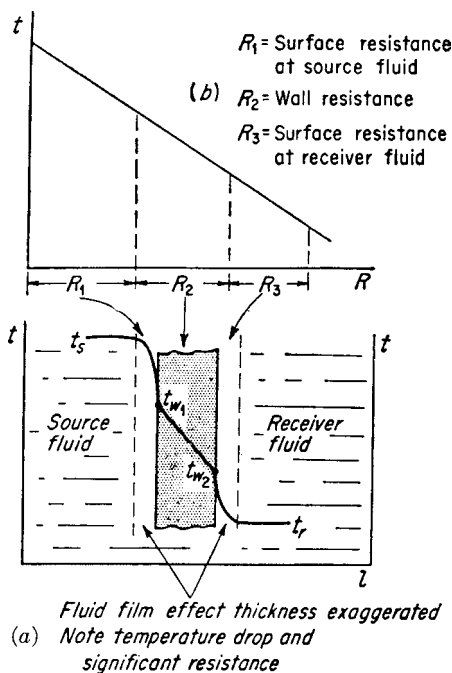


FIGURE 7.3 (a) Surface-film effects as factors in overall heat transfer; note that overall temperature drop is from source fluid t_s to receiver fluid t_r . (b) Temperature-resistance diagram applies also to film effects; note that $t_{w1} - t_{w2}$ for thin metal walls may be insignificant compared with $t_s - t_{w1}$ or $t_{w2} - t_r$.

heat-transfer arrangement: from a source fluid, through a wall, to a receiver fluid. Usually the fluid heat transfer depends more on the fluid films clinging to the wall surfaces than the wall itself. The film conductivities are much lower than the metal-wall conductivity. These films are very thin and stationary, while the fluids flow at considerable velocity,

$$\frac{q}{A} = h_1(t_s - t_{w1}) = \alpha(t_{w1} - t_{w2}) = h_2(t_{w2} - t_r)$$

where h_1, h_2 = film coefficients of heat transfer for convection, Btu per sq ft · h · °F

α = wall coefficient = k/l for a single material plane wall

Also,

$$\frac{q}{A} = \frac{t_s - t_r}{\sum R} = U(t_s - t_r) \quad (8.3)$$

$$\sum R = \frac{1}{h_1} + \frac{1}{\alpha} + \frac{1}{h_2} = \frac{1}{U} \quad (8.4)$$

where U is the overall heat-transfer coefficient in Btu per square foot per hour per degree Fahrenheit. Film effects must be accounted for also on circular or curved walls of tubes. For radial-heat-flow paths as in Fig. 7.2a the overall resistance based on the inner wall is

$$\sum R = \frac{1}{h_1} + \frac{1}{h_2} + \frac{r_1 \ln(r_2/r_1)}{k} = \frac{1}{U}$$

where h_1, h_2 = film coefficients based on inner-wall area

U = overall coefficient of heat transfer based on inner-wall area

In all conduction through circular walls, the inner wall has been arbitrarily taken as the reference area; the outer wall may also be used. In using a consistent area for circular walls, the temperature-resistance diagrams (Fig. 7.3b) can be applied just as with a plane wall.

7.4 Convection Heat-Transfer Coefficients

These coefficients depend on

1. Type and rate of fluid motion
2. Properties of the fluid
3. Size of the fluid-flow path
4. Nature of state change, if any, produced by heat transfer.

7.4.1 Turbulent Forced-Convection Flow Inside Long Circular Tubes

($L/D > 60$). $N_{\text{Re}} > 300$ to 10,000 (depending on fluid).

$$\frac{hD}{k} = 0.023 \left(\frac{DG}{\mu} \right)_{\text{Re}}^{0.8} \left(\frac{c_p \mu}{k} \right)_{\text{Pa}}^n$$

where $n = 0.3$ for cooling and 0.4 for heating.

Properties are evaluated at the mean fluid temperature.
where

$$N_{\text{Re}} = \frac{DG}{\mu}$$

D = diameter (ft)

G = mass flux (lb_m per sq ft $\cdot$ h)

μ = viscosity (lb_m per h $\cdot$ ft)

c_p = specific heat (Btu per lb_m $^\circ\text{F}$)

k = conductivity (Btu per sq ft $\cdot$ h $\cdot$ $^\circ\text{F} \cdot$ ft)

h = heat-transfer coefficient (Btu per sq ft $\cdot$ h $\cdot$ $^\circ\text{F}$)

W = mass-flow rate, lb_m per hour

7.4.2 Streamlined Forced-Convection Flow Inside Tubes (Water and Oils)

This is expressed as follows:

$$\frac{hD}{k} = 2.53 \left(\frac{wc_p}{kD} \right)^{1/3}$$

7.4.3 Turbulent Forced-Convection Flow Across N Onbaffled Tube Banks with Circular Tubes

Tubes in heat exchangers may be arranged staggered or inline (Fig. 7.4.).

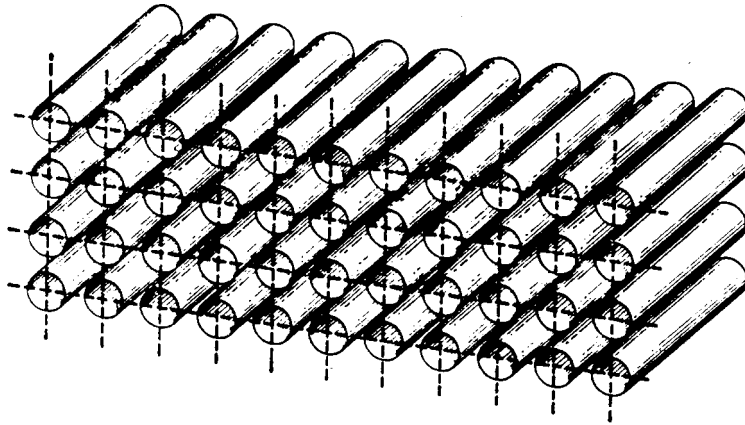
For normal gas flow to a bank of staggered nonbaffled tubes ($N_{\text{Re}} = 2000$ to 40,000; 10 rows minimum)

$$\frac{hD}{k} = 0.33 \left(\frac{c_p \mu}{k} \right)^{0.33} \left(\frac{DG}{\mu} \right)^{0.6}$$

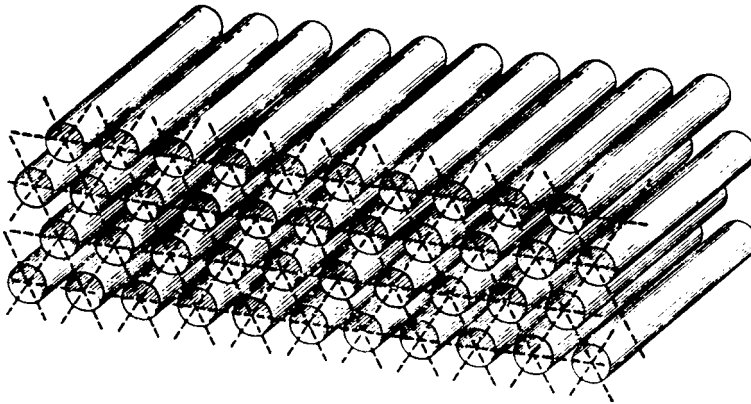
where D = outside tube diameter, ft

G = mass current based on minimum flow area, lb_m per sq ft $\cdot$ h

For less than 10 rows h is multiplied by factors of the order of 0.7 for 1 tube to 1.0 for 10 tubes. For inline tubes h is multiplied by about 0.75.



(a)



(b)

FIGURE 7.4 Tubes may be arranged in either form. Staggered tubes give substantially higher coefficients of heat transfer.

7.5 Boiling Liquids and Condensing Vapors

We can compute heat transfer to liquids from tube surfaces by convection coefficients if no phase changes take place in either liquid or vapor. Boiling needs high heat-transfer rates, and additional factors should be considered (i.e., saturation and boiling curves). Note that the overall heat-transfer coefficient is affected heavily by the scale buildup on the tubes.

7.6 Heat Exchangers

In power-plant heat exchangers heat from one fluid transfers to another. In giving up heat one fluid may cool or condense from vapor to liquid; in absorbing heat the other fluid may warm or evaporate from liquid vapor. Tubular or flat metal walls usually

separate the fluids. Specific examples in power plants include steam generators, condensers, feedwater heaters, oil coolers, evaporators, hydrogen coolers, air heaters, superheaters, and economizers.

Heat-transfer calculations take account of the thermal resistance of fluid and films on each side of the wall and of the wall interior. When both fluids are gases, the wall resistance may be neglected. The overall resistance depends on the flow pattern, fluids, and the presence of scale and fouling. Equations (7.3) and (7.4) are transposed to

$$q = UA(t_s - t_r) \quad (7.5)$$

$$U = \frac{1}{\sum R} = \frac{1}{1/h_1 + 1/\alpha + 1/h_2} \quad (7.6)$$

In this equation A is arbitrary, but all resistance must be based on the same area, either inner or outer; normally we use outside tube-surface areas. In using outer areas for a circular tube

$$\sum R = \frac{1}{h_1} \frac{D_2}{D_1} + \frac{r_2 \ln(r_2/r_1)}{k} + \frac{1}{h_2}$$

In using the inner area of circular tubes

$$R = \frac{1}{h_1} + \frac{r_1 \ln(r_2/r_1)}{k} + \frac{1}{h_2} \frac{D_1}{D_2}$$

where subscripts 1 and 2 refer to inner and outer annulus conditions, respectively. In Eq. (7.5) U is assumed constant over the heat-flow path. The fluid properties are assumed to correspond to the arithmetic average temperature of the range through which the fluid changes. Table 7.1 lists some typical film coefficients used in practice.

Since t_s and t_r may vary in heat exchangers, we have a problem in finding the value of $t_s - t_r$ to be used in Eq. (7.5). Figure 7.5 shows possible temperature variations in heat exchangers; note the significance of *parallel flow* and *counterflow*. The proper value of the mean temperature difference can be shown to be the *logarithmic mean temperature difference* (LMTD) computed as

$$(t_s - t_r)_{\text{mean}} = \text{LMTD} = \frac{\Delta t_{\text{max}} - \Delta t_{\text{min}}}{\ln(\Delta t_{\text{max}}/\Delta t_{\text{min}})}$$

The LMTD applies to the four arrangements shown in Fig. 7.5. For exchangers having two constant temperatures during boiling and condensing, the mean temperature difference is their arithmetic difference. The two basic flow arrangements compare as follows:

1. Parallel flow:
 - a. Final temperatures of both fluids approach the same limit.
 - b. Effective for temperature control of wall material.
 - c. Transfers less heat than counterflow for same area and conditions.

Heat-Transfer Process	h , Btu per sq ft · h · °F
Condensing vapor	
Steam	1000–2000
Ammonia	500–800
Light oils (volatile fuels)	200–300
Heavy oils	20–50
Evaporating liquids	
Water	800–2000
Ammonia	400–500
Light oil	150–300
Heavy oil	10–50
Heating or cooling	
Water	300–1500
Gases	5–250
Oils	10–120

*Courtesy of Foster Wheeler Corporation.

TABLE 7.1 Power-Equipment Film Coefficients*

2. Counterflow:

- Permits greatest possible temperature change in each fluid.
- Transfers more heat than parallel flow for same area and conditions.
- Temperature of cooler fluid more nearly approaches the initial temperature of the hotter fluid.

7.6.1 Shell-and-Tube Heat Exchangers

In this type of exchanger one fluid flows through parallel tubes arranged in a bundle between tube sheets and the other fluid over the tube exteriors, but confined in a shell holding the bundle (Fig. 7.6). These designs use neither pure counterflow nor parallel flow but incorporate a counterflow arrangement in which the shell fluid passes over the tubes at right angles to their axes.

The average temperature difference may be computed by

$$(t_s - t_r)_{\text{mean}} = F \Delta t_{\text{LMTD}}$$

where F is the correction factor for the given shell-and-tube arrangement. Figure 7.7 shows factors for two types of exchangers.

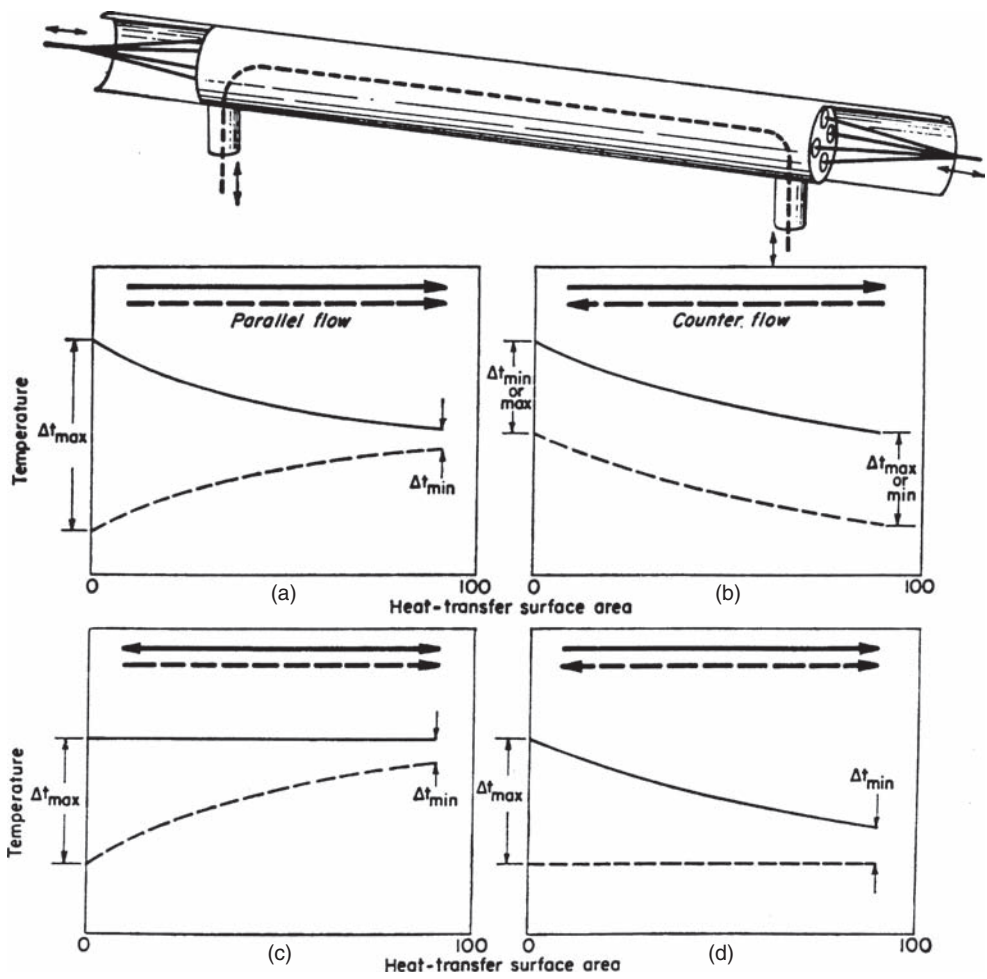


FIGURE 7.5 Possible temperature patterns in heat exchangers. In (a) and (b) both hot- and cold-fluid temperatures change in passing through heater. In (c) hot fluid may be a condensing vapor; in (d) cold fluid may be a boiling saturated liquid. Splitting up fluid streams into several paths (top sketch) increases heat-transfer surface between them for given volume of exchanger.

Figure 7.8 illustrates a shell-and-tube heat exchanger. In order to increase the efficiency, the fluid flowing outside the tubes in the shell is routed by means of baffles (Fig. 7.9). straight tube bundle. The advantage of this type is that it is easy to clean mechanically. The tube expansion is accomodated by having a free end, commonly referred to as the floating end. The U-tube bundle solves tube expansion problems (Fig. 7.10) [most steam generators (boilers) and heat exchangers use U-tube]. Also, the number of tube joints has been reduced by one-half. The “tube sheets” are the end plates into which the tubes are sealed. (The leakage will most likely occur at the

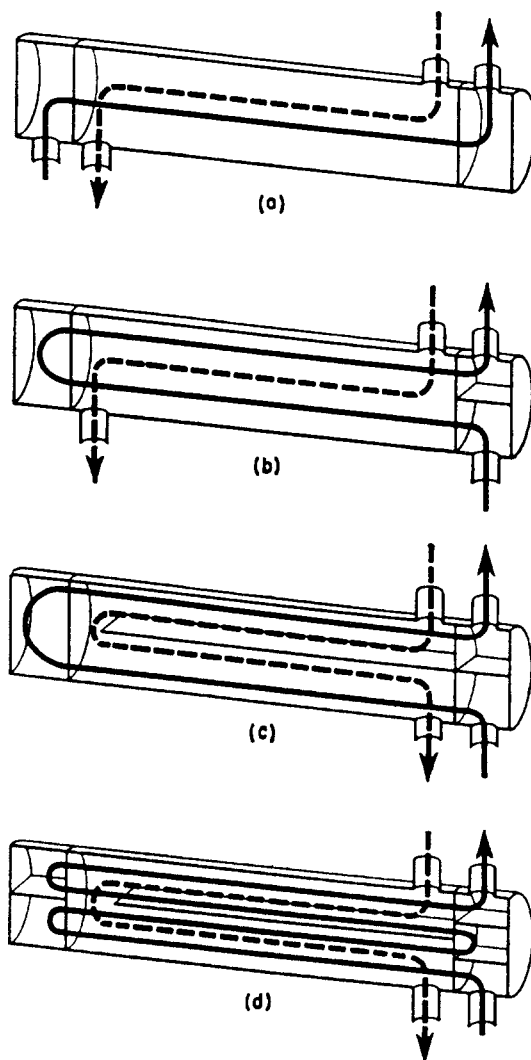


FIGURE 7.6 Paths of fluids through shell-and-tube exchangers can be varied in many ways depending on the job to be done. (a) Single-pass tube and single-pass shell; (b) two-pass tube and single-pass shell; (c) two-pass tube and two-pass shell; (d) four-pass tube and two-pass shell. This last arrangement can be duplicated by putting two heaters like (b) in series for both tube-fluid and shell-fluid flows.

“tube sheets.”) Rolled tube joint is usually used to fasten (Fig. 7.11) the tubes in the tube sheets. Cold rolling flows the tube metal into annular grooves machined in the tube-sheet holes. Welded tube joints are usually tighter than a rolled joint when considerable expansion is expected. Many plants today use rolled joints that are also welded. Figure 7.12 illustrates the parts of a typical heat exchanger.

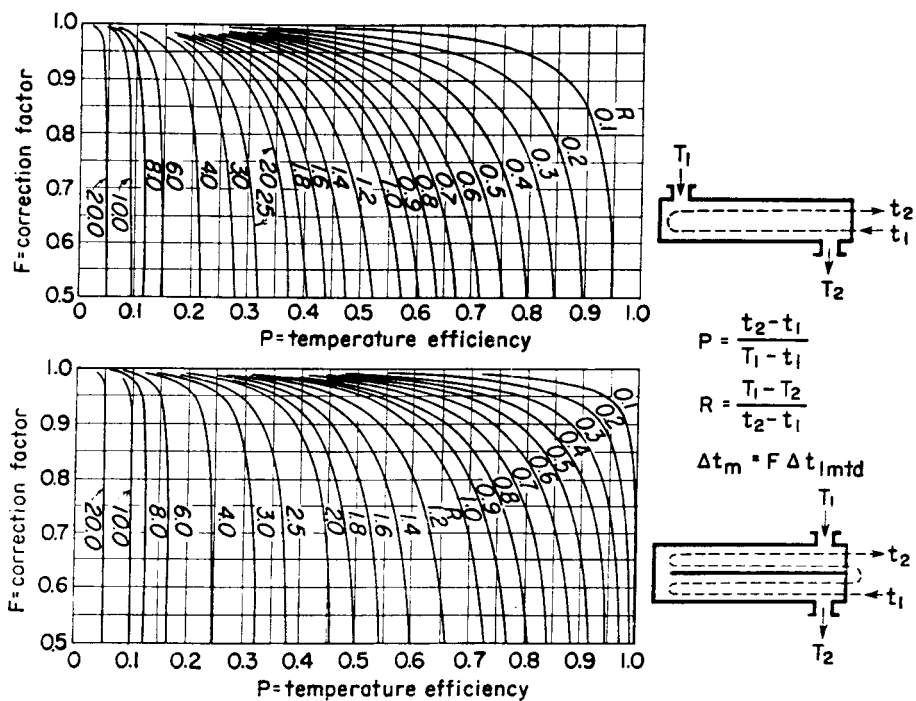


FIGURE 7.7 When heater flow paths differ from true counterflow, a correction factor must be applied to the logarithmic mean temperature difference.

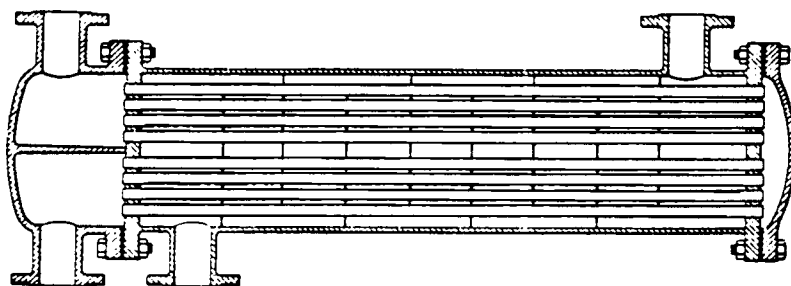


FIGURE 7.8 Shell-and-tube heat exchanger.

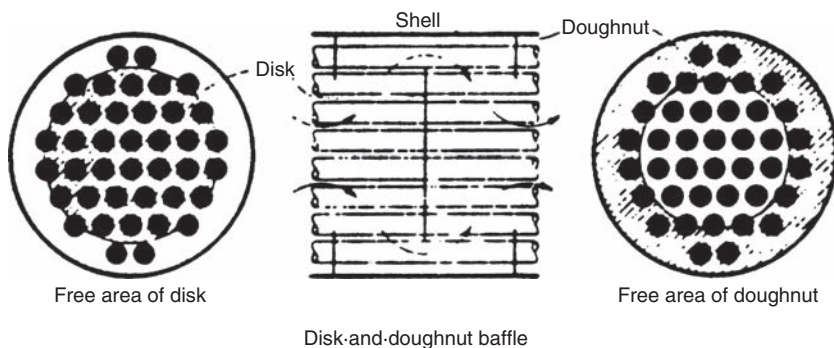
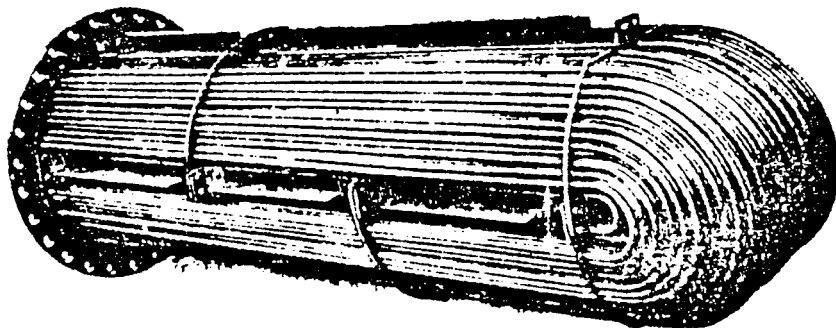


FIGURE 7.9 Heat exchanger baffles.



Tube Nest

FIGURE 7.10 Tube bundle for H.P. feedwater heater.

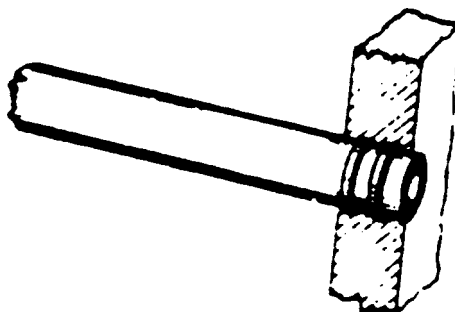


FIGURE 7.11 Rolled tube joint for a heat exchanger.

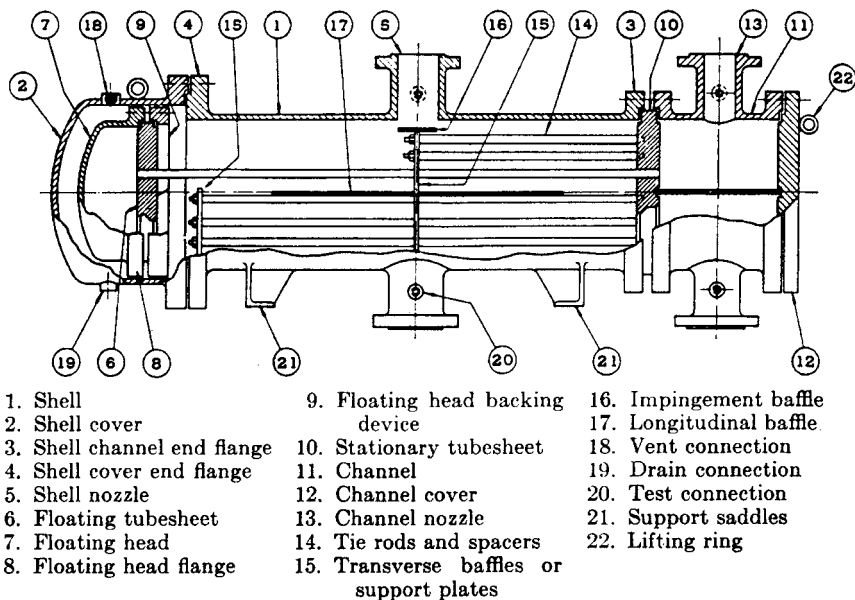


FIGURE 7.12 Nomenclature of typical heat-exchanger parts. (Courtesy of Tubular Exchanger Manufacturers Association.)

Condensers

Condensers are shell-and-tube heat exchangers. Cooling water flows through the tubes and prime-mover (turbine) exhaust steam surrounding the tubes. The steam is condensed at constant temperature. The shell is vacuum-tight. The vacuum is maintained by vacuum pumps that are used to remove air which leaks in and gases such as oxygen that gets entrained with the steam. Note that for good performance, heat exchangers must be drained and vented.

Closed Feedwater Heaters

These shell-and-tube units ordinarily use bleed steam for heating. The steam condenses on tubes carrying the feedwater. Low-pressure heaters may use steam under vacuum and high-pressure heaters at pressures above 600 psig. Steam and water need not be at the same pressure, and one pump may push water through several heaters in series. Water pressure may be as high as 4000 psig. Heater performance may be based on the first-law equation,

$$W_s (h_1 - h_2) = W_f (h_4 - h_3)$$

where W_s = steam flow to heater, lb per h

W_f = feedwater flow through heater, lb per h

h_1 = inlet-steam enthalpy, Btu per lb

h_2 = leaving-condensate enthalpy, Btu per lb

h_3 = inlet-feedwater enthalpy, Btu per lb

h_4 = leaving-feedwater enthalpy, Btu per lb

Figure 7.13 illustrates a modern high-pressure feedwater heater.

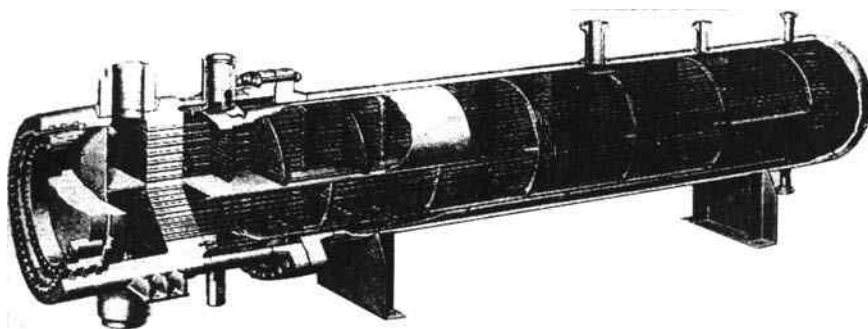


FIGURE 7.13 Modern high-pressure feedwater heater.

Integrated Gasification Combined Cycles

8.1 Introduction

Integrated gasification combined cycles (IGCC) use a process called gasification to convert low-value fuel such as coal, petroleum coke, orimulsion, biomass, or municipal waste to high-hydrogen gas. The gas generated by this process has a low heating value. It is used as the primary fuel for gas turbines. IGCC plants have achieved exceptional levels in environmental performance, availability, and efficiency at a competitive cost of electricity.

IGCC can be considered as having a two-stage combustion of a feedstock. It involves the following steps:

- Partial combustion of a feedstock in a reactor or a gasifier.
- Completion of the combustion in a gas turbine.

The output of an IGCC plant varies from 10 MW to more than 1.5 GW. These plants can be applied in any new or repowering project. They produce electricity at a lower cost than conventional solid fuel plants.

8.2 IGCC Processes

IGCC consists of the following four processes:

1. *Gasification*: Several methods are used to gasify a feedstock. The most common methods involve the following steps:
 - a. Partial oxidation of the feedstock with pure oxygen inside a reactor.
 - b. Conversion of the carbon and hydrogen from the feedstock into a mixture composed mainly of hydrogen and carbon monoxide. This mixture is called synthetic gas, or syngas. Its heat value is 4.65 to 13.02 MJ/scm* (125 to 350 Btu/scf[†]). This value is three to eight times lower than that of natural gas.

*scm: standard cubic meter (i.e., at 1 atmosphere, and 25°C)

[†]scf: standard cubic foot (i.e., at 1 atmosphere, and 25°C)

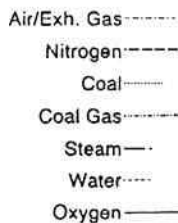


FIGURE 8.1 Integrated gasification combined cycle. (Source: Reprinted with permission from General Electric.)

2. *Syngas Cleanup*: The syngas generated in the reactor must be cleaned before it can be used as a fuel for a gas turbine. The cleanup process involves normally the removal of sulfur compounds, ammonia, metals alkalytes, ash, and particulates. Methanol, ammonia, fertilizers, and other chemicals can be made from the compounds removed from the syngas. These marketable products improve the economics of the IGCC.
3. *Combustion of the Syngas*: The cleaned syngas is combusted in the gas turbine combustors.
4. *Cryogenic Air Separation*: Ambient air enters a cryogenic air separation unit. It is supplemented with air from the discharge of the gas turbine compressor. The air is separated into oxygen and nitrogen. The oxygen enters the gasification reactor. Figure 8.1 illustrates an IGCC plant.

8.3 IGCC Plant Considerations

8.3.1 Turnkey Cost

The typical turnkey cost for an IGCC plant that burns coal is \$1250 USD per kW installed. This cost is divided as follows:

- Gasification – 30%
- Syngas cleanup – 15%
- Power island – 40%
- Cryogenic air separation – 15%

8.3.2 Size of IGCC

The IGCC plant has a similar physical size to a conventional coal-fired power boiler. However, additional space is required for a conventional coal plant for scrubber sludge treatment or ash dewatering.

8.3.3 Output Enhancement

A gas turbine used in an IGCC plant receives more fuel than a gas turbine that burns natural gas. The reason is that syngas has a lower heating value than natural gas. Thus, the mass flow and the resulting power output of an IGCC gas turbine are higher than those of a natural gas turbine.

8.4 Emission Reduction

The SO_x (SO and SO₂), NO_x (NO and NO₂), and particle emissions of an IGCC plant are fractions of those of a conventional coal power plant. Thus, less effort and time are required to obtain local and government environmental permits. Environmental agencies have classified IGCC plants as the best environmental solution to generate power from coal.

8.4.1 Nitrogen Oxides

The NO_x emissions can be reduced by injecting steam, water, carbon dioxide, and/or nitrogen into the combustors of the gas turbine. Since nitrogen is available from the cryogenic air separation unit, it is normally injected into the combustors. Fuel moisturizing is also used frequently to reduce NO_x emissions. All these methods can reduce the NO_x emissions to around 20 parts per million in volume, dry (ppmvd). The dry low NO_x (DLN) technology can reduce the NO_x emissions to around 6 ppmvd.

8.4.2 Air Pollutants

In an IGCC plant, the harmful pollutants are removed from the synthetic gas. Thus, cleanup of the exhaust gas from the gas turbine is not necessary.

8.4.3 Mercury

Most of the mercury is removed in IGCC plants at low cost. Expensive mercury removal systems from the back-end flue gas of an IGCC gas turbine is not required. An activated carbon bed filters the syngas. Around 95% of the mercury is removed by recycled water streams for only \$20 to \$30 USD per kW installed. The lifetime of the carbon bed is around 12 to 18 months.

8.4.4 Carbon Dioxide

Carbon is removed from the syngas in IGCC plants. This is done to create a high-hydrogen fuel. Thus, the carbon dioxide emissions are eliminated.

Carbon is removed from the exhaust gas (after combustion) in conventional boiler plants. This process is about 10 times more expensive due to the larger volume of the post-combustion gas.

8.5 Reliability, Availability, and Maintenance

The reliability, availability, and maintenance (RAM) of a plant have a major effect on the cost of electricity. IGCC plants have a similar RAM performance to natural gas combined-cycle plants.

The increase in hydrogen content of the syngas fuel and the increase in the flow of the syngas have resulted in an increase in the metal temperatures of the hot gas path. Modern IGCC gas turbines operate at a lower firing temperature than natural gas turbines to maintain similar metal temperatures.

8.6 Bibliography

Tomlinson, L. O. and McCullough, S. "Single-Shaft Combined-Cycle Power Generation System," General Electric Power Systems, GER-3767C, New York, N.Y. 1996.

Single-Shaft Combined-Cycle Power Generation Plants

9.1 Introduction

Combined-cycle power plants have outstanding generation economics. The following are their main advantages:

- High thermal efficiency
- Low capital cost
- Low operation and maintenance cost
- Fuel flexibility—ability to use a wide range of gas and liquid fuels
- Operating flexibility—base load, peak shaving, and daily start
- High reliability
- High availability
- Short installation time
- Minimal environmental impact—low stack emissions and heat rejection

The two basic configurations of combined-cycle plants are single-shaft and multi-shaft. The single-shaft configuration consists of a gas turbine, steam turbine, and generator (STAG) installed in a tandem arrangement on a single shaft. This design has demonstrated to be the preferred configuration due to its simplicity of controls and operation and high reliability. Multi-shaft combined-cycle plants have one or more gas turbines and heat recovery steam generators (HRSGs) that supply steam through a common header to the a steam turbine. The single-shaft configuration has provided excellent performance in base load and mid range (20–150 MW) power generation applications.

The multi-shaft combined-cycle configuration is more commonly used in phased installations. The gas turbines are installed and operated in these applications before the installation of the steam power plant. This configuration is also used in applications where it is desirable to operate the gas turbines independently of the steam power plant. The single-shaft combined-cycle plants have proven to be the preferred configuration for applications where the installation and operation of the gas turbine and the steam turbine are concurrent.

The first single-shaft combined-cycle units entered the market in the late 1960s. These simple, efficient, reliable, and compact units demonstrated excellent generation economics. Their efficiency has been maximized by implementing the following features:

- Increasing the gas turbine firing temperature
- Minimizing the usage of compressed air for cooling
- Cooling the turbine using a closed steam circuit

9.2 Performance of Single-Shaft Combined-Cycle Plants

Table 9.1 lists the performance ratings of some of the modern single-shaft STAG plants that burn natural gas. Figure 9.1 illustrates a three-pressure non-reheat steam cycle with a natural circulation heat recovery steam generator (HRSG). The exhaust gas temperature of this unit is 538°C (1000°F). The three-pressure reheat cycle is shown in Fig. 9.2. It is used for gas turbines having an exhaust gas temperature higher than 538°C (1000°F).

Figure 9.3 illustrates the typical ambient temperature effect on the performance of a single-shaft combined-cycle plant. Figure 9.4 illustrates the variation of heat rate with power output of a typical single-shaft combined-cycle plant. The inlet guide vanes of the gas turbines are modulated in the 80 to 100% load range to maintain the heat rate

Unit Designation	Steam Cycle	Net Plant Power	Net Plant Heat Rate (LHV)		Thermal Efficiency
			Btu/kWh	KJ/kWh	
60 Hz					
S106B	Non-reheat, 3-pressure	59.8	7005	7390	48.7
S106FA	Reheat, 3-pressure	107.1	6440	6795	53.0
S107EA	Non-reheat, 3-pressure	130.2	6800	7175	50.2
S107FA	Reheat, 3-pressure	258.8	6090	6425	56.1
S107G	Reheat, 3-pressure	350.0	5885	6210	58.0
S107H	Reheat, 3-pressure	400.0	5690	6000	60.0
50 Hz					
S106B	Non-reheat, 3-pressure	59.8	7005	7390	48.7
S106FA	Reheat, 3-pressure	107.4	6420	6775	53.2
S109E	Non-reheat, 3-pressure	189.2	6570	6935	52.0
S109EC	Reheat, 3-pressure	259.3	6315	6660	54.0
S109FA	Reheat, 3-pressure	376.2	6060	6395	56.3
S109H	Reheat, 3-pressure	480.0	5690	6000	60.0

Notes: 1. Site Conditions-59°F, 14.7 psia, 60% RH (15°C, 1013 bar, 60%).

TABLE 9.1 Single-Shaft Stag Product Line Ratings

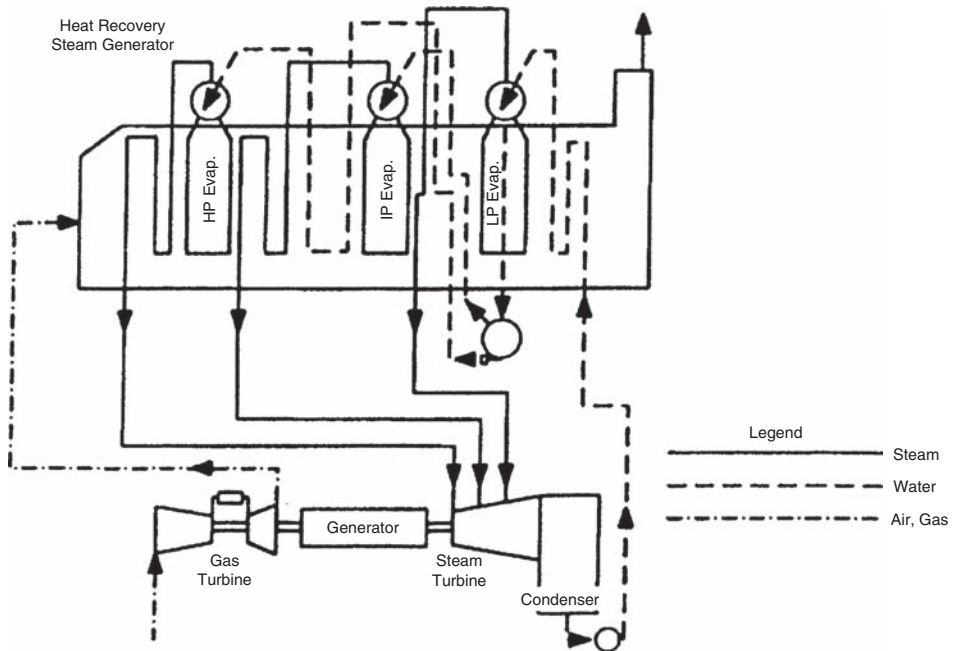


FIGURE 9.1 Single-shaft, combined-cycle, three-pressure non-reheat steam cycle. (Source: Reprinted with permission from General Electric.)

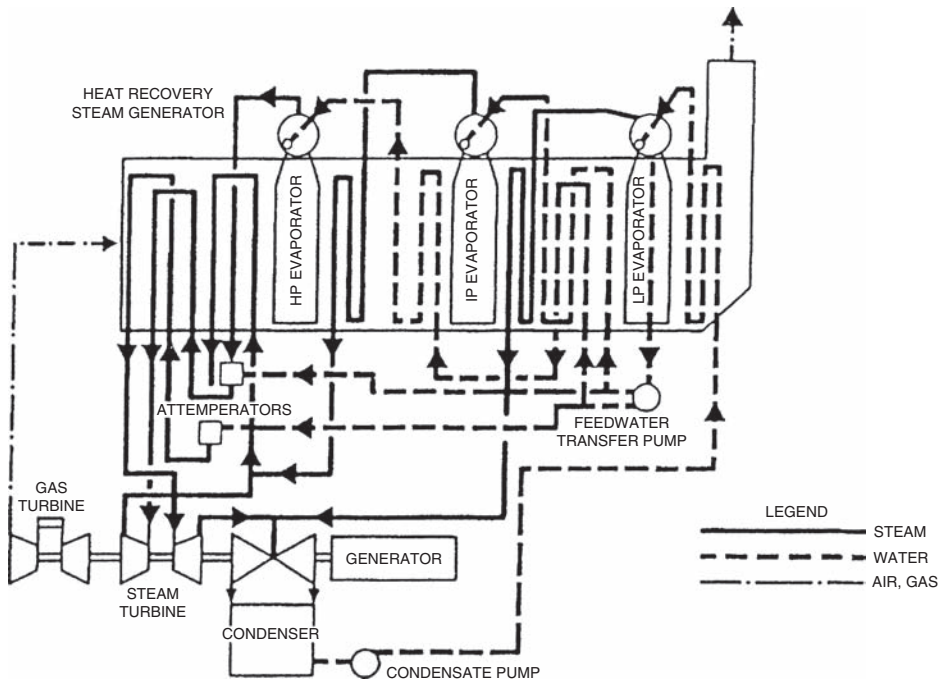


FIGURE 9.2 Single-shaft, combined-cycle, three-pressure reheat steam cycle. (Source: Reprinted with permission from General Electric.)

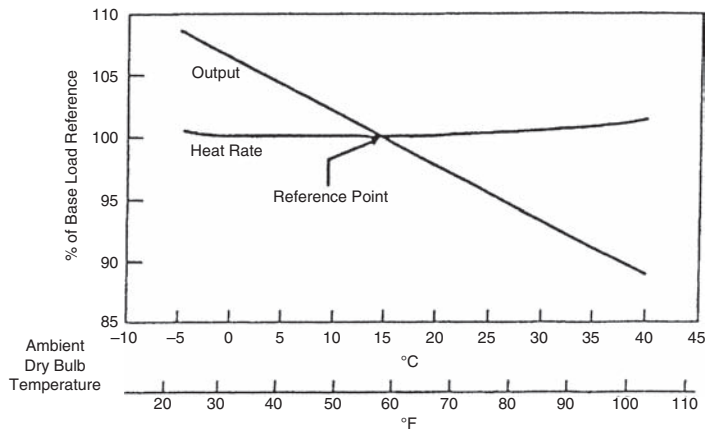


FIGURE 9.3 Ambient temperature effect on performance. (Source: Reprinted with permission from General Electric.)

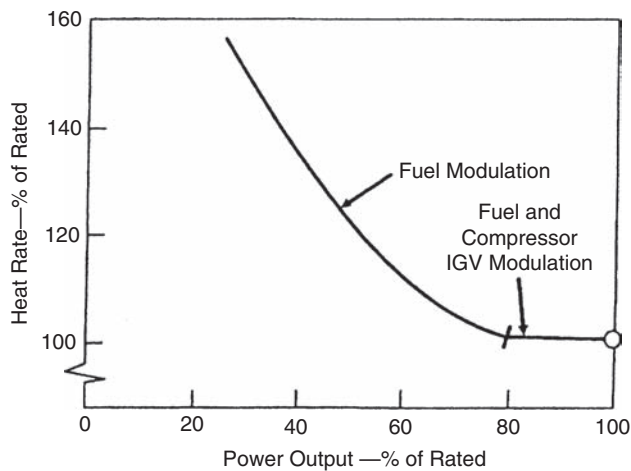


FIGURE 9.4 Typical single-shaft combined-cycle unit heat rate variation with power output. (Source: Reprinted with permission from General Electric.)

near the rated value (i.e., the inlet guide vanes close partially in the 80 to 100% load range to maintain high gas turbine exhaust temperature). The fuel flow is reduced when the load drops below 80% of the rated load, resulting in an increase in heat rate.

9.3 Environmental Impact

Combined-cycles plants have minimal environmental impact. Their exhaust emissions are low due to the following reasons:

1. High-quality combustion in the gas turbines
2. High plant efficiency (i.e., the heat rejected to the environment is low)

The single-shaft integrated gasification combined-cycle (IGCC) plants have achieved excellent environmental performance. Their NO_x emissions are lower than 9 parts per million in volume, dry (ppmvd) at 15% oxygen (16 g/GJ).

9.4 Equipment Configurations

Figure 9.5 illustrates a typical single-shaft, combined-cycle unit with non-reheat steam turbines. Its rating is less than 40 MW for 60 Hz units and 60 MW for 50 Hz units. The generator is located between the gas turbine and the steam turbine. The shaft position of the gas turbine and the steam turbine is controlled individually by a separate thrust bearing. A flexible coupling is located between the generator and the steam turbine. It accommodates the axial movement of the shaft between the generator and the steam turbine.

The cross section of the steam turbine used in a typical STAG unit is a simple, single casing unit with full arc steam admission. Figure 9.6 illustrates a typical configuration

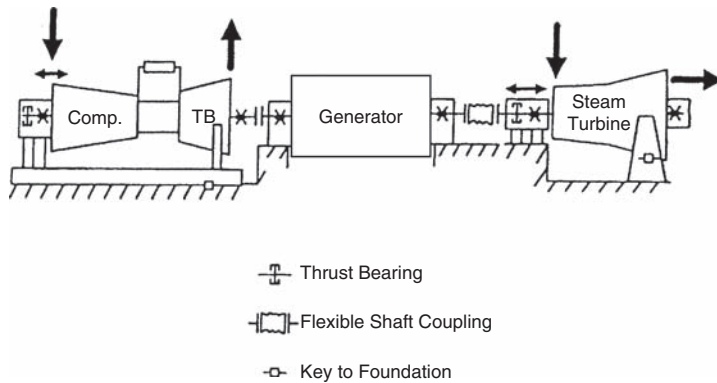


FIGURE 9.5 STAG General Electric 107EA, 109E configuration. (Source: Reprinted with permission from General Electric.)

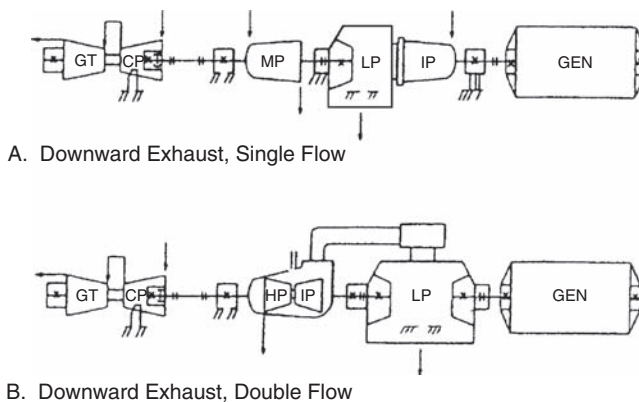


FIGURE 9.6 Large single-shaft combined-cycle equipment configuration. (Source: Reprinted with permission from General Electric.)

of a single-shaft combined-cycle unit with larger, reheat steam turbines. The generator is placed at the end of the shaft. A single thrust-bearing controls the shaft position. It is located near the gas turbine. Solid couplings are used along the equipment train.

9.5 Starting Systems

The starting systems of a single-shaft combined-cycle unit include the following:

- 1. Steam supplied from an auxiliary boiler to the steam turbine.
- 2. Generator equipped with a variable frequency load commutated inverter (LCI) system.
- 3. Electric motor and torque converter. This option is normally used on small units.

9.6 Auxiliary Steam Supply

A small auxiliary steam supply is required during the starting phase for the steam turbine seals. The auxiliary steam conditions are around 316°C (600°F) and 690 kPa (100 psi). In multiple units applications, the steam is supplied through an auxiliary steam header from an operating unit.

9.7 Plant Arrangement

Figures 9.7 through 9.9 illustrate a typical plant arrangement for the GE STAG 107FA. The condenser, auxiliaries, and piping are housed in the basement of the turbine building.

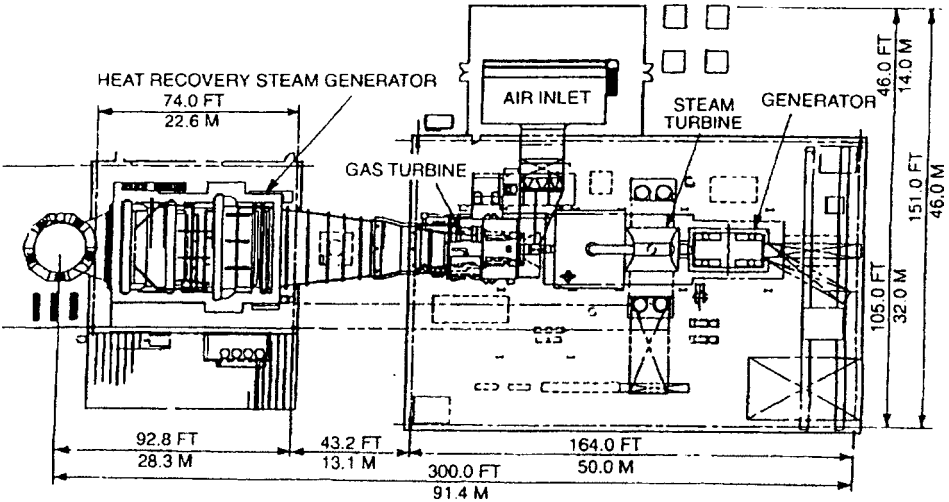


FIGURE 9.7 STAG 107FA equipment arrangement plan. (Source: Reprinted with permission from General Electric.)

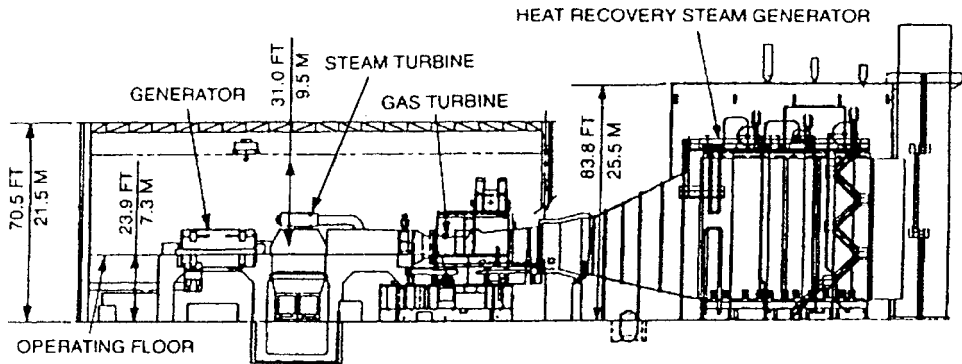


FIGURE 9.8 STAG 107FA equipment arrangement elevation. (Source: Reprinted with permission from General Electric.)

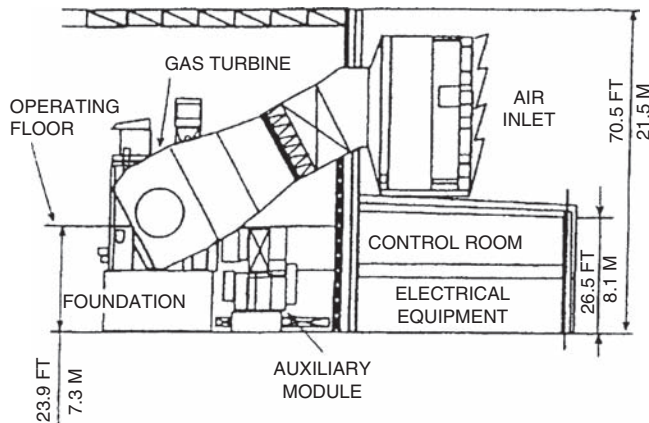


FIGURE 9.9 STAG 107FA elevation section gas turbine. (Source: Reprinted with permission from General Electric.)

9.8 Maintenance

The inspection interval of the gas turbine combustion is 8000 hours. It coincides with the annual HRSG inspection for base-loaded units. The inspection interval of the HRSG and the gas turbine combustion is performed every 2 years on units that operate 4000 hours per year (mid-range peaking service). The inspection interval of the hot gas path of the gas turbine is 24,000 hours. The major inspection interval of the gas turbine is 48,000 hours. It coincides with the major inspection of the steam turbine and the generator rotor.

9.9 Advantages of Single-Shaft Combined-Cycle Plants

The following are the advantages of single-shaft combined-cycle plants:

- Simplicity of operation.
- High reliability and availability.

- Independence of unit operation and maintenance.
- Low maintenance cost.
- Simplicity of control and operation with multiple pressure and reheat steam cycle. The installed cost and heat rate of a single-shaft combined-cycle plant is comparable to a multi-shaft combined-cycle plant.

9.10 Bibliography

Tomlinson, L. O., and McCullough, S., "Single-Shaft Combined-Cycle Power Generation System," General Electric Power Systems, GER-3767C, New York, NY. 1996.

CHAPTER 10

Selection of the Best Power Enhancement Option for Combined-Cycle Plants

10.1 Plant Description

Figure 10.1 illustrates a 230 MW combined-cycle plant having a 155 MW gas turbine (GT) burning natural gas in a dry low NO_x combustor. The unit ratings are based on the International Organization for Standardization (ISO) conditions. The heat recovery steam generator (HRSG) has a triple pressure design and an integral deaerator. The steam power plant has a reheater and a condensing low-pressure (L-P) steam turbine (ST) coupled to a cooling tower. The steam conditions at the inlet to the high-pressure (H-P) turbine are 10 MPa/538°C (1450 psi/1000°F). The exhaust pressure of the steam turbine is 9.5 cm Hg absolute (3.75 in Hg). Select the most suitable power enhancement option for this plant assuming that the summer peak conditions are 35°C (95°F) dry bulb (DB) temperature and 60% relative humidity (RH). The cost of water-treatment, consumption, and disposal in this zero-leakage facility are assumed to be \$3/3.8 m³ (\$3/1000 gal) of raw water \$6/3.8 m³ (\$6/1000 gal) of treated demineralized water, and \$5/3.8 m³ (\$5/1000 gal) of water disposal. The GT in this plant has the capability of burning a variety of liquid distillate fuels in addition to natural gas.

Table 10.1 summarizes the performance of the various power enhancement options of combined-cycle plants. The gas turbine (GT) combined-cycle (CC) plants have numerous advantages including low capital, and operating and maintenance cost. However, the power output from these plants decreases significantly due to increase in ambient temperature. This is caused by the reduction of GT mass flow as a result of the lower density of warm air. Since this decrease in power output occurs during peak power loads, power enhancement options are used to compensate for the decrease in power output.

10.2 Evaluation of Inlet-Air Pre-Cooling Option

Evaporative cooling enhances the GT power by increasing the air density. The air temperature drops to near the ambient wet-bulb temperature due to spraying water or fog into the air stream entering the gas turbine. The inlet air conditions will change from 35°C (95°F) DB and 60% RH to 29°C (85°F) and 92% RH when the effectiveness of the

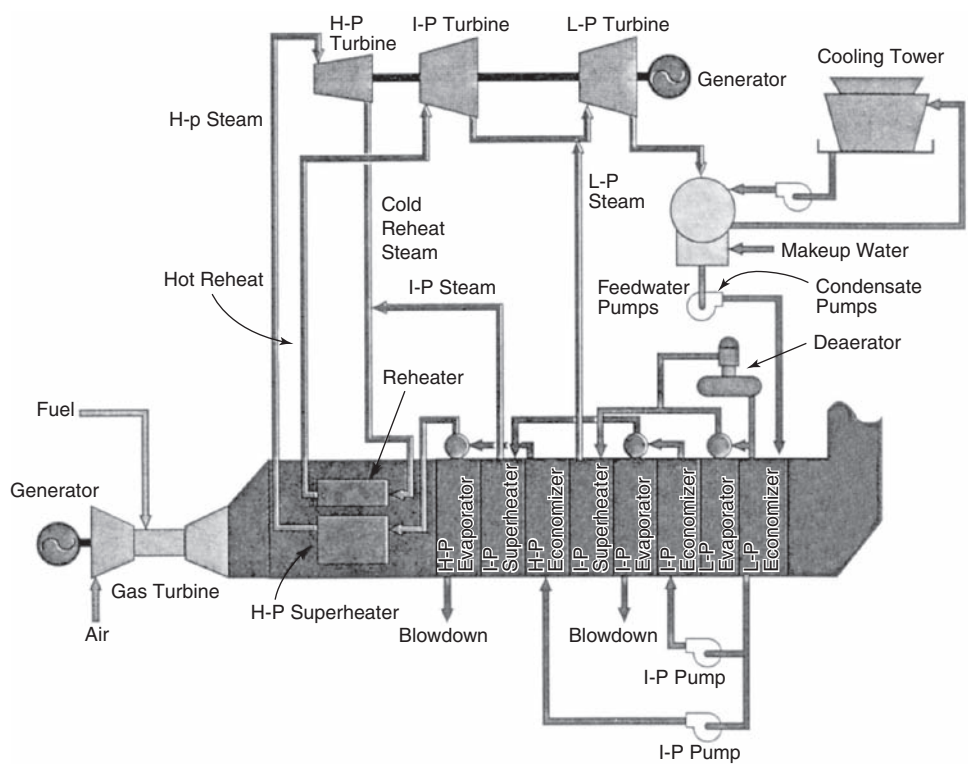


FIGURE 10.1 155-MW natural-gas-fired gas turbine featuring a dry low NO_x combustor. (Source: Reprinted with permission from *Power* magazine.)

evaporative cooler is 85%. This result is obtained using the conventional humidity chart calculations explained in Ref. 1. The power output from the GT and ST increase by 5.8 MW and 0.9 MW, respectively as a result of using this option. The increase in the CC heat rate is 15 Btu/kWh (14.2 kJ/kWh) or 0.2%. It should be noted that the CC efficiency will decrease by 0.2% due to 0.2% increase in heat rate because the efficiency = 1/(heat rate).

The incremental cost of the evaporative cooling process is:

$$\$180/\text{kW} \times 6650 \text{ kW} = \$1,200,000.$$

The effectiveness of this option increases when the air is dryer, for example, 95°F DB (35°C) and 40% RH. The inlet-air temperature in this case will decrease to 75°F DB (23.9°C) by increasing RH to 88%. The increase in the CC output and heat rate here will be 7% and 1.9%, respectively. The incremental installed cost in this case will be \$85/kW. Clearly, the effectiveness of evaporative cooling increases when the air is dryer. Make-up water is required for the evaporation process and blowdown requirements. The increase in the water supply costs relative to the base case is \$15/h of operation for the first scenario and \$20/h for the second. The cost of disposing the evaporative cooler blowdown is \$1/h for the first scenario and \$2/h for the second.

Measured Change from Base Case	Case 1 Evap. Cooler	Case 2 Mech. Chiller	Case 3 Absorp. Chiller	Case 4 Steam Injection	Case 5 Water Injection	Case 6 ¹ Supp.-Fired HRSG	Case 7 ² Supp.-Fired HRSG
GT output, MW	5.8	20.2	20.2	21.8	15.5	0	0
ST output, MW	0.9	2.4	-2.1	-13	3.7	8	35
Plant aux. load, MW	0.05	4.5	0.7	400	0.2	0.4	1
Net plant output, MW	6.65	18.1	17.4	8.4	19	7.6	34
Net heat rate, Btu/kWh ³	15	55	70	270	435	90	320
Incremental costs							
Change in total water cost, \$/h	15	35	35	115	85	35	155
Change in wastewater cost, \$/h	1	17	17	2	1	1	30
Change in capital cost/net output, \$/kW	180	165	230	75	15	70	450

¹Partial supplementary firing.

²Full supplementary firing.

³Based on lower heating value of fuel.

TABLE 10.1 Performance Summary for Enhanced-Output Options

10.3 Evaluation of Inlet-Air Chilling Option

The inlet air to the GT cannot be cooled below its wet-bulb (WB) temperature using the evaporative cooling option. Chillers are used in some cases to cool the inlet air below its wet-bulb temperature. Figure 10.2 illustrates inlet air chilling systems using centrifugal (mechanical) and absorption-type chillers. Both systems consist of the following:

- A power source to drive the chiller.
- A cooling medium (water or a refrigerant).
- A heat exchanger to cool the inlet air.
- A cooling tower to reject the heat.

Table 10.1, case 2 shows the effects of the mechanical chiller on the plant. The chiller reduces the inlet air temperature from 35°C (95°F) to 15.6°C (60°F) DB. Ice will start to form at the compressor inlet if the temperature is reduced further. The increase in the power output of the plant is given by:

Increase in the power output of the plant = 20.2 MW (for GT) + 2.4 MW (for ST) – 4.5 MW
(increase in the power required for the plant auxiliary load) = 18.1 MW

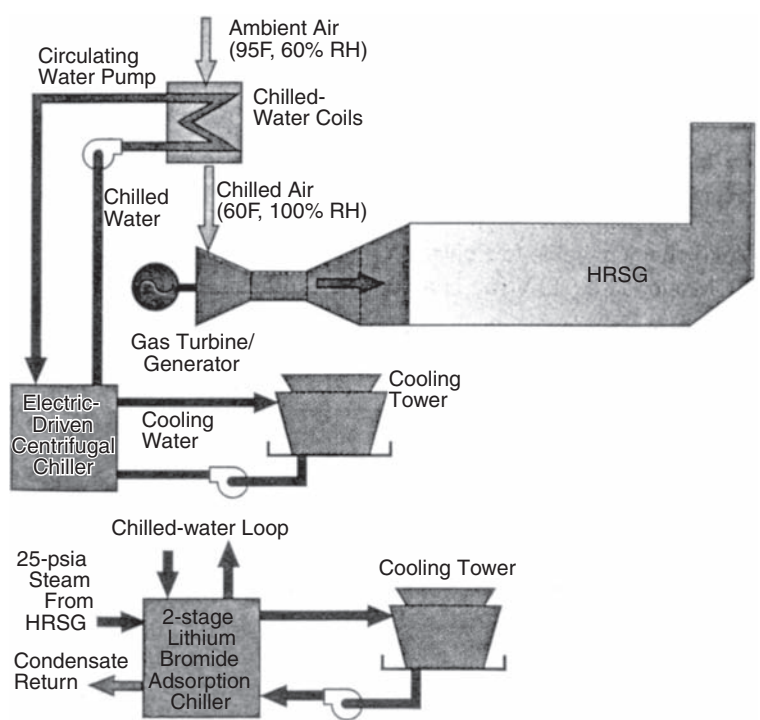


FIGURE 10.2 Inlet-air chilling using either centrifugal or absorption-type chillers boosts the achievable mass flow and power output during warm weather. (Source: Reprinted with permission from *Power* magazine.)

This represents an 8.9% increase in the power output. However, there is a decrease in the plant efficiency by 0.8% (increase in the plant heat rate by 0.8%) and an air pressure drop of 3.8 cm (1.5 in) H_2O across the heat exchanger located at the inlet to the GT. The plant's water requirements increased by 47.3 m³/min (12,500 gal/min) due to the chilling system. The installed cost of the chilling system is: \$3,000,000/18,100 kW = \$165.75/kW. The plant's operating and maintenance cost increased by \$35/h, due to raw-water consumption. There is also an increase of \$17/h due to disposing of additional blowdown from the cooling tower.

10.4 Evaluation of Absorption Chilling System

Steam or hot water are normally used in absorption chilling systems. The saturated steam or water requirement for the absorption chiller to cool the inlet air to the same conditions as the mechanical chiller [15.6°C (60°F), 100% RH] are 50,576 kg/h at 70.9 kPa (111,400 lb/h at 10.3 psi), or 25.9 m³/min at 188°C (6830 gal/min at 370°F) of hot water, respectively. The steam is normally extracted at a pressure of 139.9 kPa (20.3 psi) from the low-pressure (L-P) turbine. It is attemperated until it becomes saturated. The power output from the plant will increase in this case by 17.4 MW (8.7%) due to the absorption chilling system. However, the steam turbine output and the CC plant efficiency will decrease by 2.1 MW and 1% (heat rate increase by 1%), respectively. Table 10.1 shows that the overall cost of the absorption chilling system is around \$4 million (\$230/KW). The increase in the operating and maintenance (O&M) cost due to raw-water consumption is \$35/h. The cost of disposing the blowdown of the additional cooling water is \$17/h.

Absorption chillers cannot handle load variations as effectively as mechanical chillers. Thus, they cannot be used for cycling applications. Their efficiency decrease significantly when they operate at part-load. They also require additional monitoring and maintenance than mechanical chillers. The inlet air temperature can also be reduced by thermal storage. However, this option consumes significant amount of power. It consists of the formation of ice or cool water during off peak periods. The ice or cool water are used during peak periods to reduce the temperature of the incoming air to the GT. The advantage of this option is the significant increase in return due to the high cost of peak power.

10.5 Evaluation of the Steam and Water Injection Options

The power output of GTs will increase significantly when water or steam is injected into the combustors (Fig. 10.3). However, this will decrease the efficiency of the CC plant. The steam is normally extracted at 2825 to 2997 kPa (410–435 psi) from the high-pressure (H-P) section of the heat recovery steam generation (HRSG). The power augmentation from steam injection is obtained from Table 10.1, case 4:

$$\begin{aligned}\text{Power augmentation from CC} &= \text{power increase from GT} - \text{power decrease from ST} \\ &= 21.8 - 13 = 8.8 \text{ MW}\end{aligned}$$

However, the efficiency of the CC will decrease by 4%. The steam injected must be demineralized. The cost of water treatment in this case is \$130/h of operation. The cost of water treatment in the water injection option is much lower than the steam injection option. This is because only filtered water that meets the pH, turbidity, and hardness

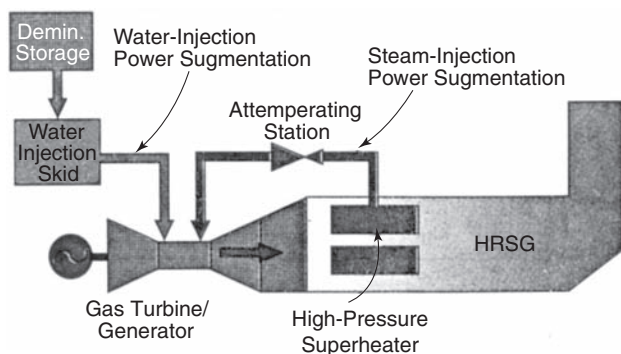


FIGURE 10.3 Water or steam injection can be used for both power augmentation and NO_x control. (Source: Reprinted with permission from *Power* magazine.)

specifications is required. Table 10.1 indicates the CC output will increase by 19.2 MW (9.4%), However, the plant efficiency will decrease by 6.4%.

10.6 Evaluation of Supplementary Firing in HRSG Option

The air leaving the GT still contains enough oxygen to support the combustion of liquid or gaseous fuels in a supplementary firing unit located upstream of the HRSG. Table 10.1 indicates that the increase in power from the ST is 8 and 35 MW for partial and full supplementary firing, respectively. The main disadvantage of supplementary firing is the decrease in CC efficiency and increase in emissions of NO_x and CO. The capital and operating and maintenance (O&M) cost of the plant will also increase.

When 29.3 MW (100-million Btu/h) of supplementary firing heat is added to this plant, the power output will increase by 5.5%. However the efficiency will decrease by 2%. The capital cost of this option is significant because all the components and equipment included in the HRSG and ST will be affected.

10.7 Comparison of All the Power Enhancement Options

The performance summary of the various power enhancement options is presented in Table 10.1. It indicates that mechanical chilling provides the largest increase in power output (18.1 MW) from the plant for the least decrease in plant efficiency [net heat rate increase of 52.3 kJ/kWh (55 Btu/kWh)]. The capital cost of this option is \$3 million. It is the highest among all the remaining options. The water injection option has the advantages of increased power output and NO_x control. Its installed cost is relatively low. However, the installed cost of steam injection is significantly higher than water injection due to water-treatment requirements. Table 10.1 also indicates that the supplementary firing option is more suitable for applications requiring increased power output for extended periods of time. This is due to the high capital cost per net output (\$450/kW).

10.8 Bibliography

Hicks, T., *Handbook of Mechanical Engineering Calculations*, McGraw-Hill, New York, 1997.
 Kiameh, P., *Power Generation Handbook*, McGraw-Hill, New York, 2002.

CHAPTER 11

Economics of Combined-Cycle and Cogeneration Plants

11.1 Introduction

In an attempt to encourage efficient power generation, Class 34 Capital Cost Allowance (CCA) was legislated. It provided special tax incentives for power plants having an efficiency higher than 50% (e.g., a 3-year write-down of the capital cost of equipment and accessories).

Recent advancements in metallurgy and aerodynamics have lead to the development of highly efficient cogeneration and combined-cycle plants. Relatively large plants (100–400 MW) are built today for \$625/kW installed. They require 20 to 25 operators for around-the-clock attendance.

Current studies indicate that the demand for electric power will exceed the projected generating capacity. Thus, future industrial expansion will be limited unless more power-generating plants are installed. Cogeneration and combined-cycle plants have proven to be the most economical source of electric power generation after hydraulic station.

Recent deregulations in the electric power industry have created opportunity for cogeneration and combined-cycle plants. Industrial and commercial users will have the option to purchase power from the most competitive supplier. However, investors are concerned about the possibility of significant increase in the price of natural gas and accuracy of the economic growth projections.

11.2 Natural Gas Prices

In 1992, Canadian and wellhead prices averaged about \$1.35/mcf. This price is 50% below the peak reached in 1984. The factors that caused this significant decrease are deregulation and surplus production.

The United States and Canada have large established gas reserves and potential gas reserves. Mexico also has large reserves, and, due to NAFTA, it could become a player in limiting the long-term increase in the price of natural gas. The established reserves in North America are estimated to be equivalent to 13 years of production at current rates. This amount would be considerably higher if the potential reserves are considered.

	1995*	2000	2010
Coles	2.05	2.35	2.35
TCPL	1.70	1.95	2.60
NEB			
• Current tech.	1.68	2.10	3.90
• High tech.	1.40	1.35	2.15
Power West	2.00	2.47	n.a.
Dobson Resource			
• 17 Consultants	2.00	2.45	3.05
• 8 Banks	1.85	1.95	2.15
NRCan	1.75	2.00	2.45

*Actual.

Sources:

Coles Gilbert Associates Ltd., January 1994.
TCPL, 1994/95 Facilities Application to NEB, April 1994.
NEB, Canadian Energy: Supply & Demand 1993–2010, July 1994.
Power West Financial Ltd., Energy Update Report, July / August 1994.
Dobson Resource Management Ltd., July 1994.

TABLE 11.1 Natural Gas Price Projections

Table 11.1 lists the projected natural gas prices by different sources until the year 2010. These numbers do not include the cost of transportation from local gas utilities. All sources agree that the price of natural gas is not expected to increase significantly before 2010.

11.3 Economic Growth

The economic growth in Canada is driven mainly by demographics and economic growth in the U.S. The Canadian population is expected to grow at a rate of 1.0% annually until 2020 mainly due to immigration. The growth in the Canadian economy is expected to be around 2.5% annually until 2020.

Table 11.2 indicates that the projected growth of major industries will be about 2.4% until 2020. Interest and inflation rates are expected to remain low (around 6.0% and 2.9%) on average over the same period. Informatica also projects an average growth of 3.1% in the industrial sector compared with 2.2% in the services sector. Since the industrial sector is energy intensive, a higher demand for electricity is expected.

Canada economic future will depend mainly on the performance of exports due to the significant anticipated growth in the industrial sector. Increased competitiveness of Canadian industries will be the driving force for improved performance of exports. Since energy-intensive industries are expected to have a high growth, there will be great opportunities for cogeneration and combined cycles.

Informatica economic projections indicate that the maritime provinces are expected to have a relatively lower economic growth while Ontario and the western provinces have a relatively higher economic growth.

Major Industries	1982/1988	1989/1993	1994/1998*	1994/2000*	2000/2020*
Pulp and paper	4.3	-1.1	2.1	2.2	2.3
Iron and steel	8.3	-0.7	3.3	2.7	1.9
Smelting and refining	8.9	3.2	4.2	3.6	2.3
Chemicals	7.0	0.3	2.9	2.4	2.8
Total	5.9	-1.9	5.3	4.7	2.9
Manufacturing					
Total industrial	5.0	-1.6	4.6	4.1	2.4
Total economy	4.3	0.6	3.3	3.0	2.2

Source: Informetrica Limited/NRCan, December 1993.

*Projections.

TABLE 11.2 Average Annual Growth Projections by Major Industries

11.4 Financial Analysis

This example illustrates the feasibility of installing a combined-cycle plant near an industry which operates 8500 hours (364 days) per year. The energy requirements of the industry are

1. 200 MW of electricity.
2. 200,000-lb/h process steam at 175 psig and 400°F.

11.5 Base Case

The local utility supplies electricity to the plant at 6.0 ¢/kWh. The steam is generated within the plant using natural gas-fired boilers. The local gas utility supplies gas to the plant at \$3.25/mcf.

The plant energy costs are:

$$\begin{aligned} \text{Electricity cost} &= 200,000 \text{ kW} \times 6.0 \text{ ¢/kWh} \times 8500 \text{ h/yr} \\ &= \$102.0 \text{ million/yr} \end{aligned}$$

$$\begin{aligned} \text{Natural gas cost} &= (200,000 \text{ lb/h} \times 1018 \text{ Btu/lb}) / (82\% \times 1015 \text{ Btu/scf}) \\ &= 244,623 \text{ scf/h} \\ &= 244.623 \text{ mcf/h} \\ &= 244.623 \text{ mcf/hr} \times \$3.25/\text{mcf} \times 8500 \text{ h/yr} \\ &= \$6.8 \text{ million/yr} \end{aligned}$$

Unit process steam cost	$\$6.8 \text{ million/year} / (200,000 \text{ lb/h} \times 8500 \text{ hours})$ $= \$3.98 \text{ per thousand lb of process steam}$
Total energy costs	$\$102.08 + \$6.8 \text{ million/yr}$ $= \$108.8 \text{ million/yr}$

A nonutility generator (NUG) proposed to generate the electrical and steam loads of the plant at a lower cost. The plant management would sign a contract with the NUG if power is produced at 5.5 ¢/kWh and steam at \$3.25 per 1000 lb/h.

11.6 Combined-Cycle Configuration

The NUG proposed to produce both loads using a combined-cycle plant. Four different Westinghouse gas turbines (W251B12, 501D5, 501D5A, and 501F) were considered.

The following four configurations were proposed:

1. W251B12, two combustion turbine and single steam turbine
2. 501D5, single combustion turbine and single steam turbine
3. 501D5A, single combustion turbine and single steam turbine
4. 501F, single combustion turbine and single steam turbine

Table 11.3 summarizes the performance estimations of the proposed configurations. Figures 11.1 to 11.4 show the results of the heat balances performed for each configuration.

11.7 Capital Cost

Gryphon International Consultants provided estimates of the capital cost for each of the four configurations. These estimates are based on projects which they have estimated their capital cost or built a comparable plant for in the last 5 years.

Figure 11.5 illustrates the variation in total project cost for combined-cycle plants (this information was obtained from Black and Veatch which designed and built power plants around the world. They are located in Kansas City, Missouri).

Note that Solar Turbines and Black and Veatch have developed the instant power stations (IPS). These plants have a delivery schedule of 8 to 12 months while normal plants require at least 2 years. Their capital and operating and maintenance costs are about 20% lower those of than normal plants. These advantages are obtained by using standardized plant designs as well as pre-engineered, prefabricated, and skid-mounted standard components to facilitate fast track construction and maintenance operation. Their range is 15–50 MW.

11.8 Operating and Maintenance Cost

The annual recurring operating and maintenance costs for these plants include

- Electrical utility interconnection
- The cost of utility-supplied supplemental, standby, interruptible, and maintenance power

- The cost of conventional fuel
- Insurance costs
- Federal and provincial income taxes
- Federal and provincial sales tax
- Local property taxes

Combustion Turbine Conditions					
Plant configuration		W251B12— 2 × 1	501D5— 1 × 1	501D5A— 1 × 1	501F— 1 × 1
Fuel type		Natural gas	Natural gas	Natural gas	Natural gas
Fuel heating value (LHV)		21,520	21,520	21,520	21,520
Emission control method		Dry low NOx	Dry low NOx	Dry low NOx	Dry low NOx
Steam Turbine Conditions					
Steam turbine condenser		Yes	Yes	Yes	Yes
Process steam extraction		Yes	Yes	Yes	Yes
Process steam flow	lb/h	200,000	200,000	200,000	200,000
Process steam pressure	psia	175	175	175	175
Process steam temp	°F	400	400	400	400
Case Variables					
Ambient temperature (ISO)	°F	59	59	59	59
Relative humidity (ISO)	%	60	60	60	60
Barometric pressure (ISO)	psia	14.696	14.696	14.696	14.696
ISO CT inlet loss	inH ₂ O	4.5	4.5	4.5	4.5
ISO CT exhaust loss	inH ₂ O	10.0	10.0	10.0	10.0
ST exhaust pressure	inH ₃ A	1.5	1.5	1.5	1.5
Estimated Plant Performance					
Plant load level	%	100	100	100	100
Gross CT power	MW	99.0	106.1	118.9	172.5
Gross ST power	MW	33.4	37.6	42.3	76.1
Gross plant power	MW	132.4	143.7	161.2	248.6
Plant aux. load (est.)	MW	3.3	3.6	4.0	6.2
New plant power	MW	129.1	140.0	157.2	242.4
CT heat input	MMBtu/h	1029	1071	1176	1609
Net heat rate (LHV)	Btu/kWh	8035	7670	7535	6595
NOx @ 15% O ₂ ISO	ppm	25 or less	25 or less	25 or less	25 or less

TABLE 11.3 Combined-Cycle Performance Estimation

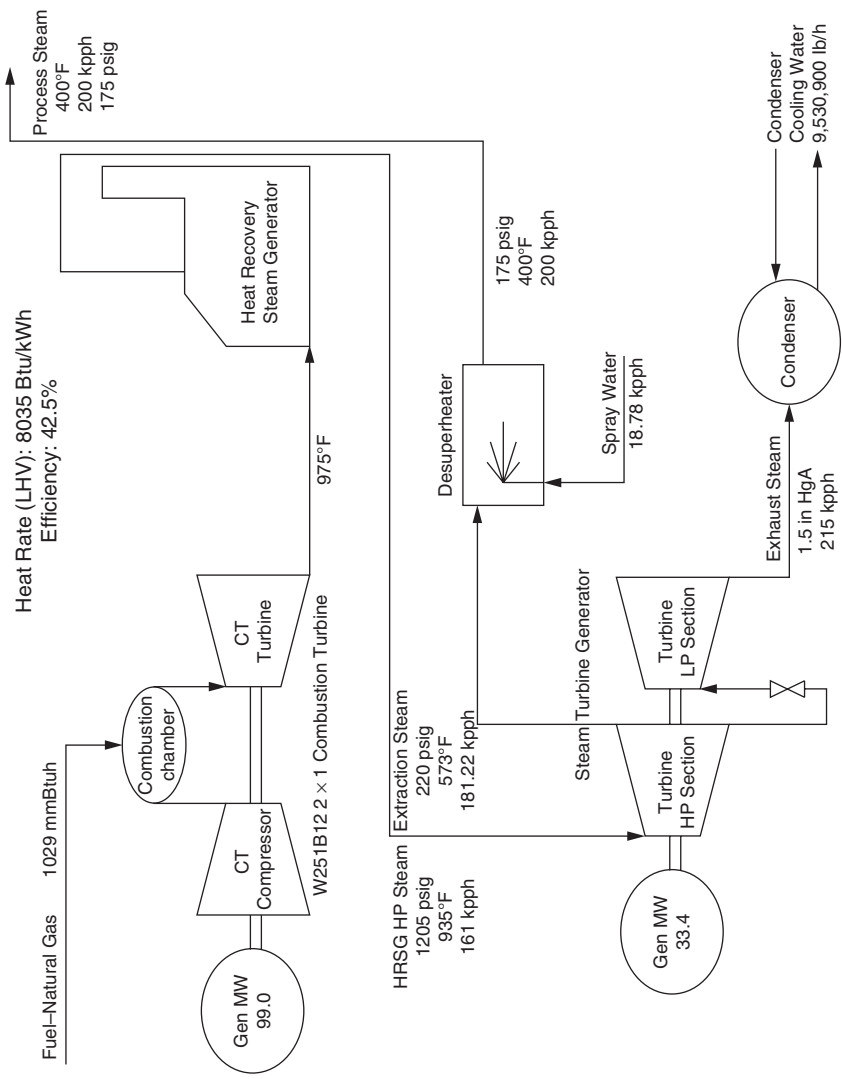


FIGURE 11.1 W252B12 2 x 1 Combined-cycle heat balance. (Source: Reprinted with permission from Michael Cucuz, *The Economics of Combined Cycle Cogeneration Plants*, Westinghouse Power Manufacturing Division, Hamilton, 1997.)

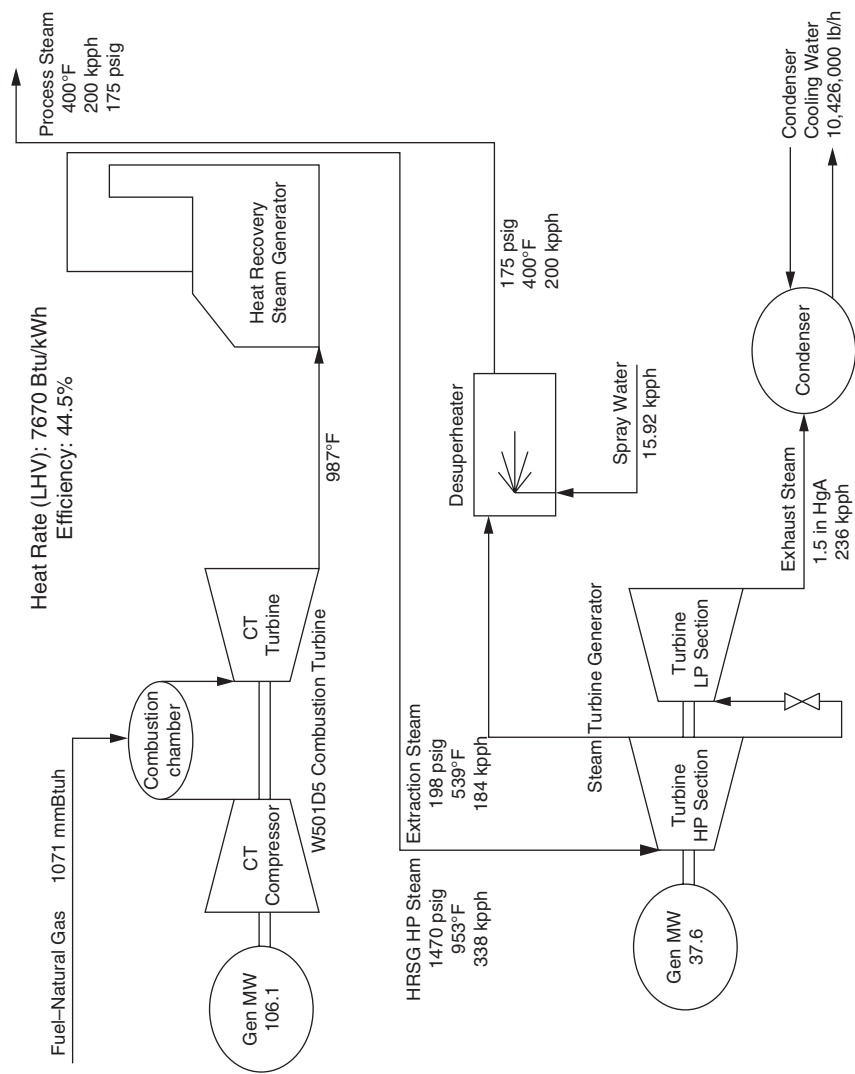


FIGURE 11.2 W501D5 1 × 1 Combined-cycle heat balance. (Source: Reprinted with permission from Michael Cucuz, *The Economics of Combined Cycle Cogeneration Plants*, Westinghouse Power Manufacturing Division, Hamilton, 1997.)

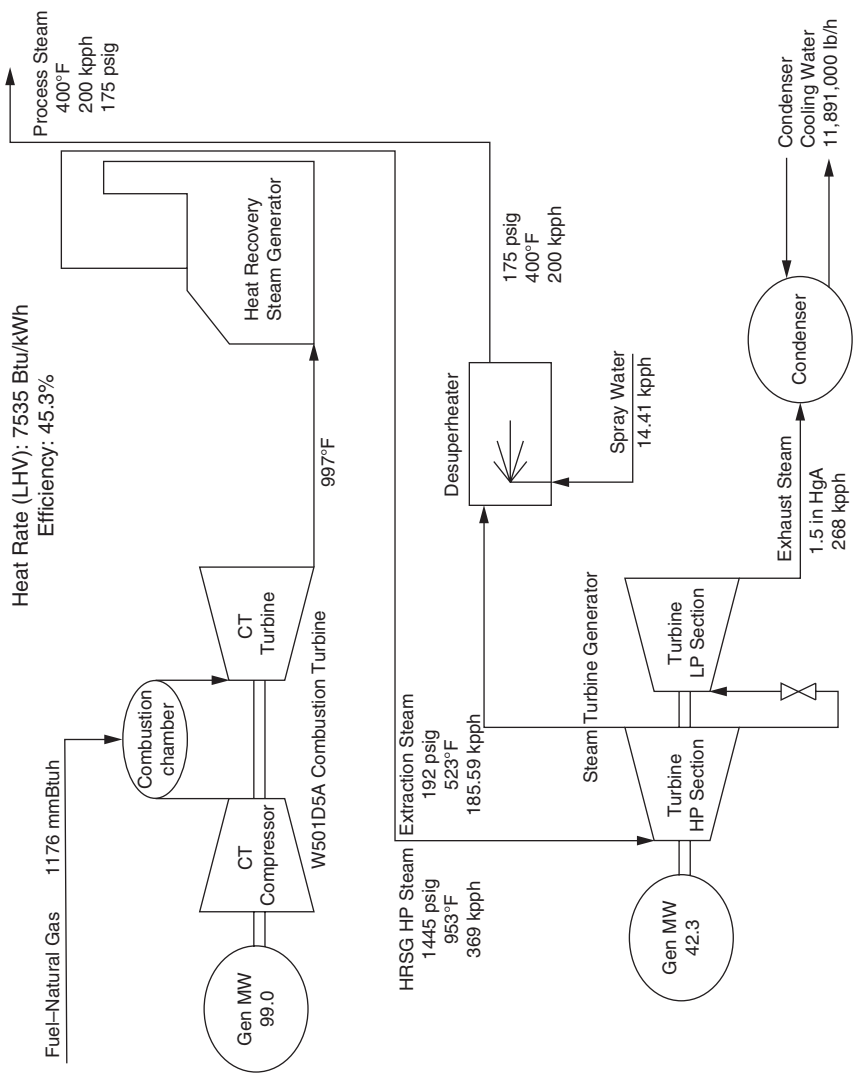


FIGURE 11.3 W501D5A 1 × 1 Combined-cycle heat balance. (Source: Reprinted with permission from Michael Cucuz, *The Economics of Combined Cycle Cogeneration Plants*, Westinghouse Power Manufacturing Division, Hamilton, 1997.)

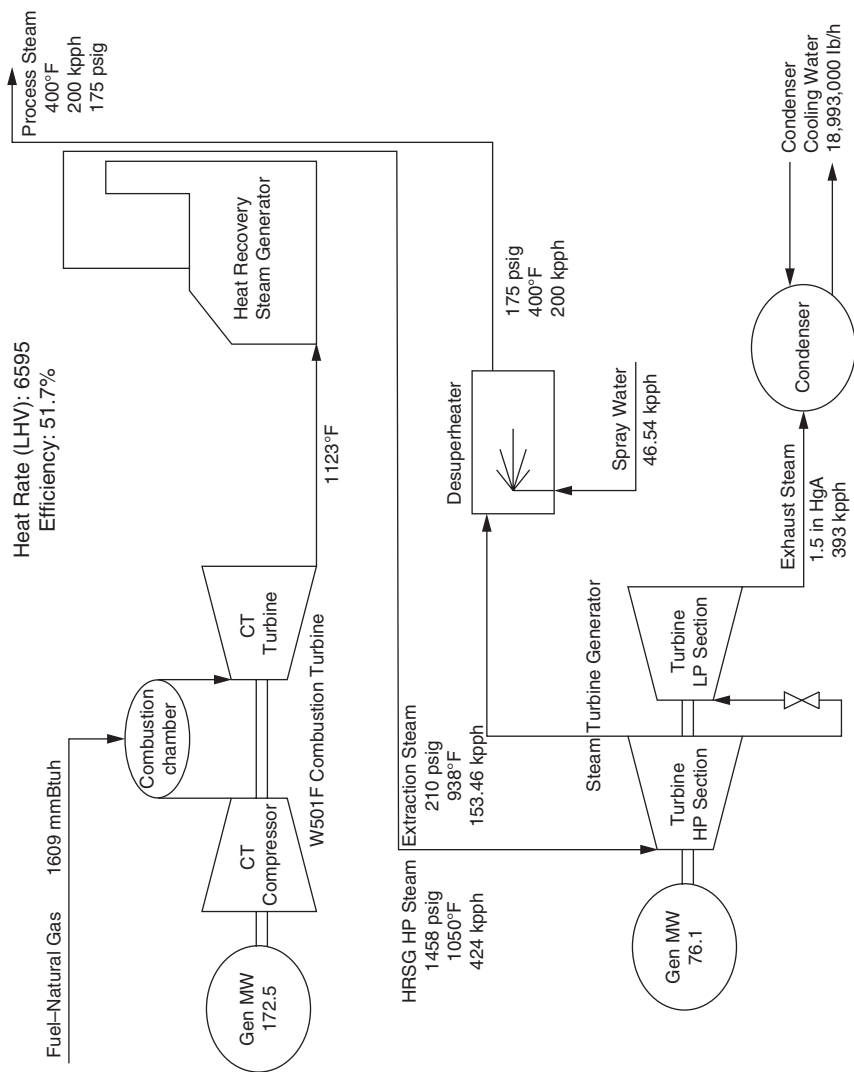


FIGURE 11.4 W501F 1 × 1 Combined-cycle heat balance. (Source: Reprinted with permission from Michael Cucuz, *The Economics of Combined Cycle Cogeneration Plants*, Westinghouse Power Manufacturing Division, Hamilton, 1997.)

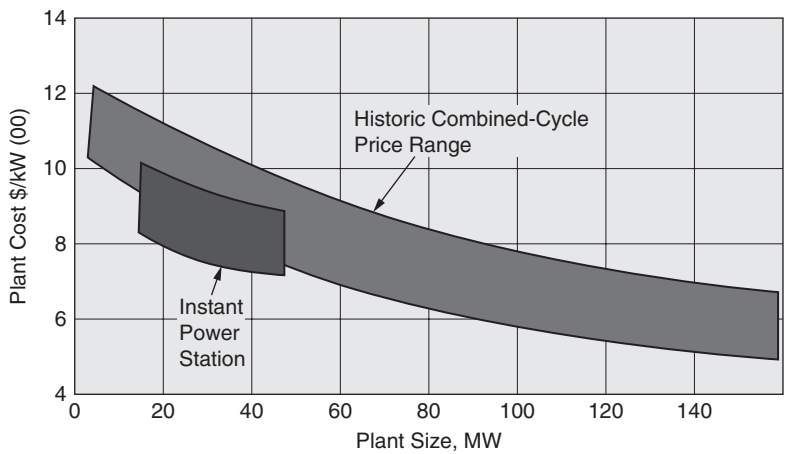


FIGURE 11.5 Combined-cycle project cost. (Source: Reprinted with permission from Michael Cucuz, *The Economics of Combined Cycle Cogeneration Plants*, Westinghouse Power Manufacturing Division, Hamilton, 1997.)

- Staffing
- Space requirements
- Maintenance, both scheduled and unscheduled
- Administrative and management costs

Figure 11.6 illustrates typical annual operating and maintenance costs in \$/kW based on operating hours of the plant.

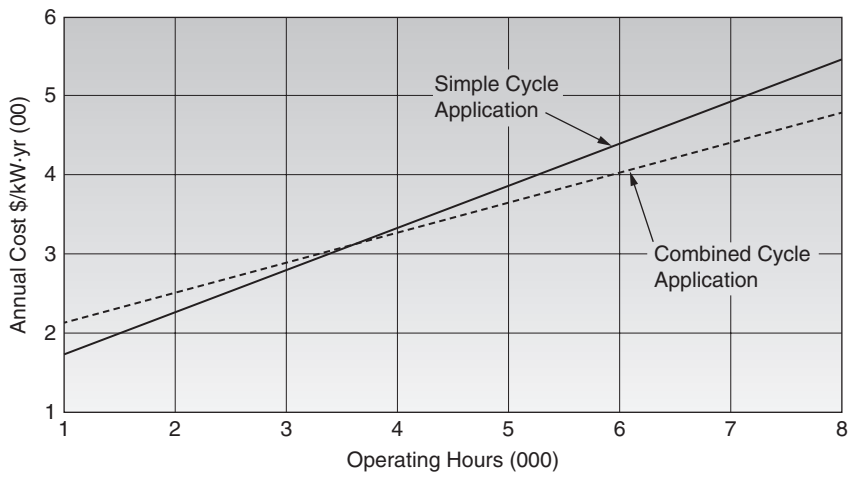


FIGURE 11.6 Combined-cycle operating costs. (Source: Reprinted with permission from Michael Cucuz, *The Economics of Combined Cycle Cogeneration Plants*, Westinghouse Power Manufacturing Division, Hamilton, 1997.)

11.9 Economic Evaluation of Different Combined-Cycle Configurations

Table 11.4 lists the assumptions that were made to evaluate each configuration (the definitions of some of the terms are presented in Sec. 11.3). All the values are presented in Canadian dollars. In addition, the following assumptions were made:

1. The initial fuel cost is \$3.25/mcf increasing at 2.5% per year.
2. The contracted electricity cost is 5.5 ¢/kWh increasing at 3% per year.
3. The cost of steam is \$3.25/1000 lb increasing at 2.5% per year.

General Information				
Configuration	W251B12	501D5	501D5A	501F
Number of combustion turbines	2	1	1	1
Number of steam turbines	1	1	1	1
Construction period, months	24	24	24	24
1st Operating year	2000	2000	2000	2000
NPV reference year	1997	1997	1997	1997
Capital Costs (\$5000)				
Turnkey construction cost	88,000	80,000	85,000	100,000
Owner's contingency	4,450	4,050	4,300	5,050
Utility interconnections	1,000	1,000	1,000	1,000
Pre-C.O. operating costs	2,000	2,000	2,000	2,000
Permitting	1,000	1,000	1,000	1,000
Working capital	3,160	3,350	3,630	4,870
Construction interest	10,090	9,220	9,770	11,450
Financing cost	3,194	3,012	3,134	3,507
Total capital cost	112,894	103,632	109,834	128,877
Depreciation	% of Total	% of Total	% of Total	% of Total
Personal property combined cycle	34	34	34	34
Class 43	64	64	64	64
Non-depreciable	2	2	2	2
Performance				
Capacity—average, MW	129.1	140.0	157.2	242.4
Capacity—contract, MW	129.1	140.0	157.2	200.0
Heat rate—HHV, Btu/kWh	7,877	7,520	7,387	6,466
Capacity factor	92.0%	92.0%	92.0%	92.0%
Capacity degradation	3.0%	3.0%	3.0%	3.0%
Heat rate degradation	2.0%	2.0%	2.0%	2.0%
Steam sold to host (1000 #/h)	200	200	200	200

TABLE 11.4 Reference Plant Economic Assumptions

Financing				
Construction period interest	8.00%	8.00%	8.00%	8.00%
Long-term loan	(Mortgage	(Mortgage	(Mortgage	(Mortgage
Debt, %	style)	style)	style)	style)
Term, yrs.	80%	80%	80%	80%
Interest rate	15	15	15	15
Required reserve fund (\$000)	6.50%	6.50%	6.50%	6.50%
Reserve fund, % available income	4.803	4.803	4.803	4.803
Reserve fund interest rate	25%	25%	25%	25%
Federal income tax	4.0%	4.0%	4.0%	4.0%
Provincial income tax	25.0%	25.0%	25.0%	25.0%
Discount rate	12.0%	12.0%	12.0%	12.0%
Financing costs, % capital	12.0%	12.0%	12.0%	12.0%
	2.0%	2.0%	2.0%	2.0%
Revenues				
Capacity payment (\$/kW · yr)	363.60	363.60	363.60	440.67
Steam price (\$/MMBtu)	3.25	3.25	3.25	3.25
Fuel Costs				
Commodity (\$/MMBtu)	2.30	2.30	2.30	2.30
Transportation (\$/MMBtu)	1.00	1.00	1.00	1.00
Annual Operating Costs				
Fixed O & M (\$000)	3,990	5,000	5,000	6,000
Variable O & M (¢/kWh)	0.19	0.19	0.19	0.21
Rent/misc. (\$000)	500	500	500	500
Property taxes (\$000)	500	500	500	500
Insurance	2,000	2,000	2,000	2,500

TABLE 11.4 Reference Plant Economic Assumptions (Continued)

Table 11.5 illustrates the project results including the internal rate of return for the four different configurations. The internal rate of return is the percentage rate that equates the present value of an expected future series of cash flows to the initial investment.

Table 11.6 shows the results of the analysis for the first year of operation.

This analysis proves that the largest combined-cycle plant (501F) provides the highest rate of return. Since the generating capacity of the 501F configuration is more than the requirements of the industrial plant, the additional power can be sold to the local municipality or to another facility.

The plant has also reduced their energy cost:

Electricity cost

$$200,000\text{ kW} \times 5.5\text{¢/kWh} \times 8500\text{ h/yr}$$
$$= \$93.5\text{ million/yr}$$

Natural gas cost

$$\$3.25/\text{thousand lb of steam} \times 200,000\text{ lbs/h} \times 8500\text{ h/yr}$$
$$= \$5.5\text{ million/yr}$$

PROJECT RESULTS									
1997 Net Present Value @ 15% Discount Rate (\$000)		W251B12	501D5	501D5A	501F				
25 Years		\$48,706	\$62,248	\$77,237	\$164,296				
20 Years		\$41,147	\$53,932	\$67,564	\$147,364				
15 Years		\$29,773	\$41,444	\$53,042	\$121,940				
10 Years		\$20,555	\$29,945	\$38,924	\$92,789				
Internal Rate of Return		Cash/Cash	Equity	Cash/Cash	Equity	Cash/Cash	Equity	Cash/Cash	Equity
25 Years		18.3%	42.2%	21.7%	53.5%	23.7%	60.2%	34.7%	96.0%
20 Years		17.7%	42.1%	21.2%	53.4%	23.3%	60.2%	34.5%	96.0%
15 Years		16.1%	41.7%	20.0%	53.3%	22.2%	60.1%	34.1%	96.0%
10 Years		11.5%	40.4%	16.1%	52.2%	18.7%	59.5%	31.9%	95.9%
Performance									
Annual energy generation, MWh		1,009,229		1,094,439		1,228,899		1,894,944	
Annual fuel required, MMBtu		8,109		8,395		9,259		12,498	
Average capacity—degraded, MW		125.2		135.8		152.5		235.1	
Heat Rate—degraded, Btu/kWh		8,035		7,670		7,535		6,595	
Annual steam sold, Milb		1,611,840		1,611,840		1,611,840		1,611,840	
Financing									
Amount borrowed (\$000)		\$90,315		\$82,906		\$87,867		\$103,102	
Annual debt service (\$000)		\$9,605		\$8,817		\$9,345		\$10,965	
Average power escalation		3.00%		3.00%		3.00%		3.00%	
Capital cost/installed kW (\$/kW)		\$874		\$740		\$699		\$532	
Levelized Power Pries @ 15% Discount Rate									
10 Years		5.500 ¢/kWh		5.500 ¢/kWh		5.500 ¢/kWh		5.500 ¢/kWh	
15 Years		5.753 ¢/kWh		5.753 ¢/kWh		5.753 ¢/kWh		5.753 ¢/kWh	
20 Years		5.960 ¢/kWh		5.960 ¢/kWh		5.960 ¢/kWh		5.960 ¢/kWh	
25 Years		6.124 ¢/kWh		6.124 ¢/kWh		6.124 ¢/kWh		6.124 ¢/kWh	

TABLE 11.5 Reference Plant Analysis Results

Combustion Turbine Type		W251B12	501D5	501D5A	501F
Number of CTs		2	1	1	1
Number of STs		1	1	1	1
Cogeneration Plant Performance					
Net plant output	MW	129.1	140.0	157.2	242.4
	MW · h/yr	1,009,229	1,094,439	1,228,899	1,894,944
Steam produced	lb/h	200,000	200,000	200,000	200,000
	1000 lb/yr	1,611,840	1,611,840	1,611,840	1,611,840
Total fuel used	MMBtu	8,109	8,395	9,259	12,498
Project Capital Cost					
Total	\$million	112.9	103.6	109.8	128.9
First Year Operation Gross Income					
Electricity	\$million/yr	49.8	54.0	60.6	93.4
Steam	\$million/yr	5.6	5.6	5.6	5.6
Total	\$million/yr	55.4	59.6	66.2	99.0
Costs					
Fuel	\$million/yr	28.8	29.8	32.9	44.4
Equipment O&M	\$million/yr	5.7	7.0	7.3	10.8
Operating staff	\$million/yr	0.7	0.7	0.7	0.7
Debt and interest	\$million/yr	9.6	8.8	9.3	11.0
Insurance	\$million/yr	2.1	2.1	2.1	2.6
Rent/misc	\$million/yr	1.0	1.0	1.0	1.0
Total	\$million/yr	47.9	49.4	53.3	70.5
Net Income	\$million/yr	7.5	10.2	12.9	28.5
Net Revenues/Total Revenues		13.3%	16.9%	19.4%	29.4%

TABLE 11.6 Technical and Financial Comparison of Five Combined-Cycle Plants

New total energy costs

$$\begin{aligned} & \$93.5 \text{ million/yr} + \$5.5 \text{ million/yr} \\ & = \$99.0 \text{ million/yr} \end{aligned}$$

Therefore, the industrial plant will save \$9.8 million per year on their total energy cost. The NUG and the industrial plant has benefited from this project. The reduction in the cost of plant energy will be reflected in their products. This will make them more competitive. The complete financial analysis over a 25-year period is presented in Sec. 11.14.

If this plant is built under the current legislation (regulated), the NUG, would have to sell the electric power to the provincial utility at its avoided cost (3.0 ¢/kWh). Table 11.7 shows the results of the economical evaluation. It indicates that there are

REVISED PROJECT RESULTS									
1997 Net Present Value @ 15% Discount Rate (5000)		W251B12	501D5	501D5A	501F				
25 Years		(\$65,352)	(\$61,657)	(\$61,972)	(\$50,962)				
20 Years		(\$64,442)	(\$60,788)	(\$61,332)	(\$51,992)				
15 Years		(\$62,940)	(\$59,313)	(\$60,176)	(\$53,241)				
10 Years		(\$51,766)	(\$48,835)	(\$49,663)	(\$44,710)				
Internal Rate of Return		Cash/Cash	Equity	Cash/Cash	Equity	Cash/Cash	Equity	Cash/Cash	Equity
25 Years		ERR	ERR	ERR	ERR	ERR	ERR	ERR	ERR
20 Years		ERR	ERR	ERR	ERR	ERR	ERR	ERR	ERR
15 Years		ERR	ERR	ERR	ERR	ERR	ERR	ERR	ERR
10 Years		ERR	ERR	ERR	ERR	ERR	ERR	ERR	ERR
Performance									
Annual energy generation, MWh		1,009,229	1,094,439	1,228,899	1,894,944				
Annual fuel required, MMBtu		8,109	8,395	9,259	12,498				
Average capacity—degraded, MW		125.2	135.8	152.5	235.0				
Heat rate—degraded, Btu/kWh		8,035	7,670	7,535					
Annual steam sold, Mlb		1,611,840	1,611,840	1,611,840	1,611,840				
Financing									
Amount borrowed (\$000)		\$90,315	\$82,906	\$87,867	\$103,102				
Annual debt service (\$000)		\$9,605	\$8,817	\$9,345	\$10,965				
Average power escalation		3.00%	3.00%	3.00%	3.00%				
Capital cost/installed kW (\$/kW)		\$874	\$740	\$699	\$532				
Levelized Power Price @ 15% Discount Rate									
10 Years		3.630 ¢/kWh	3.630 ¢/kWh	3.630 ¢/kWh	3.630 ¢/kWh				
15 Years		3.798 ¢/kWh	3.798 ¢/kWh	3.798 ¢/kWh	3.798 ¢/kWh				
20 Years		3.934 ¢/kWh	3.934 ¢/kWh	3.934 ¢/kWh	3.934 ¢/kWh				
25 Years		4.042 ¢/kWh	4.042 ¢/kWh	4.042 ¢/kWh	4.042 ¢/kWh				

TABLE 11.7 Revised Plant Analysis Results

no profits to be made. In reality, there are losses (the terms between parentheses are negative). Thus, no one would be willing to build a plant under a regulated legislation. This is the reason for having only a few large combined-cycle plants in Canada. They have been built due to concessions by the provincial utilities. These concessions have been in the form of higher capacity payments for electricity or attractive financing rates that resulted in acceptable rate of return for the NUG. A provincial utility offers these incentives usually when it is cost prohibitive to build a large station in a remote area.

11.10 Electricity Purchase Rate

All Canadian utilities allow independent power producers to interconnect with their grid. However, the electricity purchase rate by most utilities and their requirements discourage investors from generating power.

Most utilities in Canada are planning to build new plants or repower (convert fossil-based steam turbine plants to combine cycles) older plants over the next 10 years. These plans will be implemented despite the fact that most utilities have considerable surplus generating capacity. The cost of power produced by utilities is expected to be significantly higher than the purchase rate they are currently prepared to offer independent power producers.

Canadian utilities are unlikely to offer full avoided cost to independent power producers voluntarily. Like most monopolies, they are reluctant to encourage competition and lose market share. It is evident that rate payers would benefit from deregulation. Since international competition and free trade are encompassing every aspect of manufacturing, it is inevitable to have full deregulation of the electrical generation industry.

11.11 Economic Consideration

The Canadian Capital Cost Allowance Class 43.1 write-down is a significant incentive for building cogeneration and combined-cycle plants. The Ministry of Energy, Mines, and Resources will calculate the heat rate of a plant to determine its eligibility. If the heat rate is lower than 7000 Btu/kWh for electric energy produced, the plant will be entitled for a significant tax benefit. It consists of an accelerated write-off of 25, 50, and 25% over the first 3 years. There are also speculations that the Ministry is considering lowering this heat rate limit to 6000 Btu/kWh in the future.

11.12 Conclusions

During the last decade, major industries have been deregulated or privatized across the world. This trend is increasing as industries strive to become more competitive. In Canada, gas lines, telecommunications, railways, banks, and insurance companies have been deregulated. Electric utilities such as Nova Scotia Power have been privatized.

Deregulation and privatization resulted in increasing choice and lower prices to consumers due to increased competition. The number of combined cycles will increase dramatically after fully implementing deregulation across North America.

This is due to the significant economical and environmental advantages that combined-cycle and cogeneration plants have over conventional power plants. The size of current utilities will decrease as their market share diminishes.

11.13 Bibliography

Cucuz, M., *The Economics of Combined Cycles Cogeneration Plants*, Westinghouse Power Manufacturing Division, Hamilton, 1997.

11.14 Appendix: Definitions of Terms Used in the Tables

Term	Definition
1. Utility interconnection cost	The cost of building a substation which will stepup the voltage from the power plant to the grid voltage
2. Pre-C.O. operating cost	The cost of operating the plant before starting to produce power
3. Permitting cost	The cost of building permit, environmental permits (NOx levels, noise levels, etc.)
4. Working capital	The portion of the investment represented by assets less liabilities.
5. Financing cost	Interest paid on borrowed funds
6. Depreciation: Personal property combined cycles, class 43, non-depreciable	The different rates of depreciation depending on the equipment
7. 1997 Net present value @ 15% discount rate	The amount of return from the project in 1997 money based on 15% risk factor
8. Non-depreciable	Items such as labor and consumables

11.15 Appendix: Financial Analysis of the Different Configurations of Combined-Cycle Plants

PROJECT ASSUMPTIONS				
General Information		Financing		
Project Name:	ABC Canada Company Ltd. Somewhere, Canada	Construction Period Interest Rate:	8.00%	
Configuration:	W251B12-NUG Sells to End User	Long-Term Loan Debt, % Term, yr Interest Rate	(Mortgage Style) 80% 15 6.50%	
Construction Period, Months	30	Required Reserve Fund (\$000)	4,803	
1st Operating Year	2000	Reserve Fund, % Available Income	25%	
NPV Reference Year	1997	Reserve Fund Interest Rate	4.0%	
Capital Cost (\$000)		Federal Income Tax	25.00%	
Turnkey Construction Cost	88,000	State/Province Income Tax	12.00%	
Owner's Contingency	4,450	Discount Rate	12.0%	
Utility Interconnections	1,000	Financing Costs, % Capital	2.00%	
Other	0	Revenues	1997	Esc %/yr
Pre-C.O. Operating Costs	2,000	Capacity Payment (\$/kW · yr)	363.60	3.00%
Permitting	1,000	Energy Payment (¢/kWh)	0.00	4.00%
Working Capital	3,160	Fixed O&M Payment (\$/kW · yr)	0.00	2.50%
Development Fee	0	Variable O&M Payment (¢/kWh)	0.00	2.50%
Construction Interest	10,090	Steam Price (\$/1000 lb)	\$3.25	2.50%
Financing Costs	3,194	Fuel Costs	1997	Esc %/yr
Total Capital Cost	112,894	Commodity (\$/MMBtu)	2.30	2.50%
Depreciation	% of Total	Transportation (\$/MMBtu)	1.00	2.50%
Personal Property Combined Cycle	34.0%	Annual Operating Costs	1997	Esc %/yr
Personal Property Simple Cycle	0.0%	Fixed O&M (\$000)	3,990	3.00%
Real Property	0.0%	Variable O&M (¢/kWh)	0.19	2.50%
Financing, Legal, Closing, Dev. Costs	0.0%	Rent/Misc. (\$000)	500	2.00%
Other (e.g., Class 34)	64.0%	Property Taxes (\$000)	500	2.00%
Non-depreciable	2.0%	Insurance (\$000)	2,000	2.00%
Performance		Filename		
Capacity—Average, MW	129.1	H:\USERS\MIKEC\PEO\W251B12.WK4		
Capacity—Contract, MW	129.1			
Heat Rate—HHV, Btu/kWh	7,877			
Capacity Factor	92.0%			
Capacity Degradation	3.0%			
Heat Rate Degradation	2.0%			
Steam Sold to Host (1000 #/h)	200.0			

PROJECT RESULTS		
1997 Net Present Value @ 12% Discount Rate (\$000)		
25 Years	\$48,706	
20 Years	\$41,147	
15 Years	\$29,773	
10 Years	\$20,555	
Internal Rate of Return	Cash/Cash	Equity
25 Years	18.3%	42.2%
20 Years	17.7%	42.1%
15 Years	16.1%	41.7%
10 Years	11.5%	40.4%
Debt Coverage Ratios		
Average	2.35	
Minimum	1.77	
Performance		
Annual Energy Generation, MWh	1,009,229	
Annual Fuel Required, MMBtu	8,109	
Average Capacity—Degraded, MW	125.2	
Heat Rate—Degraded, Btu/kWh	8,035	
Annual Steam Sold, Mlb	1,611,840	
Financing		
Amount Borrowed (\$000)	\$90,315	
Annual Debt Service (\$000)	\$9,605	
Average Power Escalation	3.00%	
Capital cost/Installed kw (\$/kW)	\$874	
Levelized Power Price @ 12% Discount Rate		
10 Years	5.500 ¢/kWh	
15 Years	5.753 ¢/kWh	
20 Years	5.960 ¢/kWh	
25 Years	6.124 ¢/kWh	

INCOME STATEMENT (\$000)											
	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024	Totals
Capacity Payments		49,755	51,247	52,785	54,368	55,999	64,919	75,258	87,245	101,141	1,814,017
Energy Payments		0	0	0	0	0	0	0	0	0	0
Fixed O&M Payments		0	0	0	0	0	0	0	0	0	0
Variable O&M		0	0	0	0	0	0	0	0	0	0
Total Electric Revenue		49,755	51,247	52,785	54,368	55,999	64,919	75,258	87,245	101,141	1,814,017
Total Steam Revenue		5,641	5,782	5,927	6,075	6,227	7,045	7,971	9,018	10,204	0
Total Revenues		55,396	57,030	58,712	60,443	62,226	71,964	83,229	96,263	111,344	2,006,710
Total Revenues (cts/kWh)		5.49	5.65	5.82	5.99	6.17	7.13	8.25	9.54	11.03	
OPERATING EXPENSES	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024	Totals
Fuel Commodity Charge		20,084	20,586	21,101	21,628	22,169	25,082	28,378	32,107	36,326	686,025
Fuel Transportation Charge		8,732	8,950	9,174	9,404	9,639	10,905	12,338	13,960	15,794	298,272
Fixed Operations and Maintenance		4,360	4,491	4,626	4,764	4,907	5,689	6,595	7,645	8,863	158,962
Variable Operations and Maintenance		2,065	2,117	2,170	2,224	2,279	2,579	2,918	3,301	3,735	70,535
Miscellaneous Fees		531	541	552	563	574	634	700	773	853	16,995
Property Taxes		531	541	552	563	574	634	700	773	853	16,995
Insurance		2,122	2,165	2,208	2,252	2,297	2,536	2,800	3,092	3,414	67,982
Total Expenses		38,425	39,391	40,382	41,398	42,440	48,060	54,430	61,651	69,839	1,315,765
Total Expenses (cts/kWh)		3.81	3.90	4.00	4.10	4.21	4.76	5.39	6.11	6.92	
1999		2000	2001	2002	2003	2004	2009	2014	2019	2024	Totals
OPERATING INCOME	(112,894)	16,971	17,638	18,329	19,045	19,786	23,904	28,799	34,612	41,505	578,051
Cash on Cash I											
10 Yr.		15 Yr.	20 Yr.	25 Yr.							
11.5%		16.1%	17.7%	18.3%							

PROJECT CASH FLOWS (\$000)												
	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024	Totals	
OPERATING INCOME		16,971	17,638	18,329	19,045	19,786	23,904	28,799	34,612	41,505	690,945	
+ Reserve Fund Interest		37	114	174	192	192	192	96	0	0	2,534	
– Depreciation		19,502	38,897	20,626	2,371	2,193	1,713	1,713	1,713	0	110,636	
– Interest Payment		5,870	5,628	5,369	5,094	4,801	3,022	586	0	0	53,764	
INCOME BEFORE TAXES		(8,365)	(26,772)	(7,492)	11,772	12,984	19,361	26,597	32,900	41,505	529,079	
– Principal Payment		3,735	3,978	4,236	4,511	4,805	6,583	9,019	0	0	90,315	
+ Depreciation		19,502	38,897	20,626	2,371	2,193	1,713	1,713	1,713	0	110,636	
– Equity Investment	22,579	0	0	0	0	0	0	0	0	0	22,579	
– Reserve Fund		1,851	2,037	915	0	0	0	(4,803)	0	0	0	
PRE-TAX CASH FLOW	(22,579)	5,552	6,110	7,983	9,632	10,373	14,491	24,093	34,612	41,505	526,821	
– Tax Payment @	34.00%	(2,844)	(9,103)	(2,547)	4,003	4,415	6,583	9,043	11,186	14,112	179,887	
AFTER TAX CASH FLOW	(22,579)	8,396	15,213	10,530	5,629	5,958	7,908	15,050	23,426	27,393	346,934	
			10 Yr.	15 Yr.	20 Yr.	25 Yr.						
	EQUITY IRR		40.4%	41.7%	42.1%	42.2%						
	1997 NPV @ 12%		20,555	29,773	41,147	48,706						
	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024		
Cash Available to Service Debt		17,008	17,752	18,503	19,237	19,978	24,096	28,896				
Total Debt Service		9,605	9,605	9,605	9,605	9,605	9,605	9,605				
COVERAGE RATIOS		1,771	1,848	1,926	2,003	2,080	2,509	3,008	*****	*****		
SUMMARY	Avg.-25 Yr.	2000	2001	2002	2003	2004	2009	2014	2019	2024		
Reserve Fund		1,851	3,887	4,803	4,803	4,803	4,803	0	0	0		
Total Revenues	68,436	55,396	57,030	58,712	60,443	62,226	71,964	83,229	96,263	111,344		
Total Costs (Including Debt Service)	55,599	48,030	48,997	49,988	51,004	52,046	57,665	64,035	61,651	69,839		
Total Net Revenues	12,837	7,366	8,033	8,724	9,440	10,181	14,299	19,194	34,612	41,505		
Net Revenues/Total Revenues	18.38%	13.30%	14.09%	14.86%	15.62%	16.36%	19.87%	23.06%	35.96%	37.28%		

DEPRECIATION AMOUNTS (\$000)		2000	2001	2002	2003	2004	2009	2014	2019	2024
Personal Property—CC	38,384	1,439	2,771	2,563	2,371	2,193	1,713	1,713	1,713	0
Personal Property—SC	0	0	0	0	0	0	0	0	0	0
Real Property	0	0	0	0	0	0	0	0	0	0
Financing, Legal, Closing, etc.	0	0	0	0	0	0	0	0	0	0
Other (e.g., Class 34)	72,252	18,063	36,126	18,063	0	0	0	0	0	0
Non-depreciable	2,258									
Total	112,894	19,502	38,897	20,626	2,371	2,193	1,713	1,713	1,713	0
LONG-TERM DEBT SCHEDULE		2000	2001	2002	2003	2004	2009	2014	2019	2024
Principal Balance		90,315	86,580	82,603	78,367	73,855	46,499	9,019	0	0
Interest Expense		5,870	5,628	5,369	5,094	4,801	3,022	586	0	0
Principal Repayment		3,735	3,978	4,236	4,511	4,805	6,583	9,019	0	0
Total Debt Service		9,605	9,605	\$5	9,605	9,605	9,605	9,605	0	0

PROJECT ASSUMPTIONS				
General Information		Financing		
Project Name:	ABC Canada Company Ltd. Somewhere, Canada	Construction Period Interest Rate:	8.00%	
Configuration:	501D5-NUG Sells to End User	Long-Term Loan Debt, %	(Mortgage Style) 80%	
Construction Period, Months	30	Term, yr	15	
1st Operating Year	2000	Interest Rate	6.50%	
NPV Reference Year	1997	Required Reserve Fund (\$000)	4,409	
		Reserve Fund, % Available Income	25%	
Capital Cost (\$000)		Reserve Fund Interest Rate	4.0%	
Turnkey Construction Cost	80,000	Federal Income Tax	25.00%	
Owner's Contingency	4,050	State/Province Income Tax	12.00%	
Utility Interconnections	1,000	Discount Rate	12.0%	
Other	0	Financing Costs, % Capital	2.00%	
Pre-C.O. Operating Costs	2,000	Revenues	1997	Esc %/yr
Permitting	1,000	Capacity Payment (\$/kW·yr)	363.60	3.00%
Working Capital	3,350	Energy Payment (¢/kWh)	0.00	4.00%
Development Fee	0	Fixed O&M Payment (\$/kW·yr)	0.00	2.50%
Construction Interest	9,220	Variable O&M Payment (¢/kWh)	0.00	2.50%
Financing Costs	3,012			
Total Capital Cost	103,632	Steam Price (\$/1000 lb)	\$3.25	2.50%
Depreciation	% of Total	Fuel Costs	1997	Esc %/yr
Personal Property Combined Cycle	34.0%	Commodity (\$/MMBtu)	2.30	2.50%
		Transportation (\$/MMBtu)	1.00	2.50%
Personal Property Simple Cycle	0.0%	Annual Operating Costs	1997	Esc %/yr
Real Property	0.0%	Fixed O&M (\$000)	5,000	3.00%
Financing, Legal, Closing, Dev. Costs	0.0%	Variable O&M (¢/kWh)	0.19	2.50%
		Rent/Misc. (\$000)	500	2.00%
Other (e.g., Class 34)	64.0%	Property Taxes (\$000)	500	2.00%
Non-depreciable	2.0%	Insurance (\$000)	2,000	2.00%
Performance		Filename		
Capacity—Average, MW	140.0	H:\USERS\MIKEC\PEO\501D5.WK4		
Capacity—Contract, MW	140.0			
Heat Rate—HHV, Btu/kWh	7,520			
Capacity Factor	92.0%			
Capacity Degradation	3.0%			
Heat Rate Degradation	2.0%			
Steam Sold to Host (1000 #h)	200.0			

PROJECT RESULTS		
1997 Net Present Value @ 12% Discount Rate (\$000)		
25 Years	\$62,248	
20 Years	\$53,932	
15 Years	\$41,444	
10 Years	\$29,945	
Internal Rate of Return	Cash/Cash	Equity
25 Years	21.7%	53.5%
20 Years	21.2%	53.4%
15 Years	20.0%	53.3%
10 Years	16.1%	52.5%
Debt Coverage Ratios		
Average	2.84	
Minimum	2.15	
Performance		
Annual Energy Generation, MWh	1,094,439	
Annual Fuel Required, MMBtu	8,359	
Average Capacity—Degraded, MW	135.8	
Heat Rate—Degraded, Btu/kWh	7,670	
Annual Steam Sold, Mlb	1,611,840	
Financing		
Amount Borrowed (\$000)	\$82,906	
Annual Debt Service (\$000)	\$8,817	
Average Power Escalation	3.00%	
Capital cost/Installed kW (\$/kW):	\$740	
Levelized Power Price @ 12% Discount Rate		
10 Years	5.500 ¢/kWh	
15 Years	5.753 ¢/kWh	
20 Years	5.960 ¢/kWh	
25 Years	6.124 ¢/kWh	

PROJECT CASH FLOW (\$000)												
	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024	Totals	
OPERATING INCOME		18,877	19,608	20,364	21,147	21,957	26,457	31,801	38,139	45,647	763,182	
+ Reserve Fund Interest		50	139	176	176	176	176	88	0	0	2,393	
– Depreciation		17,902	35,706	18,934	2,176	2,013	1,572	1,572	1,572	0	101,560	
– Interest Payment		5,389	5,166	4,929	4,676	4,407	2,774	538	0	0	49,353	
INCOME BEFORE TAXES		(4,364)	(21,126)	(3,323)	14,471	15,714	22,287	29,779	36,567	45,647	614,662	
– Principal Payment		3,426	3,651	3,889	4,141	4,411	6,043	8,279	0	0	82,906	
+ Depreciation		17,902	35,706	18,934	2,176	2,013	1,572	1,572	1,572	0	101,560	
– Equity Investment	20,726	0	0	0	0	0	0	0	0	0	20,726	
– Reserve Fund		2,528	1,881	0	0	0	0	(4,409)	0	0	0	
PRE-TAX CASH FLOW	(20,726)	7,583	9,048	11,723	12,506	13,316	17,817	27,481	38,139	45,647	612,589	
– Tax Payment @	34,00%	(1,484)	(7,183)	(1,130)	4,920	5,343	7,578	10,125	12,433	15,520	208,985	
AFTER TAX CASH FLOW	(20,726)	9,066	16,231	12,852	7,586	7,974	10,239	17,356	25,706	30,127	403,604	
			10 Yr.	15 Yr.	20 Yr.	25 Yr.						
			EQUITY IRR	52.5%	53.3%	53.4%	53.5%					
			1997 NPV @ 12%	29,945	41,444	53,932	62,248					
	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024		
Cash Available to Service Debt		18,928	19,746	20,540	21,323	22,133	26,634	31,889				
Total Debt Service		8,817	8,817	8,817	8,817	8,817	8,817	8,817				
COVERAGE RATIOS		2,147	2,239	2,330	2,418	2,510	3,021	3,617	*****	*****		
SUMMARY	Avg.-25 Yr.	2000	2001	2002	2003	2004	2009	2014	2019	2024		
Reserve Fund		2,528	4,409	4,409	4,409	4,409	4,409	0	0	0		
Total Revenues	73,645	59,597	61,356	63,168	65,034	66,954	77,445	89,583	103,630	119,884		
Total Costs (Including Debt Service)	57,603	49,537	50,566	51,622	52,704	53,814	59,805	66,599	65,491	74,237		
Total Net Revenues	16,042	10,060	10,790	11,546	12,329	13,140	17,640	22,984	38,139	45,647		
Net Revenues/Total Revenues	21.44%	16.88%	17.59%	18.28%	18.96%	19.63%	22.78%	25.66%	36.80%	38.08%		

DEPRECIATION AMOUNTS (\$000)		2000	2001	2002	2003	2004	2009	2014	2019	2024
Personal Property—CC	35,235	1,321	2,544	2,353	2,176	2,013	1,572	1,572	1,572	0
Personal Property—SC	0	0	0	0	0	0	0	0	0	0
Real Property	0	0	0	0			0	0	0	0
Financing, Legal, Closing, etc.	0	0	0	0	0	0	0	0	0	0
Other (e.g., Class 34)	66,325	16,581	33,162	16,581	0	0	0	0	0	0
Non-depreciable	2,073									
Total	103,632	17,902	35,706	18,934	2,176	2,013	1,572	1,572	1,572	0
LONG-TERM DEBT SCHEDULE		1999	2001	2002	2003	2004	2009	2014	2019	2024
Principal Balance		82,906	79,478	75,826	71,938	67,796	42,685	8,279	0	0
Interest Expense		5,389	5,166	4,292	4,676	4,407	2,774	538	0	0
Principal Repayment		3,428	3,651	3,889	4,141	4,411	6,043	8,279	0	0
Total Debt Service		8,817	8,817	8,817	8,817	8,817	8,817	8,817	0	0

PROJECT ASSUMPTIONS				
General Information		Financing		
Project Name:	ABC Canada Company Ltd. Somewhere, Canada	Construction Period Interest Rate:	8.00%	
Configuration:	501D5A-NUG Sells to End User	Long-Term Loan	(Mortgage Style)	
Construction Period, Months	30	Debt, %	80%	
1st Operating Year	2000	Term, yr	15	
NPV Reference Year	1997	Interest rate	6.50%	
Capital Cost (\$000)		Required Reserve Fund (\$000)	4,672	
		Reserve Fund, % Available Income	25%	
		Reserve Fund Interest Rate	4.0%	
		Federal Income Tax	25.00%	
		State/Province Income Tax	12.00%	
		Discount Rate	12.0%	
		Financing Costs, % Capital	2.00%	
		Revenues	1997	Esc %/yr
		Capacity Payment (\$/kW·yr)	363.60	3.00%
		Energy Payment (¢/kWh)	0.00	4.00%
Turnkey Construction Cost	85,000	Fixed O&M Payment (\$/kW·yr)	0.00	2.50%
Owner's Contingency	4,300	Variable O&M Payment (¢/kWh)	0.00	2.50%
Utility Interconnections	1,000	Steam Price (\$/1000 lb)	\$3.25	2.50%
Other	0	Fuel Costs	1997	Esc %/yr
Pre-C.O. Operating Costs	2,000	Commodity (\$/MMBtu)	2.30	2.50%
Permitting	1,000	Transportation (\$/MMBtu)	1.00	2.50%
Working Capital	3,630	Annual Operating Costs	1997	Esc%/yr
Development Fee	0	Fixed O&M (\$000)	5,000	3.00%
Construction Interest	9,770	Variable O&M (¢/kWh)	0.19	2.50%
Financing Costs	3,134	Rent/Misc. (\$000)	500	2.00%
Total Capital Costs	109,834	Property Taxes (\$000)	500	2.00%
Depreciation	% of Total	Insurance (\$000)	2,000	2.00%
Personal Property Combined Cycle	34.0%	Filename	H:\USERS\MIKEC\PEO\501D5A.WK4	
Personal Property Simple Cycle	0.0%			
Real Property	0.0%			
Financing, Legal, Closing, Dev. Costs	0.0%			
Other (e.g., Class 34)	64.0%			
Non-depreciable	2.0%			
Performance				
Capacity—Average, MW	157.2			
Capacity—Contract, MW	157.2			
Heat Rate—HHV, Btu/kWh	7,387			
Capacity Factor	92.0%			
Capacity Degradation	3.0%			
Heat Rate Degradation	2.0%			
Steam Sold to Host (1000 \$/h)	200.0			

PROJECT RESULTS		
1997 Net Present Value @ 12% Discount Rate (\$000)		
25 Years	\$77,237	
20 Years	\$67,564	
15 Years	\$53,042	
10 Years	\$38,924	
Internal Rate of Return	Cash/Cash	Equity
25 Years	23.7%	60.2%
20 Years	23.3%	60.2%
15 Years	22.2%	60.1%
10 Years	18.7%	59.5%
Debt Coverage Ratios		
Average	3.13	
Minimum	2.38	
Performance		
Annual Energy Generation, MWh	1,228,899	
Annual Fuel Required, MMBtu	9,259	
Average Capacity—Degraded, MW	152.5	
Heat Rate—Degraded, Btu/kWh	7,535	
Annual Steam Sold, Mlb	1,611,840	
Financing		
Amount Borrowed (\$000)	\$87,867	
Annual Debt Service (\$000)	\$9,345	
Average Power Escalation	3.00%	
Capital cost/Installed kW (\$/kW)	\$699	
Levelized Power Price @ 12% Discount Rate		
10 Years	5.500 ¢/kWh	
15 Years	5.753 ¢/kWh	
20 Years	5.960 ¢/kWh	
25 Years	6.124 ¢/kWh	

INCOME STATEMENT											
	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024	Totals
Capacity Payments		60,584	62,402	64,274	66,202	68,188	79,049	91,639	106,235	123,155	2,208,858
Energy Payments		0	0	0	0	0	0	0	0	0	0
Fixed O&M Payments		0	0	0	0	0	0	0	0	0	0
Variable O&M		0	0	0	0	0	0	0	0	0	0
Total Electric Revenue		60,584	62,402	64,274	66,202	68,188	79,049	91,639	106,235	123,155	2,208,858
Total Steam Revenue		5,641	5,782	5,927	6,075	6,227	7,045	7,971	9,018	10,204	0
Total Revenues		66,226	68,184	70,201	72,277	74,415	86,094	99,610	115,253	133,359	2,401,551
Total Revenues (cts/kWh)		5.39	5.55	5.71	5.88	6.06	7.01	8.11	9.38	10.85	
OPERATING EXPENSES	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024	Totals
Fuel Commodity Charge		22,934	23,508	24,095	24,698	25,315	28,642	32,405	36,664	41,482	783,382
Fuel Transportation Charge		9,971	10,221	10,476	10,738	11,007	12,453	14,089	15,941	18,036	340,601
Fixed Operations and Maintenance		5,464	5,628	5,796	5,970	6,149	7,129	8,264	9,581	11,106	199,200
Variable Operations and Maintenance		2,514	2,577	2,642	2,708	2,775	3,140	3,553	4,020	4,548	85,888
Miscellaneous fees		531	541	552	563	574	634	700	773	853	16,995
Property Taxes		531	541	552	563	574	634	700	773	853	16,995
Insurance		2,122	2,165	2,208	2,252	2,297	2,536	2,800	3,092	3,414	67,982
Total Expenses		44,067	45,180	46,322	47,492	48,693	55,168	62,513	70,843	80,292	1,511,042
Total Expenses (cts/kWh)		3.59	3.68	3.77	3.86	3.96	4.49	5.09	5.76	6.53	
OPERATING INCOME	(109,834)	22,158	23,004	23,879	24,785	25,722	30,926	37,098	44,411	53,067	780,674
		Cash on Cash I									
		10 Yr.	15 Yr.	20 Yr.	25 Yr.						
		18.7%	22.2%	23.3%	23.7%						

PROTECT CASH FLOWS (\$000)											
	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024	Totals
OPERATING INCOME		22,158	23,004	23,879	24,785	25,722	30,926	37,098	44,411	53,067	890,508
+ Reserve Found Interest		64	158	187	187	187	187	93	0	0	2,558
- Depreciation		18,974	37,843	20,067	2,307	2,134	1,666	1,666	1,666	0	107,637
- Interest Payment		5,711	5,475	5,224	4,956	4,670	2,941	570	0	0	52,307
INCOME BEFORE TAXES		(2,463)	(20,156)	(1,225)	17,709	19,105	26,506	34,955	42,744	53,067	733,123
- Principal Payment		3,634	3,870	4,121	4,389	4,674	6,404	8,775	0	0	87,867
+ Depreciation		18,974	37,843	20,067	2,307	2,134	1,666	1,666	1,666	0	107,637
- Equity Investment	21,967	0	0	0	0	0	0	0	0	0	21,967
- Reserve Fund		3,219	1,453	0	0	0	0	(4,672)	0	0	0
RE-TAX CASH FLOW	(21,967)	9,658	12,363	14,721	15,627	16,564	21,768	32,519	44,411	53,067	730,926
- Tax Payment @ 34.00%		(837)	(6,853)	(416)	6,021	6,496	9,012	11,885	14,533	18,043	249,262
AFTER TAX CASH FLOW	(21,967)	10,495	19,217	15,137	9,606	10,069	12,756	20,634	29,877	35,024	481,664
			10 Yr.	15 Yr.	20 Yr.	25 Yr.					
	EQUITY IRR		59.5%	60.1%	60.2%	60.2%					
	1997 NPV @ 12%		38,924	53,042	67,564	77,237					
	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024	
Cash Available to Service Debt		22,222	23,162	24,066	24,972	25,909	31,112	37,191			
Total Debt Service		9,345	9,345	9,345	9,345	9,345	9,345	9,345			
COVERAGE RATIOS											
SUMMARY	Avg.-25 Yr.	2000	2001	2002	2003	2004	2009	2014	2019	2024	
Reserve Fund		3,219	4,672	4,672	4,672	4,672	4,672	0	0	0	
Total Revenues	81,864	66,226	68,184	70,201	72,277	74,415	86,094	99,610	115,253	133,359	
Total Costs (Including Debt Service)	62,133	53,412	54,525	55,667	56,837	58,037	64,513	71,857	70,843	80,292	
Total Net Revenues	19,731	12,813	13,659	14,534	15,440	16,378	21,581	27,753	44,411	53,067	
Net Revenues/Total Revenues	23.77%	19.35%	20.03%	20.70%	21.36%	22.01%	25.07%	27.86%	38.53%	39.79%	

SCHEDULES												
FUEL COST SCHEDULE	1997	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024	
Fuel Commodity Cost, \$/MMBtu	2.30	2.42	2.48	2.54	2.60	2.67	2.73	3.09	3.50	3.96	4.48	
Fuel Transportation Cost, \$/MMBtu	1.00	1.05	1.08	1.10	1.13	1.16	1.19	1.34	1.52	1.72	1.95	
STEAM PRICE SCHEDULE	1997	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024	
Steam Price, \$/1000 lb	3.25	3.41	3.50	3.59	3.68	3.77	3.86	4.37	4.95	5.60	6.33	
POWER PRICE SCHEDULE	1997		2000	2001	2002	2003	2004	2009	2014	2019	2024	
Capacity Price \$/kW·yr	363.60		397.32	409.24	421.51	434.16	447.18	518.41	600.98	696.70	807.66	
Capacity Price (¢/kWh)	4.51		4.93	5.08	5.23	5.39	5.55	6.43	7.46	8.64	10.02	
Energy Price (¢/kWh)	0.00		0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
Variable O&M Price (¢/kWh)	0.00		0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
Fixed O&M Price (\$/kW)	0.00		0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
Fixed O&M Price (¢/kWh)	0.00		0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
Total Power Price (¢/kWh)	4.51		4.93	5.08	5.23	5.39	5.55	6.43	7.46	8.64	10.02	
Power Price Escalation, %				3.0%	3.0%	3.0%	3.0%	3.0%	3.0%	3.0%	3.0%	
DEPRECIATION SCHEDULE			2000	2001	2002	2003	2004	2009	2014	2019	2024	
Pers Prop (CC)—(20 Yr 150% DB)	34.0%		3.75%	7.22%	6.68%	6.18%	5.71%	4.46%	4.46%	4.46%	0.00%	
Pers Prop (SC)—(15 Yr 150% DB)	0.0%		5.00%	9.50%	8.55%	7.70%	6.93%	5.90%	5.90%	0.00%	0.00%	
Real Prop (31.5 Yr SL)	0.0%		3.17%	3.17%	3.17%	3.17%	3.17%	3.17%	3.17%	3.17%	3.17%	
Financing, Legal, Closing, etc.	0.0%		6.67%	6.67%	6.67%	6.67%	6.67%	6.67%	6.67%	0.00%	0.00%	
Other (e.g., Class 34)	64.0%		25.00%	50.00%	25.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	
Non-depreciable	2.0%											

DEPRECIATION AMOUNTS (\$000)		2000	2001	2002	2003	2004	2009	2014	2019	2024
Personal Property—CC	37,344	1,400	2,696	2,494	2,307	2,134	1,666	1,666	1,666	0
Personal Property—SC	0	0	0	0	0	0	0	0	0	0
Real Property	0	0	0	0	0	0	0	0	0	0
Financing, Legal, Closing, etc.	0	0	0	0	0	0	0	0	0	0
Other (e.g., Class 34)	70,294	17,573	35,147	17,573	0	0	0	0	0	0
Non-depreciable	2,197									
Total	109,834	18,974	37,843	20,067	2,307	2,134	1,666	1,666	1,666	0
LONG-TERM DEBT SCHEDULE		2000	2001	2002	2003	2004	2009	2014	2019	2024
Principal Balance		87,867	84,234	80,364	76,243	71,853	45,239	8,775	0	0
Interest Expense		5,711	5,475	5,224	4,956	4,670	2,941	570	0	0
Principal Repayment		3,634	3,870	4,121	4,389	4,674	6,404	8,775	0	0
Total Debt Service		9,345	9,345	9,345	9,345	9,345	9,345	9,345	0	0

PROJECT ASSUMPTIONS				
General Information		Financing		
Project Name:	ABC Canada Company Ltd. Somewhere, Canada	Construction Period Interest Rate	8.00%	
Configuration:	501F-NUG Sells to End User	Long-Term Loan	(Mortgage Style)	
Construction Period, Months	30	Debt, %	80%	
1st Operating Year	2000	Term, yr	15	
NPV Reference Year	1997	Interest Rate	6.50%	
Capital Cost (\$000)		Required Reserve Fund (\$000)	5,483	
Turnkey Construction Cost	100,000	Reserve Fund, % Available Income	25%	
Owner's Contingency	5,050	Reserve Fund Interest Rate	4.0%	
Utility Interconnections	1,000	Federal Income Tax	25.00%	
Other	0	State/Province Income Tax	12.00%	
Pre-C.O. Operating Costs	2,000	Discount Rate	12.0%	
Permitting	1,000	Financing Costs, % Capital	2.00%	
Working Capital	4,870	Revenues	1997	Esc %/yr
Development Fee	0	Capacity Payment (\$/kW · yr)	440.67	3.00%
Construction Interest	11,450	Energy Payment (¢/kWh)	0.00	4.00%
Financing Costs	3,507	Fixed O&M Payment (\$/kW · yr)	0.00	2.50%
Total Capital Costs	128,877	Variable O&M Payment (¢/kWh)	0.00	2.50%
Depreciation	% of Total	Steam Price (\$/1000 lb)	\$3.25	2.50%
Personal Property Combined Cycle	34.0%	Fuel Costs	1997	Esc %/yr
Personal Property Simple Cycle	0.0%	Commodity (\$/MMBtu)	2.30	2.50%
Real Property	0.0%	Transportation (\$/MMBtu)	1.00	2.50%
Financing, Legal, Closing, Dev. Costs	0.0%	Annual Operating Costs	1997	Esc %/yr
Other (e.g., Class 34)	64.0%	Fixed O&M (\$000)	6,000	3.00%
Non-depreciable	2.0%	Variable O&M (¢/kWh)	0.21	2.50%
Performance		Rent/Misc. (\$000)	500	2.00%
Capacity—Average, MW	242.4	Property Taxes (\$000)	500	2.00%
Capacity—Contract, MW	200.0	Insurance (\$000)	2,500	2.00%
Heat Rate—HHV, Btu/kWh	6,466	Filename	H:\USERS\MIKEC\PEO\501F.WK4	
Capacity Factor	92.0%			
Capacity Degradation	3.0%			
Heat Rate Degradation	2.0%			
Steam Sold to Host (1000 #/h)	200.0			

PROJECT RESULTS		
1997 Net Present Value @ 12% Discount Rate (\$000)		
25 Years	\$164,296	
20 Years	\$147,364	
15 Years	\$121,940	
10 Years	\$92,789	
Internal Rate of Return	Cash/Cash	Equity
25 Years	34.7%	96.0%
20 Years	34.5%	96.0%
15 Years	34.1%	96.0%
10 Years	31.9%	95.9%
Debt Coverage Ratios		
Average	4.76	
Minimum	3.67	
Performance		
Annual Energy Generation, MWh	1,894,944	
Annual Fuel Required, MMBtu	12,498	
Average Capacity—Degraded, MW	235.1	
Heat Rate—Degraded, Btu/kWh	6,595	
Annual Steam Sold, Mlb	1,611,840	
Financing		
Amount Borrowed (\$000)	\$103,102	
Annual Debt Service (\$000)	\$10,965	
Average Power Escalation	3.00%	
Capital cost/installed kW (\$/kW):	\$532	
Levelized Power Price @ 12% Discount Rate		
10 Years	5,500 ¢/kWh	
15 Years	5,753 ¢/kWh	
20 Years	5.960 ¢/kWh	
25 Years	6.124 ¢/kWh	

PROTECT CASH FLOWS (\$000)												
	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024	Totals	
OPERATING INCOME		40,089	41,544	43,049	44,605	46,214	55,121	65,644	78,066	92,716	1,577,154	
+ Reserve Fund Interest		110	219	219	219	219	219	110	0	0	3,070	
- Depreciation		22,264	44,404	23,546	2,706	2,503	1,955	1,955	1,955	0	126,300	
- Interest Payment		6,702	6,424	6,129	5,815	5,480	3,450	669	0	0	61,376	
- INCOME BEFORE TAXES		11,233	(9,065)	13,592	36,303	38,450	49,935	63,130	76,111	92,716	1,392,549	
- Principal Payment		4,264	4,541	4,836	5,150	5,485	7,515	10,296	0	0	103,102	
+ Depreciation		22,264	44,404	23,546	2,706	2,503	1,955	1,955	1,955	0	126,300	
- Equity Investment	25,775	0	0	0	0	0	0	0	0	0	25,775	
Reserve Fund		5,483	0	0	0	0	0	(5,483)	0	0	0	
PRE-TAX CASH FLOW	(25,775)	23,751	30,798	32,303	33,859	35,469	44,376	60,271	78,066	92,716	1,389,972	
Tax Payment @ 34.00%		3,819	(3,082)	4,621	12,343	13,073	16,978	21,464	25,878	31,524	473,467	
AFTER TAX CASH FLOW	(25,775)	19,931	33,880	27,681	21,516	22,396	27,398	38,807	52,188	61,193	916,505	
			10 Yr.	15 Yr.	20 Yr.	25 Yr.						
	EQUITY IRR		95.9%	96.0%	96.0%	96.0%						
	1997 NPV @ 12%		92,789	121,940	147,364	164,296						
Cash Available to service Debt	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024		
		40,198	41,763	43,268	44,824	46,434	55,341	65,754				
Total Debt Service		10,965	10,965	10,965	10,965	10,965	10,965	10,965				
COVERAGE RATIOS												
SUMMARY	Avg.-25 Yr.	2000	2001	2002	2003	2004	2009	2014	2019	2024		
Reserve Fund		5,483	5,483	5,483	5,483	5,483	5,483	0	0	0		
Total Revenues	122,574	99,058	102,002	105,033	108,155	111,369	128,933	149,273	172,826	200,101		
Total Costs (Including Debt Service)	81,595	69,935	71,423	72,950	74,515	76,120	84,777	94,594	94,760	107,385		
Total Net Revenues	40,980	29,124	30,579	32,084	33,640	35,249	44,156	54,679	78,066	92,716		
Net Revenues/Total Revenues	33.15%	29.40%	29.98%	30.55%	31.10%	31.65%	34.25%	36.63%	45.17%	46.33%		

SCHEDULES													
FUEL COST SCHEDULE	1997	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024		
Fuel Commodity Cost, \$/MMBtu	2.30	2.42	2.48	2.54	2.60	2.67	2.73	3.09	3.50	3.96	4.48		
Fuel Transportation Cost, \$/MMBtu	1.00	1.05	1.08	1.10	1.13	1.16	1.19	1.34	1.52	1.72	1.95		
STEAM PRICE SCHEDULE	1997	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024		
Steam Price, \$/1000 lb	3.25	3.41	3.50	3.59	3.68	3.77	3.86	4.37	4.95	5.60	6.33		
POWER PRICE SCHEDULE	1997		2000	2001	2002	2003	2004	2009	2014	2019	2024		
Capacity Price \$/kW·yr	440.67		481.53	495.98	510.86	526.18	541.97	628.29	728.36	844.37	978.86		
Capacity Price (¢/kWh)	5.47		5.97	6.15	6.34	6.53	6.72	7.80	9.04	10.48	12.15		
Energy Price (¢/kWh)	0.00		0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00		
Variable O&M Price (¢/kWh)	0.00		0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00		
Fixed O&M Price (\$/kW)	0.00		0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00		
Fixed O&M Price (¢/kWh)	0.00		0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00		
Total Power Price (¢/kWh)	5.47		5.97	6.15	6.34	6.53	6.72	7.80	9.04	10.48	12.15		
Power Price Escalation, %				3.0%	3.0%	3.0%	3.0%	3.0%	3.0%	3.0%	3.0%		
DEPRECIATION SCHEDULE			2000	2001	2002	2003	2004	2009	2014	2019	2024		
Pers Prop (CC)—(20 Yr 150% DB)	34.0%		3.75%	7.22%	6.68%	6.18%	5.17%	4.46%	4.46%	4.46%	0.00%		
Pers Prop (SC)—(15 Yr 150% DB)	0.0%		5.00%	9.50%	8.55%	7.70%	6.93%	5.90%	5.90%	0.00%	0.00%		
Real Prop (31.5 Yr SL)	0.0%		3.17%	3.17%	3.17%	3.17%	3.17%	3.17%	3.17%	3.17%	3.17%		
Financing, Legal, Closing, etc.	0.0%		6.67%	6.67%	6.67%	6.67%	6.67%	6.67%	6.67%	0.00%	0.00%		
Other (e.g., Class 34)	64.0%		25.00%	50.00%	25.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%		
Non-depreciable	2.0%												

(Continued)

[illegible]

INCOME STATEMENT (\$000)											
	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024	Totals
Capacity Payments		93,417	96,220	99,106	102,080	105,142	121,888	141,302	163,808	18,898	3,405,923
Energy Payments		0	0	0	0	0	0	0	0	0	0
Fixed O&M Payments		0	0	0	0	0	0	0	0	0	0
Variable O&M		0	0	0	0	0	0	0	0	0	0
Total Electric Revenue		93,417	96,220	99,106	102,080	105,142	121,888	141,302	163,808	18,898	3,405,923
Total Steam Revenue		5,641	5,782	5,927	6,075	6,227	7,045	7,971	9,018	10,204	0
Total Revenues		99,058	102,002	105,003	108,155	111,369	128,933	149,273	172,826	200,101	3,598,616
Total Revenues (cts/kWh)		5.23	5.38	5.54	5.71	5.88	6.80	7.88	9.12	10.56	
OPERATING EXPENSES	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024	Totals
Fuel Commodity Charge		30,955	31,729	32,522	33,335	34,169	38,659	43,739	49,486	55,989	1,057,355
Fuel Transportation Charge		13,459	13,795	14,140	14,494	14,856	16,808	19,017	21,516	24,343	459,720
Fixed Operations and Maintenance		6,556	6,753	6,956	7,164	7,379	8,555	9,917	11,497	13,328	239,040
Variable Operations and Maintenance		4,285	4,392	4,502	4,615	4,730	5,352	6,055	6,851	7,751	146,378
Miscellaneous Fees		531	541	552	563	574	634	700	773	853	16,995
Property Taxes		531	541	552	563	574	634	700	773	853	16,995
Insurance		2,653	2,706	2,760	2,815	2,872	3,171	3,501	3,865	4,267	84,977
Total Expenses		58,970	60,458	61,984	63,550	65,154	73,812	83,629	94,760	107,385	2,021,461
Total Expenses (cts/kWh)		3.11	3.19	3.27	3.35	3.44	3.90	4.41	5.00	5.67	
OPERATING INCOME	1999	2000	2001	2002	2003	2004	2009	2014	2019	2024	Totals
(128,877)		40,089	41,544	43,049	44,605	46,214	55,121	65,644	78,066	92,716	1,448,277
Cash on Cash I											
	10 Yr.	15 Yr.	20 Yr.	25 Yr.							
	31.9%	34.1%	34.5%	34.7%							

This page intentionally left blank

CHAPTER 12

Wind Power Turbine Generators—Brushless Double-Feed Generators

12.1 Introduction

Double-feed generators are attractive for use in wind turbine applications. These machines have enabled power generation over a wide range of rotor speeds. Their capacity varies from 50 kW to more than 3 MW. These machines provide control of output frequency and voltage when operated in the islanding mode. They have windings in the rotor. Their generation voltage is usually around 3–4 kV. A transformer is installed at the output of these machines. It steps up the voltage normally to around 30 kV. The most advance design of these generators is known as Brushless Double-Fed Machine (BDFM). This design has high reliability due to the absence of brush-gear. The BDFM can be used with or without a gear box. However, most wind turbine applications use a gear box. A four- or six-pole machine is used in applications having a gear box. The stator of the machine is connected to the grid in most applications. This provides constant-voltage, and constant-frequency operation. Some designs supply the rotor of the machine from a converter through slip rings with variable voltage, and variable frequency power. These machines have a high failure rates due to carbon dust. They also require frequent maintenance. The converter rating is only a fraction of the total generator output. The voltage is varied to the rotor to control the flow of reactive power from the generator.

Various control schemes have been developed to enhance the response of the system to changing rotor speed. The BDFM eliminates the need for brush-gear. The stator has two windings. These windings have different pole numbers. There is no direct coupling between these windings. The rotor is designed to couple both fields. One stator winding is connected to the fixed frequency in the grid. A converter supplies the second winding with variable voltage and variable frequency power. This machine can operate in the synchronous mode (e.g., a current is applied to the rotor winding to produce a rotor magnetic field). Its shaft speed is related to the two excitation frequencies. The largest BDFM is around 100 kW. It is a 12/8-pole machine. The BDFM has replaced the doubly fed induction generators in many applications.

12.2 Basic System Configuration

A BDFM can operate in several modes. However, the synchronous, or double-fed, mode is used for controlled variable speed operation. Figure 12.1 illustrates this arrangement. The power winding is connected to the grid. The second winding is the control winding. This winding is supplied with a variable voltage at variable frequency from a converter. This converter is powered from the grid. Figure 12.2 illustrates the stator and rotor quantities. The shaft angular velocity is given by:

$$\omega_r = \frac{\omega_1 + \omega_2}{p_1 + p_2}$$

- where ω_1 = angular frequency of the supply to the power winding
- ω_2 = angular frequency of the supply to the control winding
- p_1 = pole pairs of the power winding
- p_2 = pole pairs of the control winding

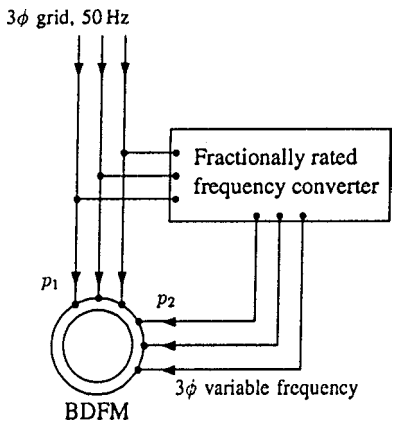


FIGURE 12.1 BDFM system configurations. (Source: Reprinted with permission from McMahon R. A., Wang X., and Abdi-Jalebi E., *The BDFM as a Generator in Wind Turbines*, Engineering Department, Cambridge University, Trumpington Street, Cambridge, CB2 1PZ, UK, 2007.)

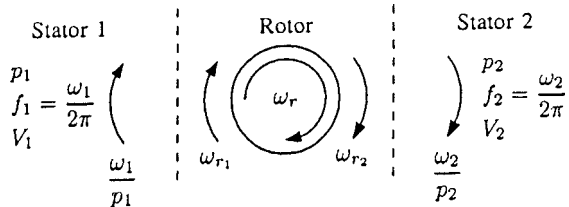


FIGURE 12.2 BDFM synchronous mode of operation. (Source: Reprinted with permission from McMahon R. A., Wang X., and Abdi-Jalebi E., *The BDFM as a Generator in Wind Turbines*, Engineering Department, Cambridge University, Trumpington Street, Cambridge, CB2 1PZ, UK, 2007.)

The slips of the two windings are defined as:

$$S_1 \triangleq \frac{\omega_1 - p_1 \omega_r}{\omega_1}$$

$$S_2 \triangleq \frac{\omega_2 - p_2 \omega_r}{\omega_2}$$

where ω_n = the natural speed. This is the synchronous speed when the control winding is fed with dc power. The ω_n is given by:

$$\omega_n = \frac{\omega_1}{p_1 + p_2}$$

12.3 Equivalent Circuit for the Brushless Double-Fed Machine

Figure 12.3 illustrates the per-phase equivalent circuit of a BDFM. This circuit enhances the analysis of the performance of the BDFM as a generator. The quantities shown are referred to the power winding. The core losses have been neglected. This circuit is valid for all modes of operation including the synchronous mode. A standard induction machine has most of the parameters shown in the equivalent circuit. However, the referred rotor inductance, L'_r , is relatively larger than that of a standard squirrel cage induction machine. There are several design constraints imposed on the rotor of a BDFM. This results in a larger level of harmonics in its output. Some BDFM machines have complex rotors. This includes the nested-loop design. The presence of multiple sets of independent rotor circuits in this design indicates that all the equivalent circuit parameters associated with the rotor change with rotor frequency. However, these changes are minimal.

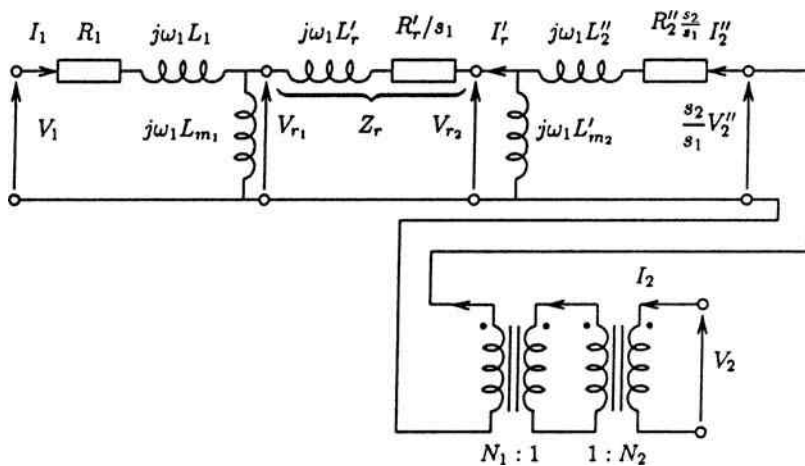


FIGURE 12.3 BDFM referred per-phase equivalent circuit. (Source: Reprinted with permission from McMahon R. A., Wang X., and Abdi-Jalebi E., *The BDFM as a Generator in Wind Turbines*, Engineering Department, Cambridge University, Trumpington Street, Cambridge, CB2 1PZ, UK, 2007.)

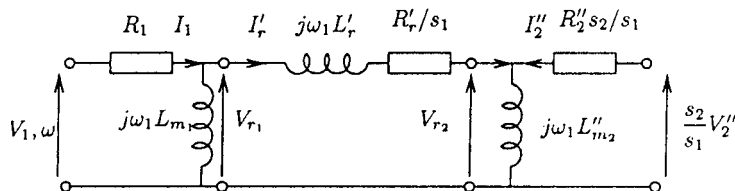


FIGURE 12.4 Alternative referred per-phase equivalent circuit. (Source: Reprinted with permission from McMahon R. A., Wang X., and Abdi-Jalebi E., *The BDFM as a Generator in Wind Turbines*, Engineering Department, Cambridge University, Trumpington Street, Cambridge, CB2 1PZ, UK, 2007.)

Figure 12.4 illustrates an alternative form of referred per-phase equivalent circuit. Table 12.1 defines the parameters. Figure 12.5a illustrates the simplified equivalent circuit for an ideal BDFM. This circuit was obtained by noting that the rotor reactance will dominate the overall rotor impedance. However, in an ideal BDFM, the rotor impedance will be zero. This leads to the model shown in Fig. 12.5b. The following is the equation for the synchronous component of the BDFM torque, T :

$$T = \frac{3|V_1 V_2'' s_2|}{\omega_n |\omega_1 L_r' s_1|} \sin \delta$$

where, δ is the load angle.

The following is the relationship that relates the power in the control winding, P_2 , to that in the power winding, P_1 :

$$P_2 = P_1 \frac{\omega_r}{\omega_n}$$

Parameters	Power Winding	Control Winding	Rotor
Resistance	R_1	R_2''	R_r'
Inductance	L_1	L_2''	L_r'
Magnetizing inductance	L_{m1}	L_{m2}''	—
Turns ratio to rotor	$N_1 : 1$	$N_2 : 1$	—

TABLE 12.1 Definition of Parameters

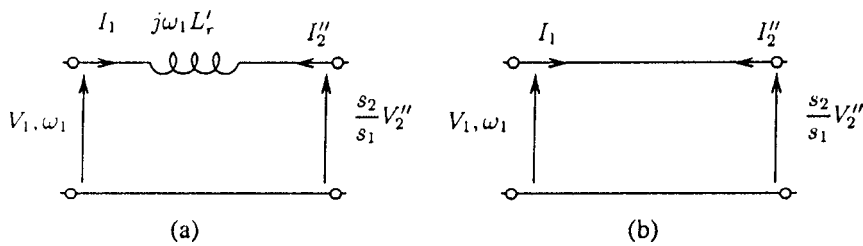


FIGURE 12.5 BDFM core model and ideal model. (Source: Reprinted with permission from McMahon R. A., Wang X., and Abdi-Jalebi E., *The BDFM as a Generator in Wind Turbines*, Engineering Department, Cambridge University, Trumpington Street, Cambridge, CB2 1PZ, UK, 2007.)

12.4 Parameter Extraction

The equivalent circuit parameters can be calculated or measured experimentally. The calculated values of these parameters should be confirmed experimentally. For example, the resistances of the two stator windings are obtained by dc measurements. However, the measured values should be corrected for temperature changes. The magnetizing reactances of the two stator windings can also be measured. This is done by driving the machine externally at the appropriate synchronous speeds.

12.5 Generator Operation

Figure 12.1 illustrates the system configuration of a BDFM. The voltage and frequency of the power winding are constant. The equation for ω_r (shown in Sec. 12.2) provides the relationship between the control winding frequency and the shaft speed. The real power generated is determined by the mechanical input power (i.e., torque $\times \omega_r$ —losses in the machine). The control winding voltage controls the flow of reactive power. However, the flow of reactive power is amplified in the machine. The reactive power amplification factor is the ratio of the reactive power generated in the power winding to the input reactive power from the control winding. This amplification factor drops with increasing speed deviation.

Figure 12.6 illustrates the control of reactive power from the machine. The reactive power flowing in the power winding has been calculated using the core model of Fig. 12.5a and the equivalent circuit of Fig. 12.4 for operation at 750 rpm with a driving torque of 25 Nm. The progressive reduction of reactive power absorbed by the power winding from the grid as the reactive power absorbed by the control winding increases is shown. The control winding voltage increases as the amount of reactive power

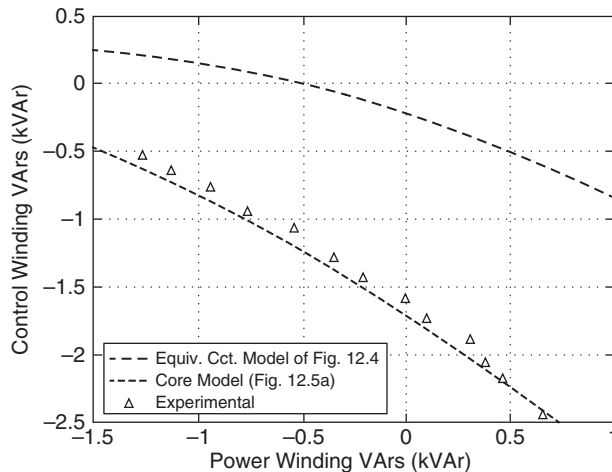


FIGURE 12.6 Variations of the control winding reactive power with the power winding reactive power. The power winding voltage is fixed at $120 V_{rms}$. The negative values of reactive power stand for absorbing reactive power from the grid. The positive values of reactive power stand for delivering reactive power to the grid. (Source: Reprinted with permission from McMahon R. A., Wang X., and Abdi-Jalebi E., *The BDFM as a Generator in Wind Turbines*, Engineering Department, Cambridge University, Trumpington Street, Cambridge, CB2 1PZ, UK, 2007.)

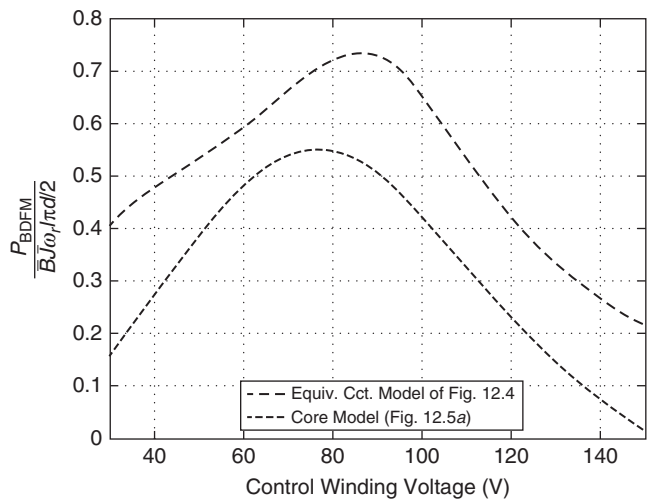


FIGURE 12.7 Dimensionless quantity of the ratio of the total BDFM power output to the product of the BDFM magnetic loading, electric loading, ω_r , and machine volume. The ratio was calculated over a range of control winding voltages. The power winding voltage was fixed at 120 V_{rms}, prime mover torque at 25 Nm, shaft speed at 750 rpm. (Source: Reprinted with permission from McMahon R. A., Wang X., and Abdi-Jalebi E., *The BDFM as a Generator in Wind Turbines*, Engineering Department, Cambridge University, Trumpington Street, Cambridge, CB2 1PZ, UK, 2007.)

absorbed from the grid increases. The effect on the control winding depends on the speed deviation from the natural speed. The amount of reactive power generated on the power winding is modestly affected up to a speed deviation of 25%. However, the amount of reactive power generated on the power winding drops significantly when the speed deviation is more than 75%. Figure 12.7 illustrates the effect of reactive power generation on available power generation capacity. This figure indicates that the power generated by the machine is compromised by both underexcitation and overexcitation of the control winding. Thus, the degree of excitation should be chosen carefully.

12.6 Converter Rating

The rating of the control winding and the inverter supplying it are affected by the variations of the control winding voltage. The converter linking the grid and the control winding is bidirectional. The machine-side inverter will handle the real and reactive power delivered to the control winding. This inverter is sized to match the rating of the control winding. However, the line-side inverter transfers real power only. Thus, it could have a lower rating. Figure 12.8 illustrates the ratings of the two inverters as a function of speed deviation. The operating conditions are 120 V rms on the power winding. The load torque is 25 Nm. The power winding has a unity power factor. This is achieved by adjusting the control winding voltage. Note that this condition involves transfer of reactive power to the power winding to supply the reactive power required to magnetize the machine. The rating of the line-side inverter in this case is equal to the real power transferred to the grid. The following is an alternative operation strategy:

1. Supply the same reactive power to the grid from the line-side inverter.
2. Allow the power winding to draw lagging reactive power.

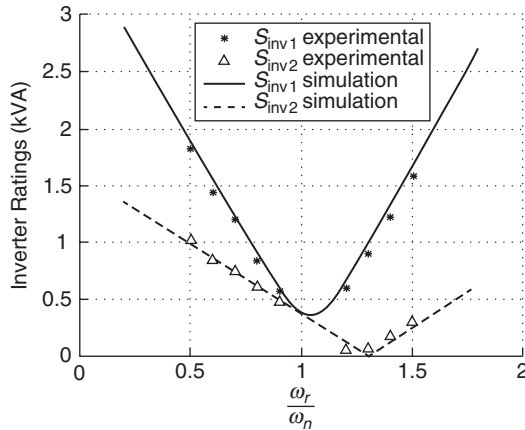


FIGURE 12.8 Inverter ratings. S_{inv1} is the machine-side inverter rating and S_{inv2} is the line-side inverter rating. (Source: Reprinted with permission from McMahon R. A., Wang X., and Abdi-Jalebi E., *The BDFM as a Generator in Wind Turbines*, Engineering Department, Cambridge University, Trumpington Street, Cambridge, CB2 1PZ, UK, 2007.)

This strategy requires the increase in the rating of the line-side inverter. However, the control winding voltage can be reduced. This results in the reduction of both the ratings of the control winding and the machine-side inverter. The converter rating can be minimized at each speed when this strategy is implemented. Figure 12.9 illustrates the minimized rating compared to the rating used in Fig. 12.8 as a function of speed deviation.

The strategy of using both inverters to supply reactive power results in a reduction in the total converter rating. In practice, both inverters can conveniently have the same size. Optimization is implemented to equalize the loadings. The converter rating and

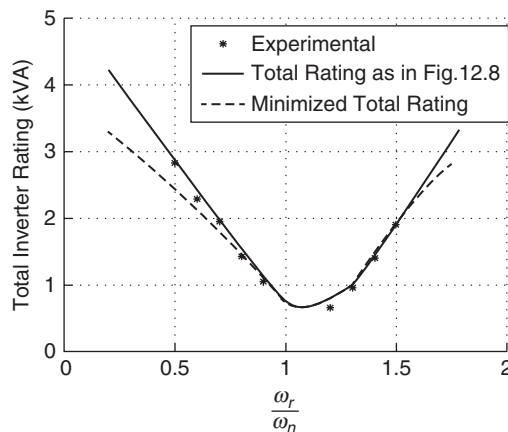


FIGURE 12.9 Comparison of minimized total inverter rating and total inverter rating shown in Fig. 12.8. (Source: Reprinted with permission from McMahon R. A., Wang X., and Abdi-Jalebi E., *The BDFM as a Generator in Wind Turbines*, Engineering Department, Cambridge University, Trumpington Street, Cambridge, CB2 1PZ, UK, 2007.)

machine size can also be optimized by supplying reactive power on the power winding side. This reduces the rating of the control winding. Thus, a smaller machine can be used. This optimization is focused on minimizing the system cost.

12.7 Machine Control

The BDFM is a synchronous system. Its control is complex due to the presence of unstable regions. Several control algorithms have been used for this machine. Further work is required to develop control schemes for this machine. These control schemes will consider the following:

1. Response of the control scheme to grid-load interaction
2. Effectiveness of the control scheme over a wide range of machine designs, especially those having unstable regions

12.8 Conclusions

Recent advancements in the study of the BDFM allowed this machine to be used for power generation over a wide range of rotor speeds. However, additional work is required for the following reasons:

1. Devise a simpler method of determining the balance of electric and magnetic loadings in the machine.
2. Determining the unstable regions of operation.
3. Development of practical control schemes for the machine.

The BDFMs operate at relatively low speed. The highest speed machine has 2/6 pole. This version is the counterpart of an 8-pole wound rotor induction machine. However, the 4/8-pole and the 2/6-pole BDFMs have a natural speed of 500 and 750 rpm, respectively. These machines are ideal for indirect drive wind turbine applications. The main advantage of the BDFMs is the elimination of the brush-gear. This has been the goal of electrical engineers for many years.

12.9 Bibliography

- DOWEC. "Estimation of Turbine Reliability Figures within the DOWEC Project," Dutch Offshore Wind Energy Converter Report Nr. 10048, October 2003.
- McMahon R. A., Wang X., Abdi-Jalebi E., *The BDFM as a Generator in Wind Turbines*, Engineering Department, Cambridge University, Trumpington Street, Cambridge, CB2 1PZ, UK, 2007.
- Runcos F., Carlson R., Oliveira A. M., Kuo-Peng P., Sadowski N., "Performance Analysis of a Brushless Doubly-Fed Cage Induction Generator," Proc. 2nd Nordic Windpower Conf., 2004, 1–2 March.

CHAPTER 13

Gas Laws and Compression Principles

13.1 Introduction

The discussion of thermodynamics presented in this chapter is limited to the gas compression processes that occur in a reciprocating-type compressor. Positive displacement compressors include reciprocating- and rotary-type compressors. A positive displacement compressor is a machine that increases the pressure of a specific initial volume of gas. This pressure increase is accomplished by reducing the volume of the gas. The only way to solve compressor problems is by thoroughly understanding all the compression laws and their applications.

13.2 Symbols

The list of symbols that are used in this chapter is presented in Sec. 13.5.

13.2.1 Compressor Operation

The *compression ratio* of a compressor is defined as the absolute discharge pressure divided by the absolute intake pressure. A very high compression ratio will cause excessive discharge temperature and other problems for the compressor. Multistage compressors should be used in this case. The gas is cooled between the stages for the following reasons:

1. To reduce the temperature and volume of the gas entering the following stage
2. To increase the efficiency of the following compressions stage (the efficiency of a compression stage drops as the inlet gas temperature increases)

The gas is compressed on only one side of the piston in a *single-acting* reciprocating compressor cylinder. The gas is compressed on both sides of the piston in a double-acting reciprocating compressor cylinder. Reciprocating compressors employ automatic spring-loaded valves. These valves open when the differential pressure across them reaches the design value. The inlet valves open when the pressure in the cylinder drops

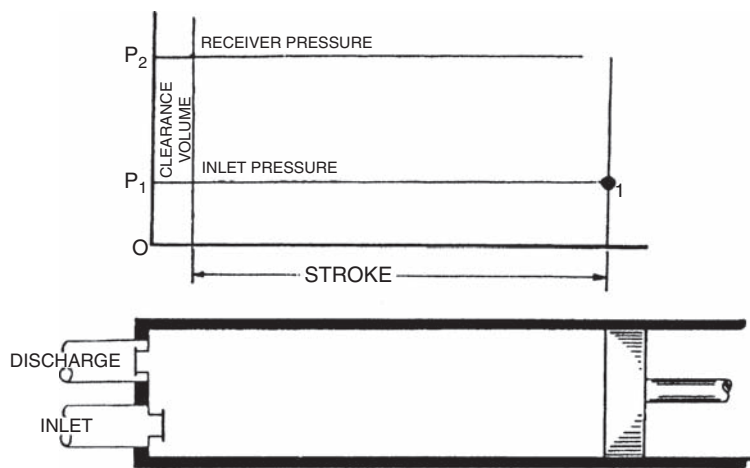


FIGURE 13.1 Basic compressor element with the cylinder full of gas. On the theoretical P-V diagram (indicator card), point 1 is the start of compression. Both valves are closed. (Source: Dresser-Rand Company, Painted Post, N.Y.)

slightly below the intake pressure. The discharge valves open when the pressure in the cylinder becomes slightly above the discharge pressure.

Figure 13.1 illustrates a reciprocating compressor cylinder full of gas. Point 1 on the theoretical pressure versus volume (P-V) diagram indicates the start of the compression process. The inlet and discharge valves are closed at this stage. Figure 13.2 illustrates the compression stroke. The piston has moved to the left. The original volume of the air has

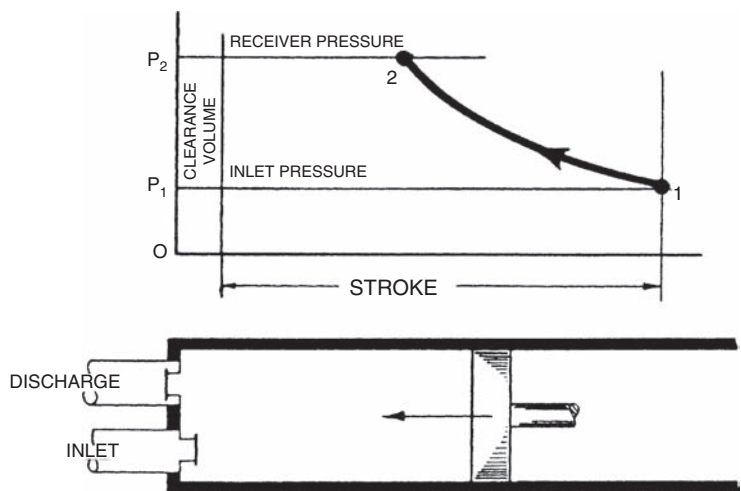


FIGURE 13.2 Compression stroke. The piston has moved to the left, reducing the original volume of gas with an accompanying rise in pressure. Valves remain closed. The P-V diagram shows compression from point 1 to 2 and the pressure inside the cylinder has reached that in the receiver. (Source: Dresser-Rand Company, Painted Post, N.Y.)

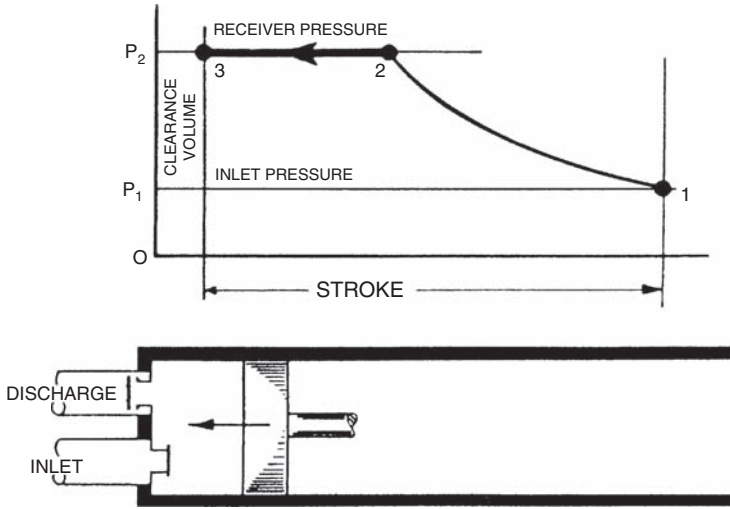


FIGURE 13.3 The piston is shown completing the delivery stroke. The discharge valves opened just beyond point 2. Compressed air is flowing out through the discharge valves to the receiver. (Source: Dresser-Rand Company, Painted Post, N.Y.)

dropped. The pressure of this air has increased. The valves still remain closed at this stage. The P-V diagram of Fig. 13.2 illustrates the compression process from point 1 to 2. The pressure inside the cylinder at point 2 has reached that in the receiver. The discharge valve opens just beyond point 2. Figure 13.3 illustrates the piston completing the delivery stroke from point 2 to 3. The discharge valve closes after the piston reaches point 3. The clearance volume remains filled with air at discharge pressure. Figure 13.4

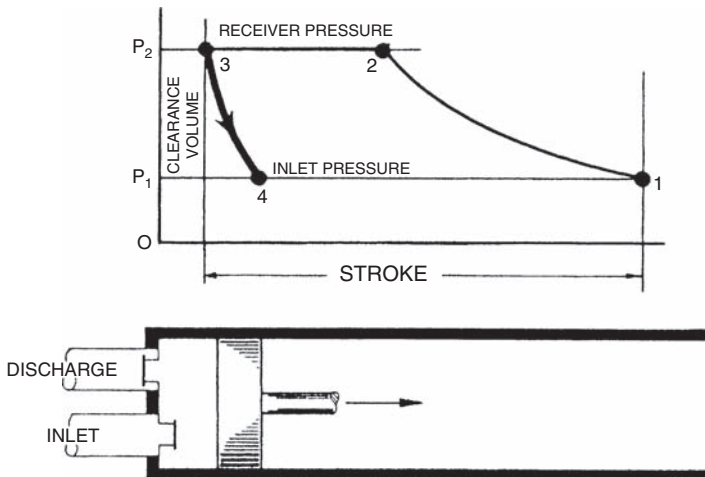


FIGURE 13.4 During the expansion stroke shown, both the inlet and discharge valves remain closed and gas trapped in the clearance space increases in volume, causing a reduction in pressure. (Source: Dresser-Rand Company, Painted Post, N.Y.)

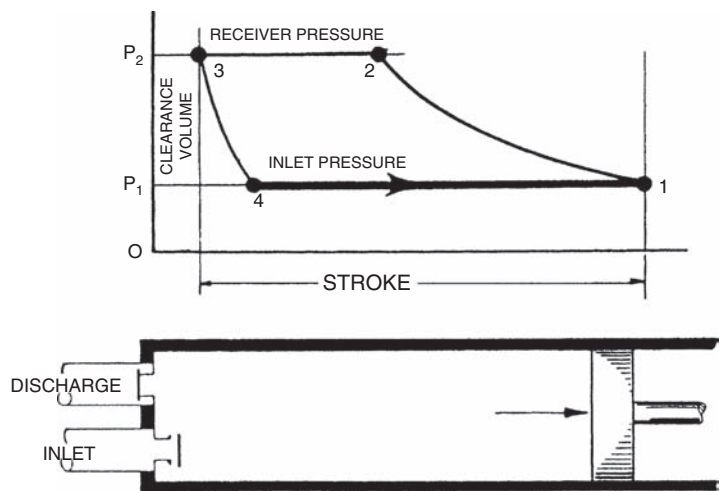


FIGURE 13.5 At point 4, the inlet valves will open and gas will flow into the cylinder until the end of the reverse stroke at point 1. (Source: Dresser-Rand Company, Painted Post, N.Y.)

illustrates the expansion stroke. The inlet and discharge valves remain closed during this process. The volume of the air that was trapped in the clearance space increases during the expansion stroke. This leads to a reduction in pressure inside the cylinder. This pressure continues to drop as the piston moves to the right. The inlet valve will open when the cylinder pressure drops below the inlet pressure at point 4. Air will be admitted into the cylinder until the end of the reverse stroke at point 1. Figure 13.5 illustrates the intake or suction stroke. The inlet valve will close when the process reaches point 1 on the P-V diagram. The next revolution of the crank will repeat the cycle. Figure 13.6

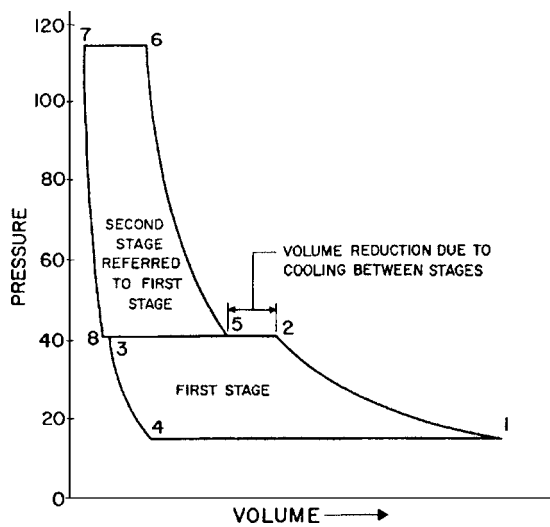


FIGURE 13.6 The P-V diagram for a two-stage compressor. (Source: Dresser-Rand Company, Painted Post, N.Y.)

illustrates the P-V diagram for a two-stage compressor. The second stage of the compressor is smaller than the first. This is due to the following:

1. The partial compressions of the gas in the first stage of the compressor.
2. The drop in the volume of the gas in the intercooler (process 2-5). Points 1 and 5 are the conditions of the air at the start of the first and second stage compression processes, respectively. Points 2 and 6 are the air conditions at the end of the compression processes. Points 3 and 7 are the air conditions at the end of the delivery processes. Points 4 and 8 are the conditions at the end of the expansion processes of the air that was trapped in the clearance space when the piston has reversed direction. Points 1 and 5 are the conditions of the air at the end of the intake strokes. The cycles are repeated starting at points 1 and 5.

13.3 First Law of Thermodynamics

The first law of thermodynamics states that energy can neither be created nor destroyed during a process. However, energy can be converted from one form to another. For example, the velocity of a gas decreases between the inlet and outlet of a diffuser (a component of increasing cross section). Based on the first law of thermodynamics, the pressure of this gas must increase between the inlet and outlet of the diffuser.

13.4 Second Law of Thermodynamics

The second law of thermodynamics can be stated as follows:

1. Heat cannot, of itself, pass from a colder to a hotter body.
2. Heat can be transferred from a body at lower temperature to one at a higher temperature *only* if external work is done.
3. The available energy of the isolated system decreases in all real processes.
4. Heat or energy (or water), of itself, will flow only downhill.

These statements indicate that energy exists at various levels. However, it is *available for use* only if it can move from a higher to a lower level. In thermodynamics, entropy is a measure of the unavailability of energy. It is defined by the following differential equation:

$$ds = dQ/T$$

Note that the entropy increases when a system loses heat. However, it remains constant when there is no gain or loss of heat (as in an adiabatic process).

13.4.1 Ideal or Perfect Gas Laws

The laws of Boyle, Charles, and Amonton apply for an ideal or perfect gas. In reality, there are no perfect gases. However, these laws can be used and corrected by compressibility factors based on experimental data.

Boyle's Law

Boyle's law is known as the *isothermal law*. It states that at a constant temperature, the volume of an ideal gas varies inversely with the pressure. The following is the equation of Boyle's law:

$$V_2/V_1 = P_1/P_2$$

$$P_2V_2 = P_1V_1 = \text{constant}$$

Charles' Law

Charles' law states that at constant pressure, the volume of an ideal gas varies directly with the absolute temperature. The following is the equation for Charles' law:

$$V_2/V_1 = T_2/T_1$$

$$V_2/T_2 = V_1/T_1 = \text{constant}$$

Amonton's Law

Amonton's law states that at constant volume, the pressure of an ideal gas varies directly with the absolute temperature. The following is the equation of Amonton's law:

$$P_2/P_1 = T_2/T_1$$

$$P_2/T_2 = P_1/T_1 = \text{constant}$$

Dalton's Law

Dalton's law applies to mixture of ideal gases. It states that the total pressure of the mixture is equal to the sum of the partial pressures of the constituent gases. The *partial pressure* of a gas is defined as the pressure it exerts if it occupied alone the volume of the mixture at the mixture temperature.

Dalton's law has been proven experimentally to be inaccurate. The total pressure of the mixture is often higher than the sum of the partial pressures. This is especially true at higher pressures. However, the error in Dalton's law is minimal. Thus, for engineering purposes, it is the best law available. The following is the equation of Dalton's law:

$$P = P_a + P_b + P_c + \dots$$

All the pressures mentioned in this law are at the same temperature and volume.

Amagat's Law

Amagat's law states that the volume of a mixture of ideal gases is equal to the sum of the partial volumes of the constituent gases if each gas existed alone at the *total* pressure and temperature of the mixture. The following is the equation of Amagat's law:

$$V = V_a + V_b + V_c + \dots$$

Avogadro's Law

Avogadro's law states that equal volumes of all gases under the same conditions of pressures and temperatures contain the same number of molecules. This is an important law. It is used in many compressor calculations.

Heat and Work

The heat *added* (entering) to a steady-state steady-flow (SSSF) system is given by Q_A . Q_R is the heat *rejected* by (leaving) the system. The net heat added is given by

$$\Delta Q = Q_A - |Q_R|$$

Note that Q_A is a positive term and Q_R is a negative term. The net heat added is also given by

$$\Delta Q = mc_n(T_2 - T_1)$$

where c_n is the specific heat. Subscript 1 and 2 represent the conditions at the inlet and outlet to the SSSF system, respectively. c_n varies with the process occurring between the inlet and the outlet of the SSSF system. Table 13.1 provides the values of c_n for various processes.

ΔW_{sf} is the net steady-flow mechanical work done by the system. It is given by

$$\Delta W_{sf} = W_{by} - |W_{on}|$$

where W_{by} is the work done by the system. W_{on} is the amount of work done on the system. The convention is that the work done by the system is positive. The work done on the system is negative. The steady-flow work is given by

$$\Delta W_{sf} = - \int_1^2 V dP$$

The relationship between pressure P and volume V is given by

$$PV^n = \text{constant}$$

where n is called the *polytropic exponent*. It varies between zero and infinity. Table 13.1 provides the values of c_n for some processes.

Process	c_n	n
Constant pressure	c_p	0
Constant temperatures	∞	1
Adiabatic reversible	0	$k = \frac{c_p}{c_v}$
Constant volume	c_v	∞
Polytropic	$c_v \frac{k-n}{1-n}$	$0 - \infty$

TABLE 13.1 Values of c_n and n for Various Processes

13.4.2 Property Relationships

Table 13.2 provides the property relationships for perfect gases for different processes. A perfect (or ideal) gas is one that obeys the *equation of state* for perfect gases at any state. The following are the perfect gas formulae:

$$PV = mRT$$

$$Pv = RT$$

$$pV = nR_oT$$

where R = specific gas constant. Different gases have different values of R ; for air $R = 53.3 \text{ ft} \cdot \text{lb}_f / (\text{lb}_m \cdot ^\circ\text{R})$, $286.8 \text{ J}/(\text{kg} \cdot \text{K})$

n = number of moles = m/M , where M is the molecular mass of the gas = 28.97 for air

R_o = universal gas constant = RM , the same for all perfect gases
 $= 1545.33 \text{ ft} \cdot \text{lb}_f / (\text{lb} \cdot \text{mol} \cdot ^\circ\text{R}) = 8314.34 \text{ J}/(\text{kg} \cdot \text{mol} \cdot \text{K})$

T = absolute temperature in degrees Rankine or Kelvin

m is the total mass of the gas in kg (lb). The letter W is used instead of m in some formulae.

Imperfect Gases

The molecules of a *non-perfect gas* are close enough to exert forces on each other. The property relationship of a non-perfect gas is given by

$$PV = mZRT$$

where Z is a compressibility factor. It depends on the pressure, temperature, and the gas itself. Figure 13.7 illustrates a generalized compressibility chart. It provides the value of Z for all gases as a function of the *reduced pressure*, P_r , and *reduced temperature*, T_r . P_r and T_r are given by the following equations:

$$P_r = P/P_c$$

$$T_r = T/T_c$$

where P_c and T_c are the pressure and temperature at the critical point for each gas, respectively. Note that $Z = 0.27$ at the critical point of all fluids (i.e., $P_r = 1$ and $T_r = 1$). Table 13.3 provides the critical constants for some fluids.

Example 13.1 Air is stored in a 1-m^3 rigid tank at 10 MPa and 25°C . Determine the following:

1. The mass of the air in the tank
2. The error if the perfect gas law were used

Solution

For air, $P_c = 37.744 \text{ bar}$, $T_c = 309.50 \text{ K}$, and $R = 286.8 \text{ J}/\text{Kg} \cdot \text{K}$; Therefore,

$$P_r = (10 \times 10^6 \text{ Pa}) / (37.74 \times 10^5 \text{ Pa}) = 2.65; T_r = (25 + 273) / 309.50 = 0.96$$

Using Fig. 13.7, $Z = 0.38$

$$m = (PV) / (ZRT) = (10^7 \times 1) / [0.38 \times 286.8 \times (25 + 273)] = 307.9 \text{ kg}$$

Process	P, v, T Relationships	$u_2 - u_1$	$h_2 - h_1$	$s_2 - s_1$	w (nonflow)	w (flow)	Q
Isothermal	$T = \text{constant}$ $P_1/P_2 = v_2/v_1$	0	0	$(R/J) \ln (v_2/v_1)$	$(P_1 v_1/J) \ln (v_2/v_1)$	$(P_1 v_1/J) \ln (v_2/v_1)$	$(P_1 v_1/J) \ln (v_2/v_1)$
Constant pressure	$P = \text{constant}$ $T_2/T_1 = v_2/v_1$	$c_v(T_2 - T_1)$	$c_p(T_2 - T_1)$	$c_p \ln (T_2/T_1)$	$P(v_2 - v_1)/J$	0	$c_p(T_2 - T_1)$
Constant volume	$v = \text{constant}$ $T_2/T_1 = P_2/P_1$	$c_v(T_2 - T_1)$	$c_p(T_2 - T_1)$	$c_v \ln (T_2/T_1)$	0	$v(P_1 - P_2)/J$	$c_v(T_2 - T_1)$
Isentropic (Adiabatic reversible)	$s = \text{constant}$ $P_1 v_1^k = P_2 v_2^k$ $T_2/T_1 = (v_1/v_2)^{k-1}$ $T_2/T_1 = (P_2/P_1)^{(k-1)/k}$	$c_v(T_2 - T_1)$	$c_p(T_2 - T_1)$	0	$\frac{(P_2 v_2 - P_1 v_1)}{J(1-k)}$	$\frac{k(P_2 v_2 - P_1 v_1)}{J(1-k)}$	0
Throttling	$h = \text{constant}$ $T = \text{constant}$ $P_1/P_2 = v_2/v_1$	0	0	$(R/J) \ln (v_2/v_1)$	0	0	0
Polytropic	$P_1 v_1^n = P_2 v_2^n$ $T_2/T_1 = (v_1/v_2)^{n-1}$ $T_2/T_1 = (P_2/P_1)^{(n-1)/n}$	$c_v(T_2 - T_1)$	$c_p(T_2 - T_1)$	$c_v \ln (P_2/P_1) + c_p \ln (v_2/v_1)$	$\frac{(P_2 v_2 - P_1 v_1)}{J(1-n)}$	$\frac{n(P_2 v_2 - P_1 v_1)}{J(1-n)}$	$c_v \left(\frac{k-n}{1-n} \right) (T_2 - T_1)$

TABLE 13.2 Perfect-Gas Relationship (Constant-Specific Heats)

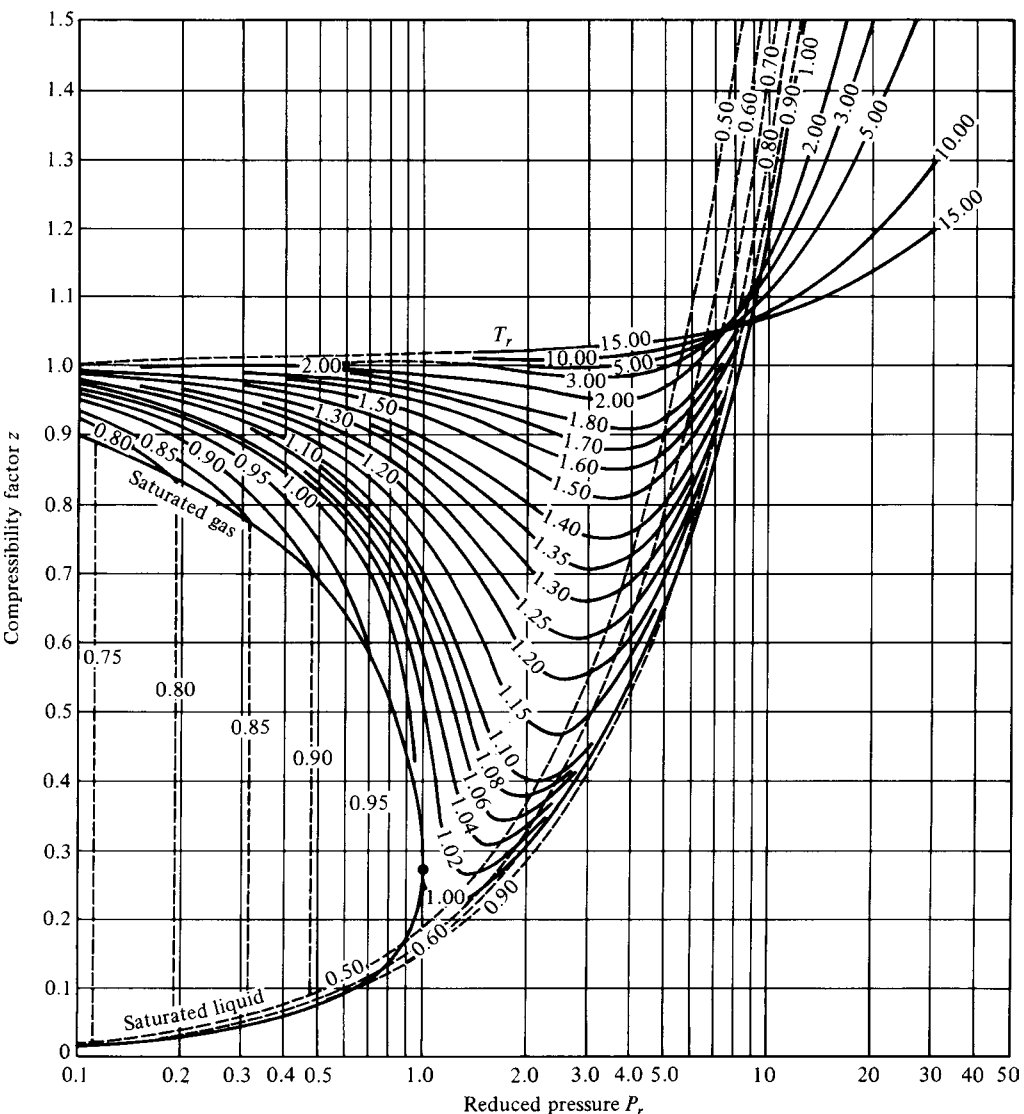


FIGURE 13.7 Generalized compressibility factor chart. (Source: El-Wakil, M. M., *Power Plant Technology*, McGraw-Hill, New York, 1984, with permission from McGraw-Hill.)

Since the mass obtained using the perfect gas law given by

$$m = PV/RT$$
$$m = (10^7 \times 1)/[286.8 \times (298)] = 117 \text{ kg}$$

The error in using the perfect gas law is given by

$$\text{Error} = (117 - 307.9)/307.9 = -0.62 \text{ or } -62\%$$

This error is very large. Thus, the compressibility chart should be used in most applications. However, the error using the perfect gas law becomes negligible for all gases when the pressure and temperature of the gas approach the standard conditions [1 atmosphere (14.7 psia), and 15.56°C (60°F)].

Fluid	M	$R, \text{ ft} \cdot \text{lb}_f / (\text{lb}_m \cdot ^\circ\text{R})$	P_c		T_c	
			psia	bar	$^\circ\text{R}$	K
Air	28.967	53.34	547.43	37.744	557.1	309.50
Ammonia	17.032	90.77	1635.67	112.803	238.34	132.41
Carbon dioxide	44.011	35.12	1071.34	73.884	547.56	304.20
Carbon monoxide	28.011	55.19	507.44	34.995	239.24	132.91
Freon-12	120.925	12.78	596.66	41.148	693.29	385.16
Helium	4.003	386.33	33.22	2.291	9.34	5.19
Hydrogen	2.016	766.53	188.07	12.970	59.83	33.24
Methane	16.043	96.40	67.31	46.418	343.26	190.70
Nitrogen	28.016	55.15	492.91	33.993	227.16	126.20
Octane	114.232	13.54	362.11	24.973	1024.92	569.40
Oxygen	32.000	48.29	736.86	50.817	278.60	154.78
Sulfur dioxide	64.066	24.12	1143.34	78.850	775.26	430.70
Water	18.016	85.80	3206.18	221.112	1165.09	647.27

*Multiply the values of R by 5.343 to convert to $\text{J}/(\text{kg} \cdot \text{K})$.

TABLE 13.3 Constants for Some Fluids*

13.4.3 Vapor Pressure

Liquids evaporate due to an increase in temperature. The molecules of the gas travel with greater velocity. These molecules create a vapor pressure above the liquid. This vapor pressure is the only pressure at which a pure liquid and its vapor can exist in equilibrium at a specified temperature.

The reduction in volume of a closed liquid-vapor system at constant temperature will increase the pressure imperceptibly. This increase in pressure will continue until a part of the vapor is condensed into liquid. The pressure will drop as a result of this condensation until it reaches the original vapor pressure corresponding to the temperature. Conversely, the pressure will be reduced imperceptibly if the volume of this system is increased at constant temperature. The liquid molecules will continue to evaporate until the original vapor pressure is restored. The temperature and vapor pressure of a given gas always vary simultaneously in the same direction.

The *boiling point* of the liquid, and also the *dew point* of the vapor, is the temperature that corresponds to any given vapor pressure. The temperature at any given vapor pressure is given by any of the following terms:

1. Saturation temperature
2. Boiling point
3. Dew point

The use of these terms depends on the context where they appear.

13.4.4 Partial Pressures

The vapor pressure created by a pure liquid will not affect the vapor pressure of a second pure liquid if all the following conditions are met:

1. The two liquids are insoluble.
2. The two liquids are non-reacting.
3. The liquids and/or vapors are mixed within the same system.

Dalton’s law states that the vapor pressure of a liquid is completely unaffected by the existence of other vapor pressures in a system. The vapor pressure of each liquid is called *partial pressure*. The total pressure of the mixture is the sum of all the partial pressures in the system. The principles of partial pressure apply during the compression of any gas. The only exception to this rule is the compression of a pure and dry gas. Water vapor is always mixed with the intake air to a compressor. The compressor must be able to handle both components. The partial pressures are used after the compression to determine the moisture condensation and removal in intercoolers and aftercoolers. The laws of partial pressures are also applied in the following applications:

1. Vacuum pumps
2. Compression of mixtures

Partial pressure problems in compressing gases are normally caused by water vapor. Thus, the discussion in this chapter will be limited to water vapor.

In a mixture, when the dew-point temperature of any component is reached, the volume occupied would become *saturated* by that component. A volume becomes *partially saturated* with water vapor at a certain temperature when the following conditions are met:

1. The vapor is superheated.
2. The dew point is lower than the actual temperature.

The partial pressure of a component can be determined, if the number of moles (see Sec. 13.1.11) of each component is known. Otherwise, the partial pressure is obtained normally by multiplying the vapor pressure of the component at the existing mixture temperature by the relative humidity.

The following terms are incorrect:

1. Saturated gas
2. Partially saturated gas

These terms give the wrong impression. The gas cannot be saturated with vapor. The volume or space occupied becomes saturated with vapor.

The quantity of moisture present in a mixture is represented by the *relative humidity*. This term is given by the following equation:

$$\begin{aligned} \text{RH}(\%) &= \frac{\text{actual partial vapor pressure} \times 100}{\text{saturated vapor pressure at existing mixture temperature}} \\ &= \frac{p_v \times 100}{p_s} \end{aligned}$$

The steam tables provide the saturated water vapor pressure at a given temperature. Thus, the existing partial vapor pressure can be calculated when the relative humidity is known. The *specific humidity* is used in some compressor applications. The specific humidity is defined as the ratio of the weight of water vapor to the weight of *dry air*. It is normally expressed in Kg (lb) of moisture per Kg (lb) of dry air:

$$SH = \frac{W_v}{W_a}$$

The *degree of saturation* defines the relationship between the weight of moisture that exists in a space and the weight that would exist if the space were saturated:

$$\begin{aligned} \text{degree of saturation (\%)} &= \frac{SH \text{ actual} \times 100}{SH \text{ saturated}} \\ &= RH \times \frac{p - p_s}{p - p_v} \end{aligned}$$

p_s and p_v are normally quite small compared to p . Thus the degree of saturation has almost the same value as the relative humidity.

13.4.5 Critical Conditions

All gases will not liquefy with pressure increases when the gas temperature is above its *critical temperature* (Table 13.3). The critical temperature of a gas is determined experimentally. The *critical pressure* is the pressure required to compress and condense a gas at its critical temperature.

13.4.6 Gas Mixtures

Gas mixtures should be considered as equivalent ideal gases. This assumption is not strictly true. However, it is satisfactory for compressor applications.

13.4.7 The Mole

Avogadro's law states that equal volume of all gases under the same pressure and temperature contains equal number of molecules. Thus, the *weight* of these equal volumes will be proportional to their molecular weights. The number of *moles* of a gas is defined as follows:

$$n = m/M$$

where n = the number of moles of the gas
 m = the total mass of the gas
 M = the molecular mass of the gas

The mole is a useful term to use when dealing with gas mixtures. The perfect gas law can be used to determine the volume of 1 mole at any desired condition:

$$PV = nR_oT$$

13.4.8 Volume Percent of Constituents

The mole percent is defined as the ratio of the number of moles of one gas constituent to the total number of moles in the gas mixture. The mole percent is also the percent by volume.

13.4.9 Molecular Weight of a Mixture

The average molecular weight of a gas mixture is obtained by multiplying the molecular weight of each component by its mole fraction (mole %/100) and adding these values as follows:

Gas	Mol % or Vol %	Mol. Wt.	Proportional Mol. Wt
H ₂	48.3	2	0.966
N ₂	23.4	28	6.552
CO ₂	21.9	44	9.636
CO	6.4	28	1.792
	100.0		18.946

Therefore, the average (or pseudo) molecular weight of the gas mixture is 18.946.

13.4.10 Specific Gravity and Partial Pressure

The specific gravity of a gas is defined as the ratio of its density at standard pressure and temperature (SPT) conditions to the density of air at SPT. The SPT conditions are 101 kPa_(absolute) (14.696 psia) and 15.56°C (60°F).

The volume of a mole does not vary with the type of the gas. However, the weight of a mole of any gas at SPT is the same as the molecular weight. Thus, the specific gravity of a gas is the ratio of the molecular weight of the gas to that of air. For example, the specific gravity of the gas mixture discussed in the previous section is

$$\gamma = 18.946/28.97 = 0.654$$

where, γ is the specific gravity of the gas mixture discussed in Sec. 13.1.13.

The partial pressure of a gas in a gas mixture is the fraction of the total pressure of the gas mixture that it exerts. The partial pressure of the gas is the ratio of its number of moles to the total number of moles in the mixture. The partial pressure of the gas is given by the following equation:

$$P_a = PN_a/N \qquad P_b = PN_b/N \qquad P_c = PN_c/N$$

where P_a , P_b , and, P_c are the partial pressure of gases a, b, and c, respectively. P is the total pressure of the mixture. N_a , N_b , and, N_c are the number of moles of gases a, b, and c. N is the total number of moles in the gas mixture.

Thus, if a gas mixture has the following specifications:

- 1. Total number of moles = 20
- 2. Total pressure = 130 kPa_(absolute) or 18.92 psia
- 3. Contain 4 moles of hydrogen

The partial pressure of hydrogen would be 4/20 of 130 kPa_(abs), or 26 kPa_(abs) or 3.78 psia. The volume fraction of the gas can be used instead of its mole fraction in this case as well to determine its partial pressure.

13.4.11 Specific Heats

The specific heats at constant volume and constant pressure are given by the following equations, respectively:

$$C_v \equiv \left(\frac{\partial u}{\partial T} \right)_v$$

$$C_p \equiv \left(\frac{\partial h}{\partial T} \right)_p$$

where $u = U/m$.

U : The *internal energy* of the fluid. It is a function of temperature only for perfect gases.

U is a strong function of temperature and weak function of pressure for non-perfect gases, vapors, and liquids. It is a measure of the molecular activity and interaction of the fluid.

m = The mass of the fluid in kg (lb)

H = The enthalpy of the fluid ($H = U + PV$)

P = The pressure of the fluid

V = The volume of the fluid

$h = H/m$

The units of c_p and c_v are J/(kg · K) or, Btu (lb_m · °R). They are related by the following equation:

$$c_p - c_v = R$$

where, R is the gas constant. The following relationships apply for ideal gases:

$$du = c_v dT$$

$$dh = c_p dT$$

where c_v and c_p are constants. They are independent of the temperature for monatomic gases such as helium. However, they increase with temperature for diatomic gases, such as air. They also increase further with temperature for triatomic gases. The following equations apply for constant-specific heats, or for small changes in temperature:

$$\Delta u = C_v \Delta T$$

$$\Delta h = C_p \Delta T$$

The following equations apply for a large increase in temperature:

$$\Delta u = \int_1^2 C_v(T) dT$$
$$\Delta h = \int_1^2 C_p(T) dT$$

The gas constant is defined by the following equation:

$$k = c_p / c_v = c_p / (c_p - R)$$

The gas constant is also defined by the following equation:

$$k = (Mc_p) / (Mc_p - MR)$$

where *M* is the weight of a mole of gas (the molecular weight).

The gas constant for a gas mixture can be calculated as shown in the following example. Assume that a gas mixture contains 60% of gas A and 40% of gas B:

The following table shows the calculations required:

Gas	Mol %	Mol Gas/Mol of Mixture	<i>Mc_p</i> at Average Gas Temp	Product
A	60	0.6	(<i>Mc_p</i>) _A	0.6(<i>Mc_p</i>) _A
B	40	0.4	(<i>Mc_p</i>) _B	0.4(<i>Mc_p</i>) _B
	100.0	1.0		

$$\text{Product} = 0.6(Mc_p)_A + 0.4(Mc_p)_B$$

Therefore, *k* of the gas mixture can be calculated using the equation shown above as follows:

$$k = (\text{Product}) / (\text{Product} - MR)$$

13.4.12 Pseudo-Critical Conditions and Compressibility

The generalized compressibility curves shown in Fig. 13.7 are applicable to gas mixtures. The pseudo-critical pressure and temperature of the gas mixture are required to determine the compressibility of the mixture using Fig. 13.7. The mol % of each gas is multiplied by the individual critical temperature (K) or (°R) of the gas to determine the contribution of this gas to the mixture pseudo-critical temperature. The same process is used for the mixture pseudo-critical pressure. The following example shows how the mixture pseudo-critical temperature and pressure are determined.

Gas	Mol %	Individual Critical Temp (K)	Pseudo <i>T_c</i> (K)	Individual Critical Pressure (bar abs)	Pseudo <i>P_c</i> (bar abs)
H ₂	60	33.24	19.94	12.97	7.78
N ₂	40	126.2	50.48	33.99	13.60
Mixture pseudo critical			70.42		21.38

The compressibility factor, Z , can be obtained now from Fig. 13.7 using the mixture pseudo-critical temperature and pressure.

13.4.13 Weight-Basis Item

The gas properties of a mixture are determined by having each component that contributes a share of its own property in proportion to its fraction of the total *weight*. The following equations show how the gas properties of a mixture are calculated using weight factors:

$$R' = \frac{W_a R'_a + W_b R'_b + W_c R'_c + \dots}{W}$$

$$c_p = \frac{W_a c_{pa} + W_b c_{pb} + W_c c_{pc} + \dots}{W}$$

$$c_v = \frac{W_a c_{va} + W_b c_{vb} + W_c c_{vc} + \dots}{W}$$

where R = the mixture gas constant

c_p = the mixture-specific heat at constant pressure

c_v = the mixture-specific heat at constant volume

W_a = weight of gas a

W_b = weight of gas b

W_c = weight of gas c

R_a = gas constant of gas a

R_b = gas constant of gas b

R_c = gas constant of gas c

W = total weight of mixture

Subscript a , b , and, c refer to gases a, b, and, c, respectively.

13.4.14 Compression Cycles

The *isothermal compression* and the *near-adiabatic (isentropic) compression* processes are applicable to positive displacement compressors. These processes are theoretical. They are not commercially attainable. However, they are used as a basis for calculations and comparisons. Isothermal compression occurs when the temperature is maintained constant and the pressure increases. This process requires continuous removal of the heat generated by compression. The following formula applies to isothermal compression:

$$P_1 V_1 = P_2 V_2 = \text{constant}$$

Near-adiabatic (isentropic) compression occurs when heat is not added nor removed from the gas during compression. The following formula applies for this process:

$$P_1 V_1^k = P_2 V_2^k$$

where k is the ratio of the specific heats (c_p/c_v).

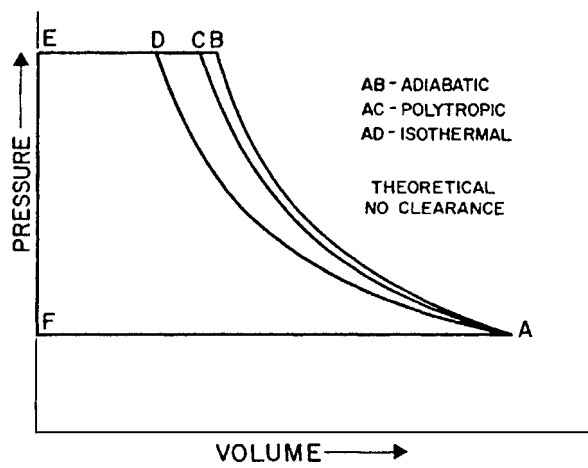


FIGURE 13.8 A P-V diagram illustrating theoretical compression cycles. (Source: Dresser-Rand Company, Painted Post, N.Y.)

Figure 13.8 illustrates the theoretical isothermal and adiabatic cycles on a pressure-volume diagram. The pressure ratio ($P_{\text{discharge}}/P_{\text{inlet}}$) of the compressor in this application is equal to 4. The work required for the isothermal compression is represented by area ADEF. The work required for the adiabatic compression is represented by area ABEF. Obviously, the work required for the isothermal compression is considerably less than that for the adiabatic compression. However, the isothermal compression cycle is not commercially achievable. Nevertheless, compressors are designed to maximize the amount of heat released from the gas during compression. The actual compression occurs along the *polytropic cycle*. The following formula applies for this process:

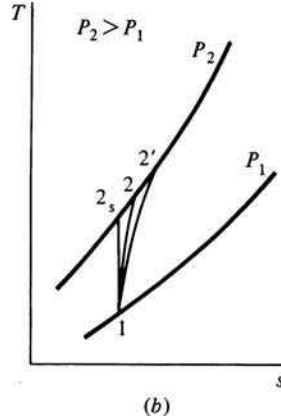
$$P_1 V_1^n = P_2 V_2^n$$

The exponent n is determined experimentally for a given application. It could be lower or higher than the adiabatic exponent k . The value of n is normally lower than k in positive displacement compressors. The polytropic compression cycle for a reciprocating compressor cylinder having a water jacket is also illustrated in Fig. 13.8. The polytropic process is thermodynamically irreversible. This indicates that this process experiences losses due to friction between the molecules of the gas. However, the isentropic process is reversible. Thus, this process does not consider the losses that occur due to friction between the molecules of the gas. The isentropic process is ideal. It is only a theoretical process. However, it is used to determine the *compressor polytropic efficiency* (Sec. 13.1.19). The value of n can be determined if the inlet and discharge pressures and temperatures are known. The following formula is normally used to determine the value of n :

$$T_2/T_1 = (P_2/P_1)^{(n-1)/n} = r^{(n-1)/n}$$

This formula is also used to determine the discharge temperature of the compressor when n is known. There has been a tendency to use the symbols n and k interchangeably

FIGURE 13.9 Compression of a gas from P_1 to P_2 on the T-S diagram. 1-2_s adiabatic reversible, 1-2 adiabatic irreversible. (Source: El-Wakil, M. M., *Power Plant Technology*, McGraw-Hill, New York, 1984, with permission from McGraw-Hill.)



to represent the ratio of specific heats (c_p/c_v). This is incorrect. The value of n is normally quite different from k . These two symbols should be differentiated carefully.

13.4.15 Compressor Polytropic Efficiency

Figure 13.9 illustrates the compression of an ideal gas from pressure P_1 to P_2 on the temperature-entropy diagram. Lines P_1 and P_2 are constant-pressure lines (isobars) for a perfect gas. The shape of these lines can be ascertained from the following perfect-gas relationship:

$$s_2 - s_1 = c_p \ln (T_2/T_1)$$

This equation is listed in Table 13.2. The isentropic compression process is represented by line 1-2_s. This process is adiabatic and reversible. Process 1-2 is adiabatic and irreversible. This indicates that this process has experienced *internal irreversibilities* (losses). The cause of this internal irreversibility is friction between the molecules of the gas. This process did not experience any *external irreversibility* (losses). This type of irreversibility consists of the following:

1. Heat loss from the compressor to the surroundings. (Since this process is adiabatic, it cannot experience losses to the surroundings.)
2. Mechanical friction in the compressor bearings.

The irreversibility manifests itself by an increase in temperature of the gas leaving the compressor at P_2 . Thus, $T_2 > T_{2s}$. The entropy at point 2 is higher than at point 2_s. Process 1-2 illustrates a compression process having more irreversibilities than process 1-2_s. Thus, a further increase in entropy occurs in an adiabatic process due to greater irreversibility. This indicates that process 1-2' has required additional power than process 1-2 to increase the pressure across the compressor by the same amount. This additional power requirement manifests itself in a higher discharge temperature from the compressor ($T_{2'}$). Therefore, the following relationships apply to this case:

$$T_{2'} > T_2 > T_{2s} \quad \text{and} \quad s_{2'} > s_2 > s_{2s}$$

However, since $dh = c_p dt$ for gases, then

$$H_{2'} > H_2 > H_{2s}$$

This means that the power required by the compressor $|W_c|$ increases with irreversibility. Therefore,

$$H_{2'} - H_1 > H_2 - H_1 > H_{2s} - H_1$$

The compressor efficiency determines the degree of irreversibility. This term is known as the polytropic compressor efficiency, η_c . It is also known as isentropic or adiabatic compressor efficiency. This efficiency is equal to the ratio of ideal work to actual work. It is given by the following formula:

$$\eta_c = \frac{H_{2s} - H_1}{H_2 - H_1} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

For constant-specific heats, it is

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1}$$

The formula of the polytropic compressor efficiency is the inverse to that of polytropic turbine efficiency. This is because the power generated by a real expansion process is less than that of an isentropic process through a turbine.

13.4.16 Compressor Power Requirement

The power requirement of any compressor should be determined for the following reasons:

1. Sizing the driver
2. Designing and selecting the compressor components

The compressor polytropic efficiency relates the actual power requirement to the isentropic (theoretical) power requirement. However, the compressor polytropic efficiency does not include mechanical friction losses. These losses include the following:

1. Friction in the bearings
2. Friction between the moving components of the compressor, etc.

The mechanical friction losses should be added to the actual power requirement of the compressor which was determined using the compressor polytropic efficiency. These losses are about 5–12% of the actual power requirement of the compressor. They vary depending on the size and type of the unit.

Historically, the actual power requirement of the compressor was compared to the isothermal cycle. However, positive displacement machines are now compared to the isentropic or adiabatic cycle (using the compressor polytropic efficiency). The compressibility, Z , must be considered in the calculation of the compressor power requirement. This is because it has considerable influence on many gases. This influence becomes significant at high pressure.

The compressor inlet volume on a perfect gas basis (V_{p1}) is different from the one on a real gas basis (V_{r1}). These volumes are at the inlet pressure and temperature (P_1 and T_1). They are related by the following formula:

$$V_{r1} = V_{p1} Z_1$$

The compressor theoretical adiabatic single-stage horsepower requirement is given by the following formula:

$$P_T(\text{ad}) = \frac{p_1 V_{r1}}{229} \frac{k}{k-1} (r^{(k-1)/k} - 1) \frac{Z_1 + Z_2}{2Z_1}$$

where $P_{T(\text{ad})}$ = the compressor theoretical adiabatic horsepower requirement (horsepower)
 Z_2 = the compressibility factor at the discharge of the compressor

$P_{T(\text{ad})}$ represents the theoretical adiabatic area on the P-V diagram for the volume per minute (V_1) being handled. Some applications prefer to have V_1 as a 100 cubic feet per minute (cfm) of real gas. The following formula applies to these applications:

$$\frac{P_T(\text{ad})}{100} = \frac{P_1}{2.29} \frac{k}{k-1} (r^{(k-1)/k} - 1) \frac{Z_1 + Z_2}{2Z_1}$$

The following equation applies to isothermal cycles regardless of the number of stages:

$$P_T(\text{iso}) = \frac{p_1 V_{r1} \ln r}{229} \frac{Z_1 + Z_2}{2Z_1}$$

where r is the overall or total compression ratio.

13.4.17 Compressibility Correction

The compressibility factor, Z , is required to use the preceding equations. This factor should be determined at the inlet and discharge conditions of the compressor. Since the compressor inlet pressure and temperature are known, the compressibility factor at these conditions can be obtained directly by using any of the following methods:

1. The specific gas charts
2. The generalized compressibility charts illustrated in Fig. 13.7

However, the compressor discharge temperature is required to determine the compressibility factor at the discharge conditions. The discharge pressure of the compressor is normally known. The *theoretical* discharge temperature obtained from the adiabatic cycle can be used in calculations involving positive displacement units. However, many factors play a role in varying the actual compressor discharge temperature from the theoretical value. The error introduced by all these factors is normally small. Thus, the theoretical adiabatic temperature would provide an adequate approximation of the actual compressor discharge temperature.

The theoretical adiabatic compressor discharge temperature can be obtained using any of the following methods:

1. The temperature-entropy diagram for the gas involved assuming that the compression is *isentropic* (i.e., the entropy remains constant)
2. The following formula:

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{(k-1)/k} = r^{(k-1)/k}$$

Note that the absolute value of the pressure and temperature should be used for all the parameters listed in this equation.

All the equations listed in the previous section are theoretical. They are not affected by gas characteristics such as molecular weight, specific gravity and density at operating conditions. All these parameters affect the actual power requirement of the compressor. Designers use proper allowances to include the effects of all these parameters on the actual compressor power requirement.

13.4.18 Multiple Staging

All *basic* single-stage compressor elements have certain limiting operating conditions. The following are the most important limitations:

1. Discharge temperature
2. Pressure differential across the stage
3. Effect of clearance
4. Minimization of compressor power requirement

Multiple-stage compressors are used when any of these limitations is approached. Intercoolers are normally employed between the stages of a multistage reciprocating compressor. Figure 13.10 illustrates the pressure-volume (P-V) combined diagram of a two-stage compressor. This diagram shows that the isothermal compression requires the least amount of power. Thus, cooling the gas between the stages to the original intake temperature (back to the isothermal line) will reduce the power required in the second stage. The reduction in power due to intercooling is given by area ABCD. The following formula is used to minimize the compressor power requirement with *perfect intercooling* between the stages:

$$r_s = \sqrt[s]{r_t}$$

where r_s = compression ratio per stage
 s = number of stages
 r_t = overall compression ratio ($p_{\text{final}}/p_{\text{initial}}$)

For example,

Two-stage:	$r_s = \sqrt[2]{r_t}$
Three-stage:	$r_s = \sqrt[3]{r_t}$
Four-stage:	$r_s = \sqrt[4]{r_t}$

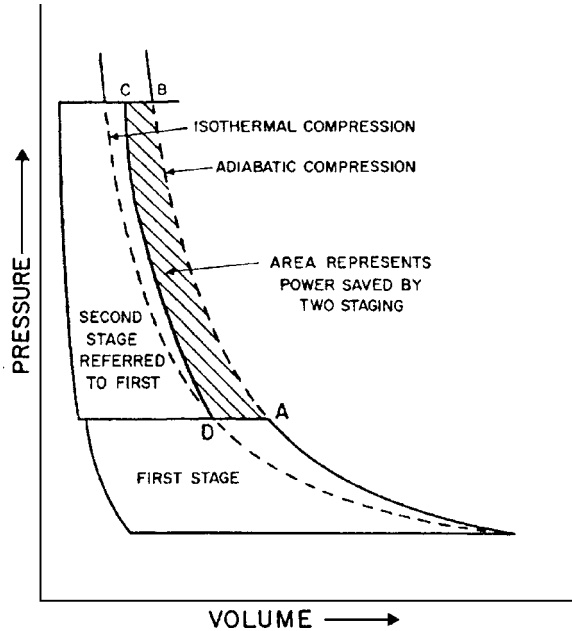


FIGURE 13.10 Combined P-V diagram for a two-stage air compressor. (Source: Dresser-Rand Company, Painted Post, N.Y.)

The implementation of this formula ensures that the compression ratio across each stage of the compressor remains constant. All the compressor stages are assumed to have the same inlet temperature as well.

The initial sizing of positive displacement compressors is based on this formula. However, the designer varies the compression ratios slightly to include other considerations.

13.4.19 Compressor Volumetric Flow Rate

The volumetric flow rate at the inlet of the compressor and inlet of the subsequent stages (m^3/s or ft^3/min) are generally the most required parameters for compressor applications. The word *dry* used in the following equations indicates that the gas or gas mixture involved does not contain any water vapor. The following equation provides the volumetric flow rate, V_1 , in cfm (ft^3/min); ($1 \text{ m}^3/\text{s} = 2118 \text{ ft}^3/\text{min}$):

$$V_1 = \text{scfm} \left(\frac{14.7}{p_1} \right) \frac{T_1}{520} Z_1$$

The scfm is measured at 14.7 psia, 60°F, dry ($101.3 \text{ kPa}_{(\text{abs})}$, 15.55°C). P_1 is in psia and T_1 is in °R.

The volumetric flow rate, V_1 in cfm, can also be obtained from the mass flow rate, W in lb/min, dry ($1 \text{ kg/s} = 132 \text{ lb/min}$) as follows:

$$V_1 = \frac{W(1545)T_1}{144p_1M} Z_1$$

The volumetric flow rate, V_1 in cfm, can also be determined from the mole flow rate (N mol/min, dry) as follows:

$$V_1 = \frac{N(379)(14.7)T_1}{p_1(520)} Z_1$$

The volumetric flow rate at the compressor inlet, V_1 in cfm, can also be determined from a volumetric flow rate measured at conditions other than those at the compressor inlet. The following formula provides V_1 as a function of cfm_{*g*} at P_g, T_g, Z_g dry:

$$V_1 = \text{cfm}_g \times \frac{p_g}{p_1} \frac{T_1}{T_g} \frac{Z_1}{Z_g}$$

The preceding equations can also be used if the gas contains water vapor. The proper molecular weight, M , should be used in the second equation listed above. The following correction factor is often used to determine the actual flow rate of a gas containing water vapor:

$$\frac{p_1}{p_1 - p_v}$$

P_v is the vapor pressure of the moisture. This correction factor is often multiplied by the volumetric flow rate calculated using any of the preceding equations to determine the flow rate of the wet gas.

13.4.20 Cylinder Clearance and Volumetric Efficiency

The cylinder clearance volume (Fig. 13.1) cannot be eliminated. It is between 4 and 16% for most cylinders. Some cylinders have a much larger clearance. These cylinders are used in special low-compression ratio applications. The normal clearance does not include the clearance volume that can be added in some applications for capacity control.

The compressor power requirement is not affected by normal variations in the clearance. However, the amount of clearance has a significant effect on the capacity of the compressor. The gas is trapped at discharge pressure in the clearance space at the end of the compression and delivery stroke. The gas pressure drops on the return stroke. The suction valves open when the pressure inside the cylinder becomes sufficiently lower than the intake pressure. Figure 13.11 illustrates the effect of the gas expansion on the quantity of gas drawn into the cylinder on a P-V diagram. The clearance has a significant effect on the compressor capacity. The volumetric efficiency is given by the following theoretical formula as a percentage:

$$\eta_v = 100 - C(r^{1/k} - 1)$$

where C is the cylinder clearance in decimal or %.

However, there are many factors that have an effect on the volumetric efficiency. These factors include the following:

1. Internal leakage
2. Gas friction

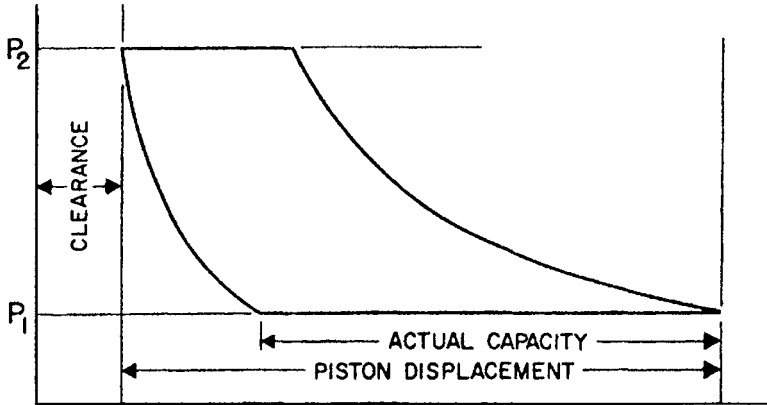


FIGURE 13.11 Work done on a volume of gas trapped in cylinder clearances (clearance volume) represents an inefficiency. (Source: Dresser-Rand Company, Painted Post, N.Y.)

3. Pressure drop across the valves
4. Inlet gas preheating

The term L is introduced to include the effects of these factors. The following is the volumetric efficiency formula that includes L :

$$\eta_v = 100 - C(r^{1/k} - 1) - L$$

The term L is difficult to determine accurately. However, the value of L for a moderate-pressure oil-lubricated air compressor is around 5%. The value of L increases as the molecular weight of the gas drops. This is due to the increased leakage.

The preceding equations indicate that the volumetric efficiency (V_E) decreases due to any of the following variations:

1. Increase in the clearance
2. Increase in the compression ratio
3. Decrease in k

Figure 13.12 illustrates a series of theoretical P-V diagrams based on the following assumptions:

1. $r = 4.0$
2. $k = 1.4$
3. Clearance of 7, 14, and 21%

The clearance effect on the volumetric efficiency is clearly shown.

Figure 13.13 illustrates the clearance effect on moderate- and high-compression ratio applications. This figure shows a P-V diagram for a pressure ratio of 7 and another for a pressure ratio of 4. All the remaining parameters of these two cases remain the

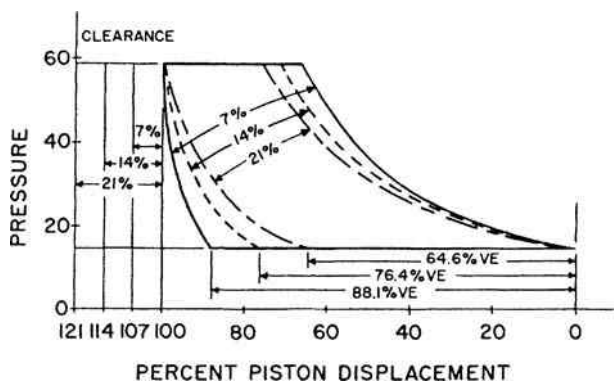


FIGURE 13.12 Theoretical P-V diagrams based on a compression ratio of 4.0, k of 1.4, and clearances of 7, 14, and 21%. (Source: Dresser-Rand Company, painted Post, N.Y.)

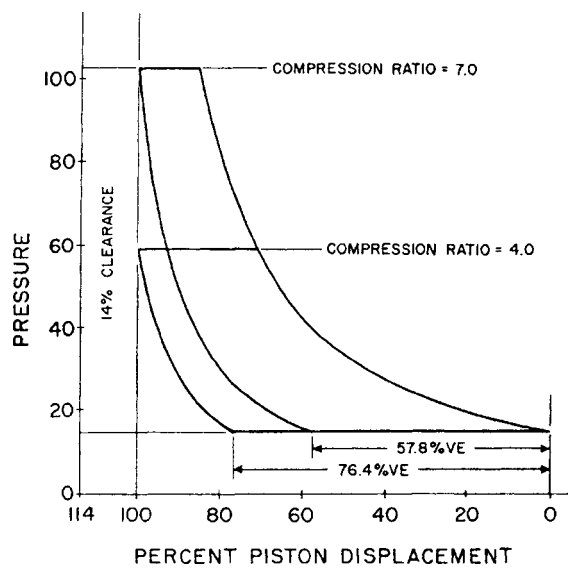


FIGURE 13.13 Effect of clearance at moderate- and high-compression ratio conditions. A P-V diagram for a ratio of 7 is superimposed on a diagram for a ratio of 4, all else being the same. (Source: Dresser-Rand Company, Painted Post, N.Y.)

same. The clearance value for both cases is relatively high (14%). This is the clearance value for any commercial compressor having a pressure ratio 7.

The effect of k on the volumetric efficiency is illustrated in Fig. 13.14. Compressor designers are obviously more concerned about the clearance value in applications having higher compression ratios or gases with low specific heat ratios. However, they will always endeavor to minimize the clearance to a suitable value for the valving and running clearances.

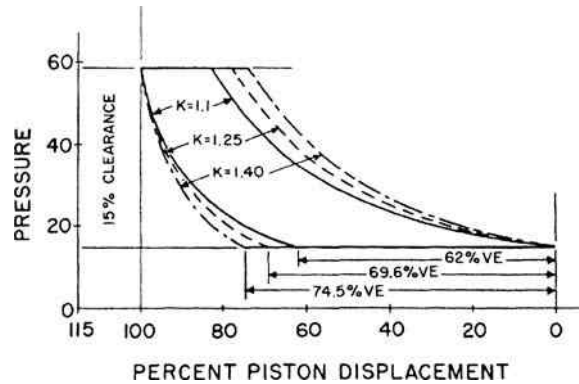


FIGURE 13.14 Effect of k on volumetric efficiency. The clearance is high, for illustrative purposes. (Source: Dresser-Rand Company, Painted Post, N.Y.)

13.4.21 Cylinder Clearance and Compression Efficiency

The area of the valves in a cylinder has a significant effect on the compressor polytropic efficiency (Sec. 13.1.19). The reduction in the size and number of these valves will result in an increase in the pressure drop across them. This will reduce the compressor polytropic efficiency [also known as compression efficiency (CE)]. However, it is necessary to reduce the size and number of these valves to obtain low clearance and high volumetric efficiency (V_E). This is because a larger clearance is required to accommodate the increase in size and number of these valves. Thus, the compressor cannot have high volumetric efficiency and polytropic efficiency. The size and number of the valves is increased when the compression ratio is low (2 or less). This will increase the compressor polytropic efficiency and reduce its volumetric efficiency. The size and number of valves is reduced as the compression ratio increases. The clearance is minimized at very high compression ratio (10:30). This will increase the volumetric efficiency and reduce the compressor polytropic efficiency.

13.5 Bibliography

- Bloch, H. P., *A Practical Guide to Compressor Technology*, 2nd ed., John Wiley & Sons Inc., Hoboken, New Jersey, 2006.
 El-Wakil, M. M., *Powerplant Technology*. McGraw-Hill, New York, 1984.

13.6 Appendix: List of Symbols

The following is the list of symbols that were used in this chapter:

c	cylinder clearance, % or decimal
c_p	specific heat-constant pressure, Btu/°F · lb
c_v	specific heat-constant volume, Btu/°F · lb
CE	compression efficiency, %
k	ratio of specific heats, dimensionless
M	molecular weight (MW), dimensionless

ME	mechanical efficiency, %
N	number of moles, dimensionless
$N_{a,b,c}$	moles of constituents, dimensionless
p	pressure, psia
$p_{a,b,c}$	partial pressure of constituents, psia
p_a	partial air pressure, psia
p_c	critical pressure (gas property), psia
p_r	reduced pressure, dimensionless
p_s	saturated vapor pressure, psia or inHg
p_v	partial vapor pressure, psia or inHg
psia	lb/in ² absolute, psi
psig	lb/in ² gauge, psi
P_t	theoretical horsepower, (work rate), hp
Q	heat, Btu
r	ratio of compression per stage, dimensionless
r_t	ratio of compression—total, dimensionless
R_0	universal or molar gas constant, ft · lb/mol · °R (1545 when p is in lb/ft ²)
R'	specific gas constant, ft · lb/lb · °R
RH	relative humidity, %
s	number of stages of compression, dimensionless
S	entropy, Btu/lb · °F
SH	specific humidity, lb moisture/lb dry gas
SPT	standard pressure and temperature, 14.696 psia and 60°F
T	absolute temperature, °R
T_c	critical temperature, °R
T_r	reduced temperature, dimensionless
v	specific volume, ft ³ /lb
$v_{a,b,c}$	partial volume of constituents, ft ³ /lb
v_r	pseudo-specific reduced volume, ft ³ /lb
V	total volume, ft ³
VE	volumetric efficiency, %
W	weight, lb
W_a	weight of dry air in a mixture, lb
W_v	weight of vapor in a mixture, lb
$W_{a,b,c}$	weight of constituents in a mixture, lb
Z	compressibility factor, dimensionless
η_v	volumetric efficiency, %

CHAPTER 14

Compressor Types and Applications

14.1 Introduction

The two basic categories of compressors are:

1. Positive displacement compressors
2. Dynamic compressors

Positive displacement compressors have the following characteristics:

1. They are constant volume, variable energy (head) machines.
2. They are affected significantly by the gas characteristics.

Dynamic compressors require the least amount of maintenance. Thus, they are the first choice when the application requirements (discharge pressure, flow) fall within their capability range. Rotary compressors are normally the next choice. This is because of the following characteristics:

1. They do not contain valves.
2. They are free of gas pulsations.

Reciprocating compressors are the last choice due to the following reasons:

1. They require the highest amount of maintenance.
2. They produce gas pulsations.

Figure 14.1 presents a flow range chart illustrating the various types of compressor applications as a function of flow (atmospheric cubic feet per minute, acfm), and discharge pressure (psig). Figure 14.2 also illustrates the application range of the various compressors used in industry. Table 14.1 lists the typical applications of the various types of compressors.

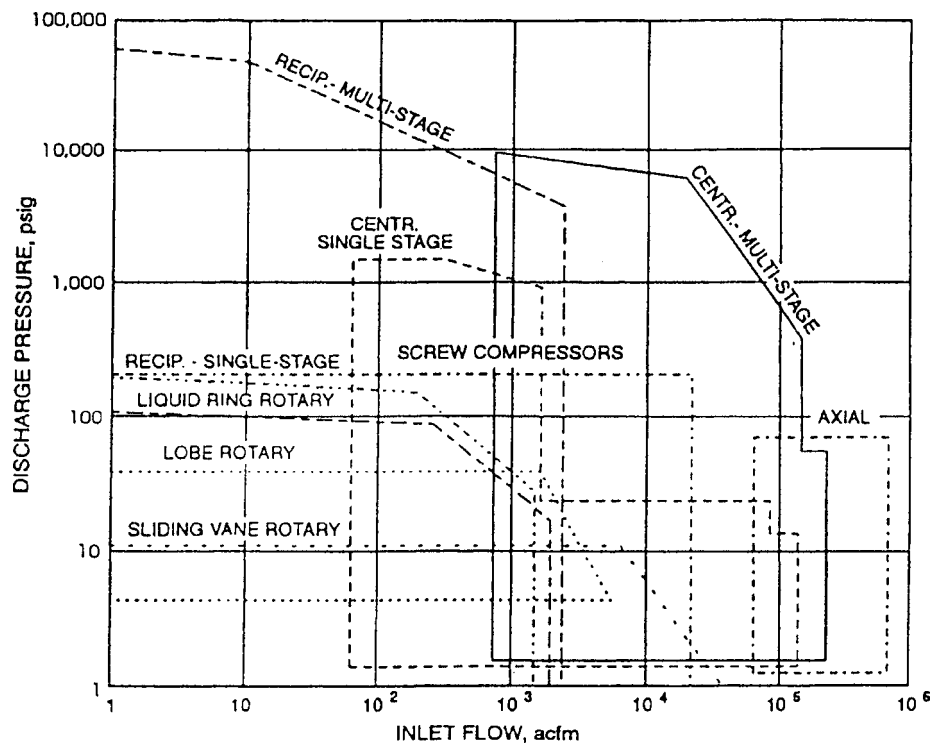


FIGURE 14.1 Compressor application range chart.

14.2 Positive Displacement Compressors

Positive displacement compressors are used in the following applications:

1. Low flows
2. Low molecular weight (hydrogen mixture) gases

The various types of positive displacement compressors (rotary and reciprocating) are described in the following sections.

14.2.1 Rotary Compressors

Rotary Lobe Compressors

Figure 14.3 illustrates a rotary lobe compressor (also known as a blower). It consists of identically synchronized rotors. These rotors are synchronized through external, oil-lubricated, timing gears. The timing gears have the following functions:

1. They positively prevent rotor contact.
2. They minimize meshing rotor clearance to optimize the efficiency of the machine.

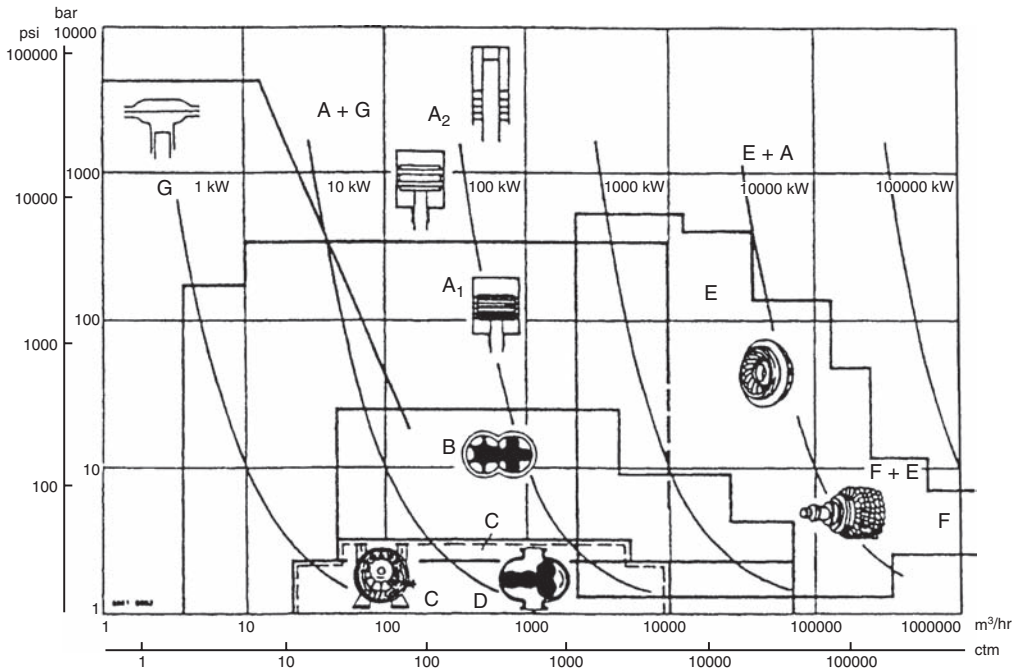


FIGURE 14.2 Application ranges for various types of compressors. (Source: Sulzer-Burckhardt, Winterthur and Basel, Switzerland.)

The text below describes the variables in Fig. 14.2:

- A_1 = reciprocating compressors with lubricated and nonlubricated cylinders
- A_2 = reciprocating compressors for high and very high pressures with lubricated cylinders
- B = helical- or spiral-lobe compressors (rotary screw compressors) with dry or oil-flooded rotors
- C = liquid ring compressors (also used as vacuum pumps)
- D = two-impeller straight-lobe rotary compressors, oil-free (also used as vacuum pumps)
- E = centrifugal turbocompressors
- F = axial turbocompressors
- G = diaphragm compressors

The most frequently used combinations of two different compressor types are identified three fields:

- A + G = oil-free reciprocating compressor followed by a diaphragm compressor
- E + A = centrifugal turbocompressor followed by an oil-free reciprocating compressor
- F + E = axial turbocompressor followed by a centrifugal turbocompressor

This feature allows the gas path of the compressor to be oil free. The rotors rotate in the opposite direction. There are two lobes on each rotor inside the compressor. During operation, the gas is trapped between the lobes and the compressor casing. The rotors push the gas from the inlet port, along the casing, to the discharge port. The leading lobe begins the discharge of the gas as its edge passes the edge of the discharge port. The entrapped gas is pushed by the trailing lobe into the discharge port. This compresses the gas against the backpressure of the downstream system. These compressors are supplied with noise enclosures or silencers to reduce their high noise level.

Machine Type	Capacity ICEM		T ₂ Max °F	P ₁ Max psia	P ₂ Max psia	P/R Min	P/R Max
	Min	Max					
Rotary lobe	1	40,000	350	35	55	1.0+	2.4
Rotary vane	45	3300	350	45	65	1.3	3.2
Rotary screw	50	20,000	350	150	615	2.0	6.0
Recip	1	10,000	800	1000	10,000	3.0	50.0
Liquid ring	10	10,000	N/A	100	140	1.0+	10.0
Centrifugal Single stage	700	150,000	500	1000	1400	1.0+	3.4
Centrifugal Multistage	300	150,000	800	2000	6000	2.0	10.0
Axial	75,000	350,000	800	30	150	1.0	10.0

TABLE 14.1 Typical Operating Range of Various Types of Gas Compressors

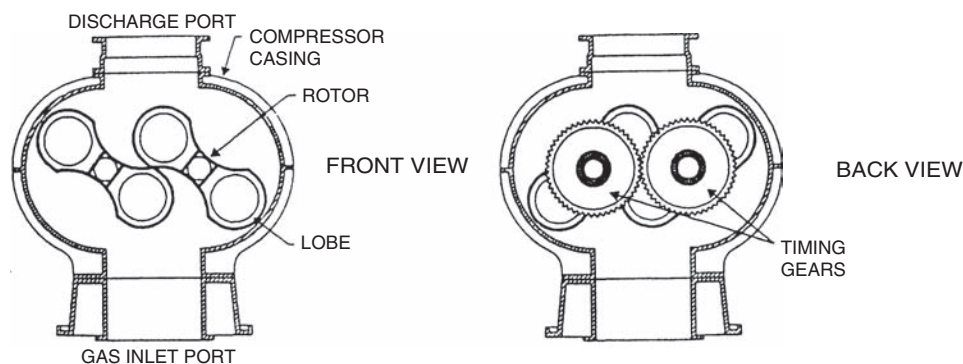


FIGURE 14.3 Rotary lobe.

Rotary Vane Compressors

Figure 14.4 illustrates a sliding rotary vane compressor. It uses a series of vanes that slide freely in radial slots that are machined into the rotor. Centrifugal force pushes the vanes outward against the casing wall. The chamber formed between the rotor, any two vanes, and the casing is known as a cell. During the operation, the volume of the cell increases as an individual vane rotates toward the end of the inlet port. This increase in the volume of the cell creates a partial vacuum in it. The vacuum draws the gas from the inlet port. The cell closes when the vane passes the inlet port. The gas becomes trapped between the two vanes, the rotor, and the casing. The volume of the cell decreases as the vanes continue to rotate toward the discharge port. This decrease in the volume of the cell increases the pressure in it. The high-pressure gas is discharged from the gas discharge port. These compressors are known to generate a high-noise level. The noise is caused by the motion of the vanes.

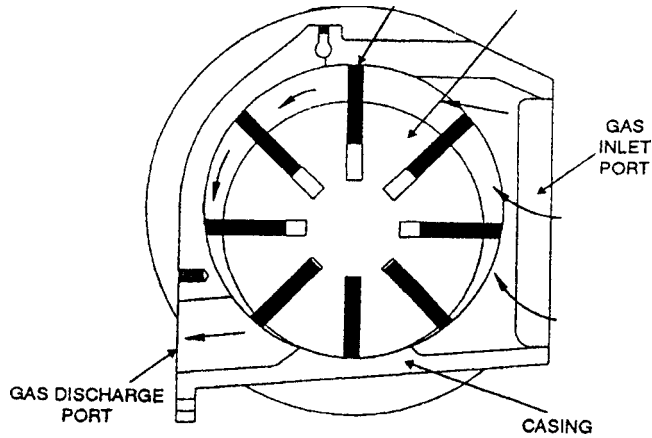


FIGURE 14.4 Rotary vane.

Rotary Screw Compressors

Figure 14.5 illustrates a typical single-stage screw compressor. This compressor consists of a pair of rotors that mesh in a dual-bore cylinder. There are four helical threads that are spaced 90° apart on the male rotor. The female rotor normally has six helical grooves that are spaced 60° apart. The thread-groove ratio is inversely proportioned to the rotor speed ratio. Thus, the female rotor will rotate at 1200 rpm in a four-thread, six-groove, screw compressor when the male rotor rotates at 1800 rpm. The male rotor normally drives the female rotor. A film of oil is usually injected between the rotors for the following reasons:

1. To provide a seal between the rotors
2. To prevent metal-to-metal contact between the rotors

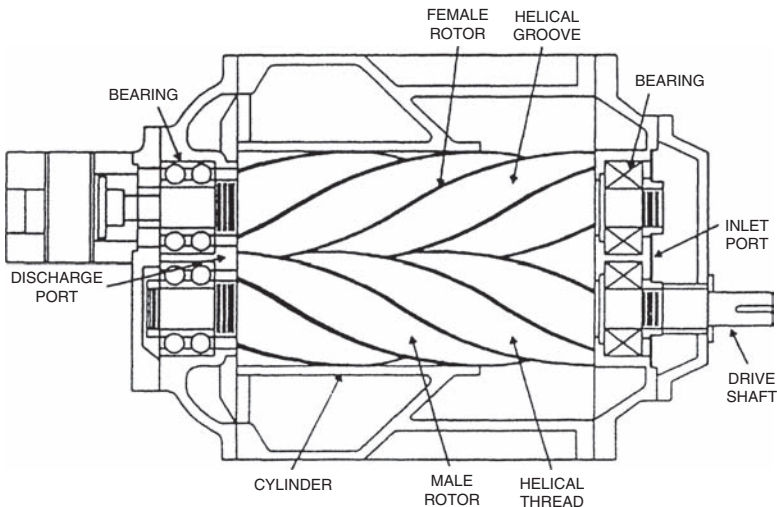


FIGURE 14.5 Rotary screw.

Applications such as plant and instrument air system require an oil-mist eliminator. This equipment is installed immediately downstream of the compressor. However, there are some screw compressor designs that do not require lubrication. These compressors are known commonly as “dry screw-type compressors.”

The inlet port and the drive-shaft are located at the same end of the compressor. The compression of the gas begins as the rotors mesh at the inlet port. The cavity between the male rotor threads and female rotor grooves draws the gas from the inlet port. As the shaft rotates, the rotor threads clear the edges of the inlet port. The gas is now trapped in a cell formed by the rotor cavities and the cylinder wall. The volume of this cell decreases with further rotation. This is caused by the male rotor threads rolling into the female rotor groove. The gas pressure in the cell increases as the volume decreases. Oil is injected in some compressors when the cell becomes closed to the inlet port. The oil performs the following functions:

1. Seals the clearances that exist between the threads and grooves
2. Absorbs the heat generated by the compression process

The gas pressure continues to increase until the rotor threads pass the edge of the discharge port. The compressed gas and oil mixture are released at this stage into the discharge port.

Rotary Liquid Ring Compressors

Figure 14.6 illustrates a typical liquid ring rotary compressor. This compressor consist of the following:

1. Elliptical casing
2. A round, multi-blade rotor that rotates in the casing

The elliptical casing is partially filled with a liquid (normally water). During operation, the rotor blades (also known as buckets) carry the liquid around the contour of the casing. The liquid leaves and returns to the space between the blades repeatedly while the shaft rotates. The space between the blades acts as a rotor chamber. The gas inlet and discharge are located around the rotor shaft. The gas is drawn into the rotor chamber when the liquid leaves it. The liquid then returns to the chamber and compresses the gas. The gas is then discharged to the process through a gas/liquid separator.

14.2.2 Reciprocating Compressors

Figure 14.7 illustrates a three-stage reciprocating compressor. The following are the basic components of this compressor:

1. Crankshaft
2. Crossheads
3. Piston rod packing
4. Cylinders
5. Pistons
6. Suction valves
7. Discharge valves

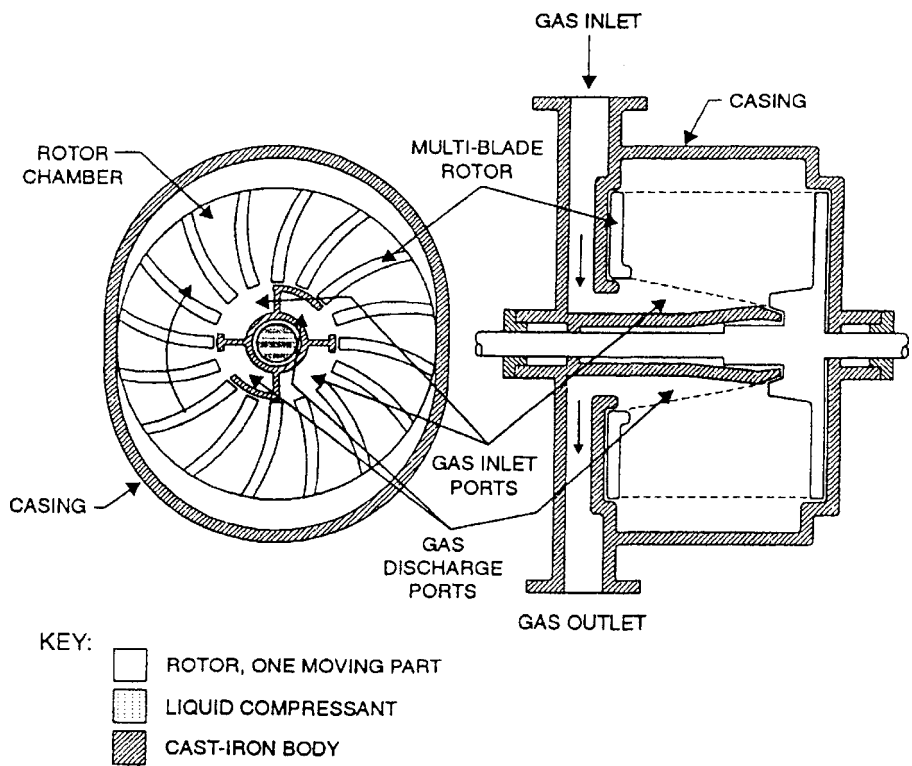


FIGURE 14.6 Rotary liquid ring. (Source: Reprinted with permission from Gardner Denver Nash LLC.)

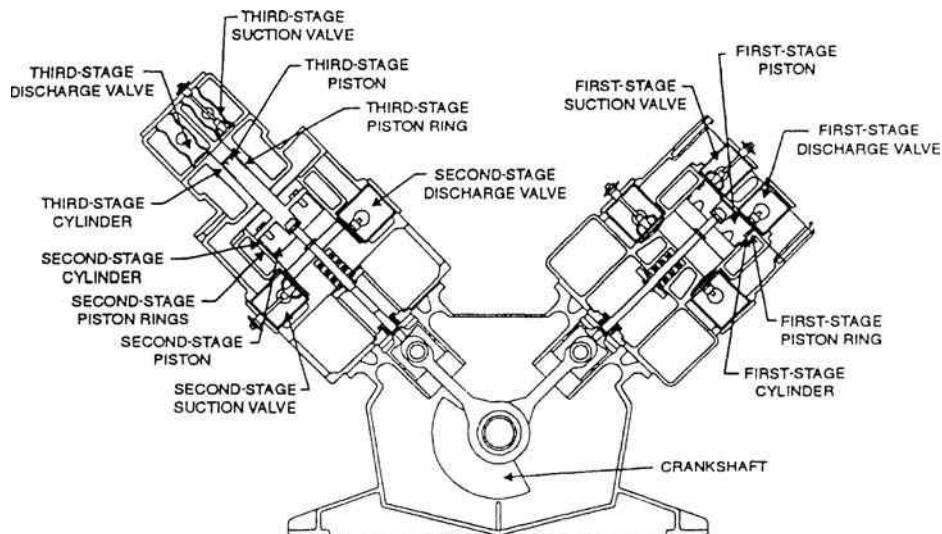


FIGURE 14.7 Reciprocating compressor.

Figure 14.8 provides the nomenclature of a reciprocating compressor. The prime mover (e.g., motor, etc.) rotates the crankshaft which converts this rotary motion into a reciprocating motion of the pistons. The compression cycle of a reciprocating compressor consists of the following:

1. A suction stroke
2. A compression stroke

The suction stroke begins with the initiation of the movement of the piston away from the inlet port of the cylinder. The pressure of the gas in the space between the piston and the inlet port drops quickly due to the rapid expansion of the gas. The pressure of this gas continues to drop until it reaches a value below that of the gas located on the opposite side of the suction valve. The suction valve opens due to the pressure difference across it. The gas starts to flow into the cylinder. It continues to flow until the piston reaches the end of its stroke. The compression stroke starts to move in the opposite direction. The suction valve closes when the pressure of the gas inside the cylinder exceeds the pressure of the gas located on the opposite side of the suction valve. The gas becomes trapped inside the cylinder at this stage. The pressure of this gas continues to increase as the piston moves further toward the end of the cylinder. The discharge valve opens when the pressure of the gas inside the cylinder reaches the design pressure of the stage. The gas is now discharged through the suction of the second stage. The gas is compressed further in the second stage. It is then discharged to the third stage. The gas is compressed to the final discharge pressure in the third stage. Figure 14.9 illustrates a balanced opposed, four-throw reciprocating compressor. This type of compressor is used normally in refinery hydrogen make-up service.

14.3 Dynamic Compressors

Dynamic compressors are used in every application where their capability (flow and pressure) meets the system requirements. This is due to their low maintenance requirements. The single-stage integral gear centrifugal compressor has substituted positive displacement compressors in many applications. These applications had previously preferred to use positive displacement compressors. The following are the types of dynamic compressors:

1. Centrifugal compressors
2. Axial flow compressors

14.3.1 Centrifugal Compressors

Principles of Operation of Centrifugal Compressors

The rotor of a centrifugal compressor increases the tangential velocity (V_t) of the gas (Fig. 14.10). Centrifugal pumps and compressors have a very similar principle of

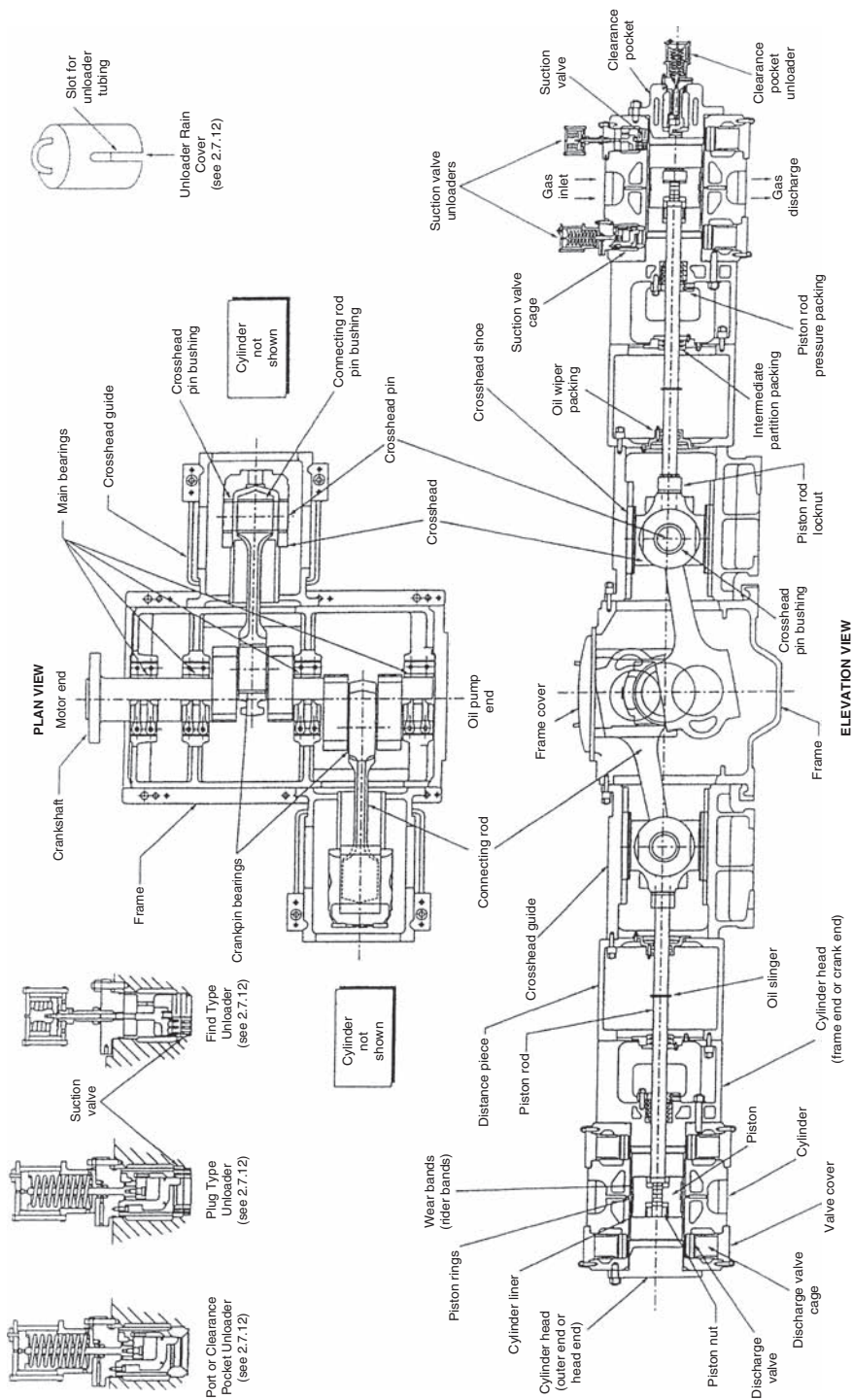


FIGURE 14.8 Reciprocating compressor nomenclature.

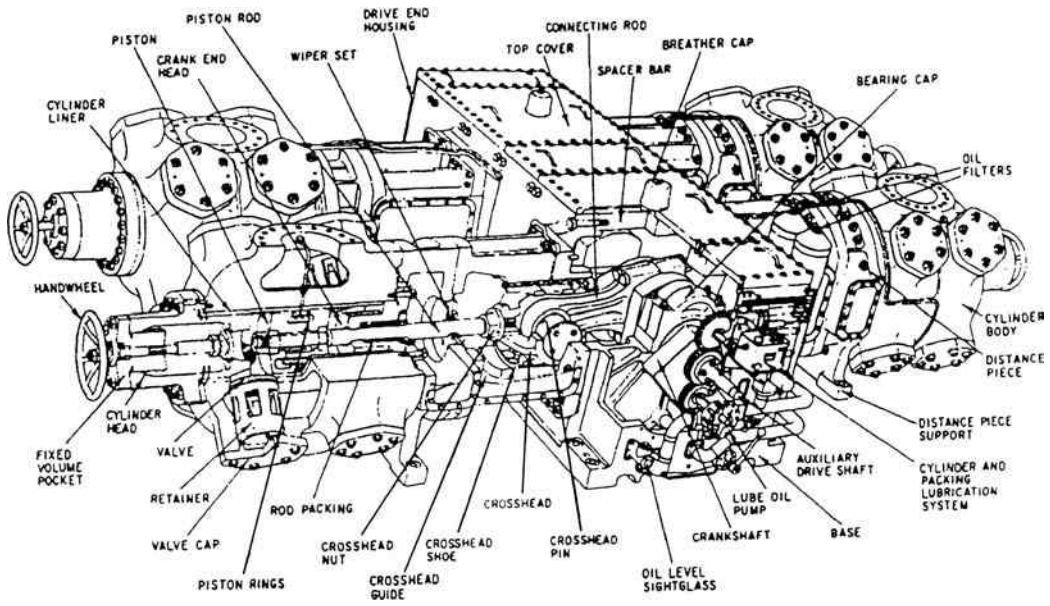


FIGURE 14.9 Four-throw reciprocating compressor.

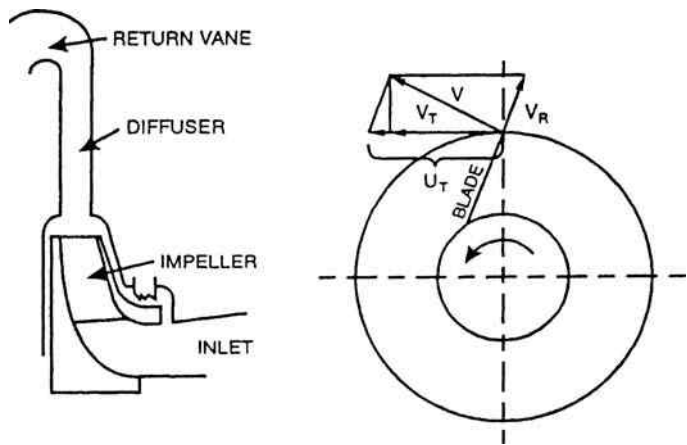


FIGURE 14.10 Centrifugal compressor.

operation. The gas flows through the inlet nozzle and enters the impeller. The inlet nozzle and the impeller are designed to ensure that the gas enters the impeller with minimum turbulence. The impeller consists of the following components:

1. Hub
2. Blades (known also as vanes)

The impeller is mounted on the shaft. It increases the energy of the gas. This increase in energy is proportional to the following product:

$$U_T \times V_T$$

where U_T = blade tip speed velocity

V_T = gas tangential velocity in the impeller

V_R is the gas velocity relative to the blade. V is the resultant velocity. It is given by the following equation:

$$V = V_R + U_T$$

The diffuser surrounds the impeller. It increases the pressure of the gas by reducing its velocity (conservation of energy—first law of thermodynamics). A volute casing surrounds the diffuser in single-stage compressors. The volute casing increases the pressure of the gas leaving the diffuser further by reducing its velocity to a lower value. The gas is discharged from the compressor through the discharge nozzle. Multi stage compressors employ return vanes. These components direct the gas leaving the diffuser into the impeller of the next stage.

Centrifugal Single-Stage (Low-Ratio) Compressors

Figure 14.11 illustrates a typical single-stage centrifugal low-ratio compressor. This equipment is known as a single-stage overhung compressor. This is because the impeller is outboard of the radial bearings. The cases of these compressors are radially split.

Centrifugal Single-Stage Integral Gear Compressors

Figure 14.12 illustrates a “Sundyne” single-stage high-speed compressor. A motor is used normally to drive this compressor through an integrally mounted gear box (not shown). These compressors have substituted positive displacement compressors in many applications. These applications include high-head and low-flow requirements. The speed of operation of these compressors varies between 8000 and 34,000 rpm. However, they are limited to 400 hp (298.4 kW).

Centrifugal Multi Stage Horizontally Split Compressors

Figure 14.13 illustrates a typical multi stage horizontally split centrifugal compressor. The casing splits horizontally along the centerline of the compressor. This feature provides access to the internal components of the compressor without disturbing the bearing alignment or the rotor to casing clearances. The piping nozzles should be installed on the lower half of the compressor casing if possible. This will allow disassembly of the compressor without removal of the process piping.

Centrifugal Multi Stage with Side Loads Compressors

Figure 14.14 illustrates a typical centrifugal multi stage with side loads compressors. These compressors are used exclusively for refrigeration applications. The only difference between these compressors and the type shown in Fig. 14.13 is that gas is induced or removed from these compressors through side-load nozzles. This type of compressor is available in either horizontally or radially split casing design.

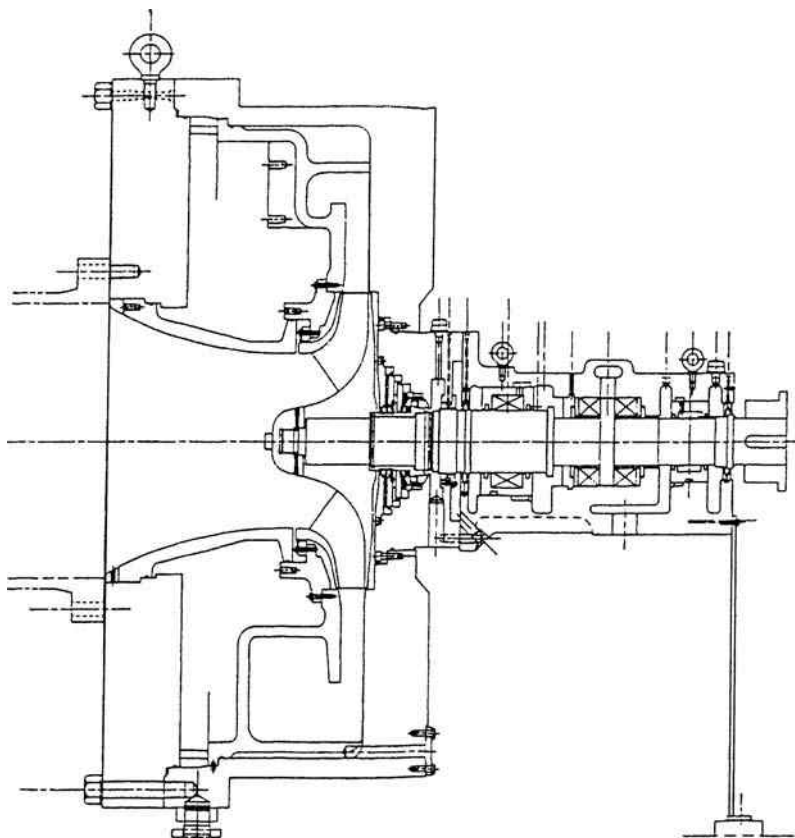


FIGURE 14.11 Single-stage overhung turbo compressor. (Courtesy of A-C Compressor Corp.)

Centrifugal Multi Stage (Barrel) Compressors

Figure 14.15 illustrates a typical multi stage, radially split, centrifugal compressor. The casing of this compressor is made from a complete cylinder. One end of the compressor can be removed to provide access to the internal components. These multi stage, radially split centrifugal compressors are normally called barrel compressors. They are used in the same applications as multi stage, horizontally split centrifugal compressors. They are also used in higher-pressure applications and some low molecular gas composition such as hydrogen gas mixtures. The barrel compressors are suitable for these applications because they are less prone to leakage than other types of compressors.

Centrifugal Multi Stage Integral-Gear Compressors

Figure 14.16 illustrates a typical four-stage, integrally geared centrifugal compressor. The prime mover rotates a low-speed (bull) gear. This gear drives two or more high-speed gears (pinions). Impellers are mounted beside one or both pinion gears. Each impeller has a separate casing. This casing is bolted to the gear casing. The gear casing in these designs is normally horizontally split to provide access to the gears.

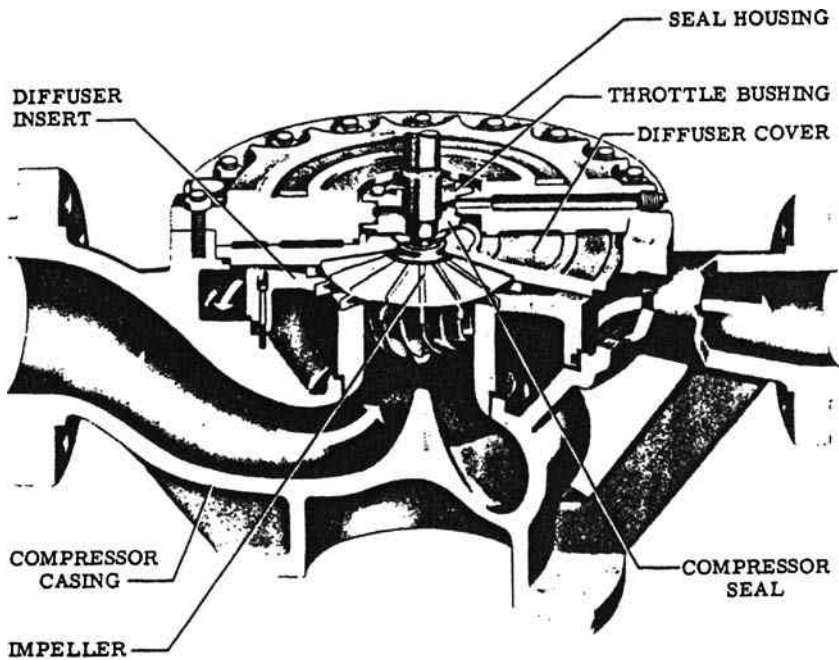


FIGURE 14.12 Single-stage high-speed compressor. (Courtesy of Sundstrand Corp.)

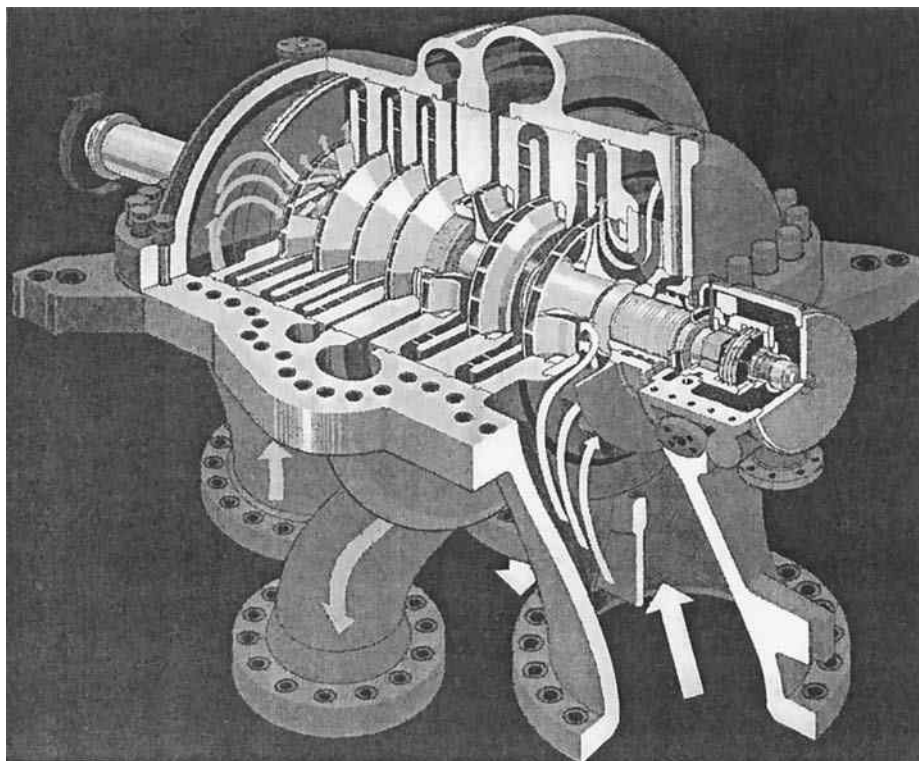


FIGURE 14.13 Centrifugal multi stage horizontal split. (Courtesy of Mannesmann Demag.)

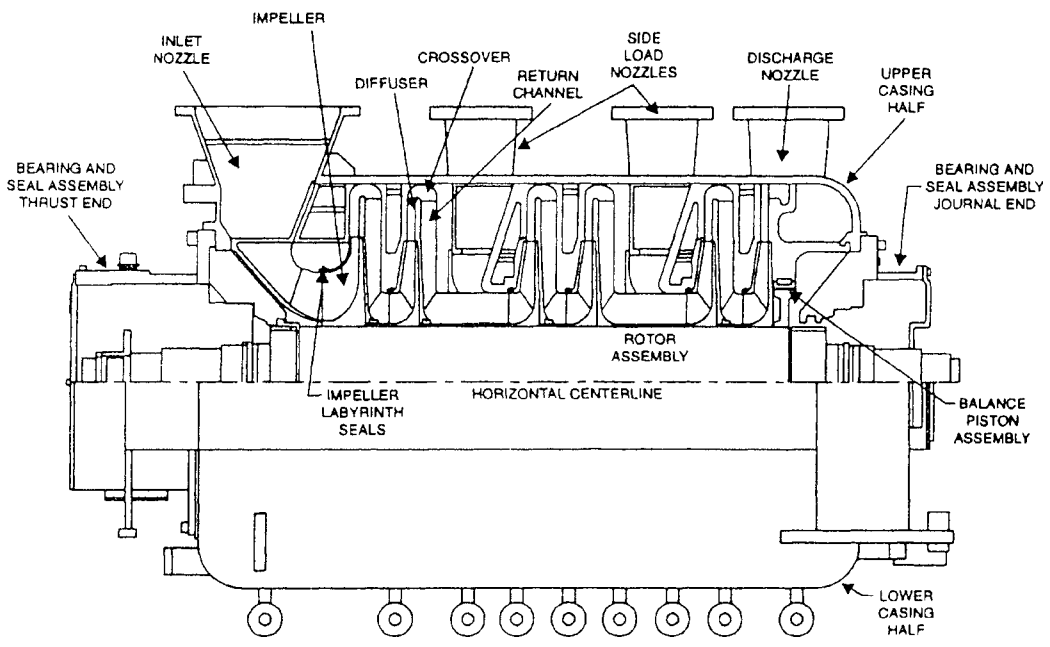


FIGURE 14.14 Typical multi stage refrigeration compressor.

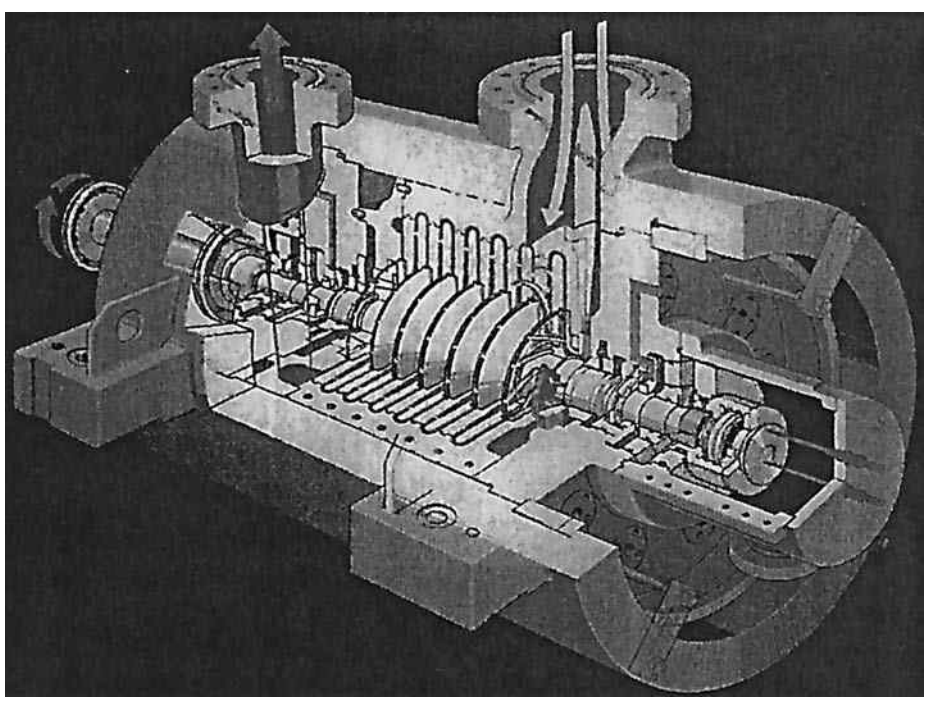


FIGURE 14.15 Typical multi stage, radially split centrifugal compressor. (Courtesy of Mannesmann Demag.)

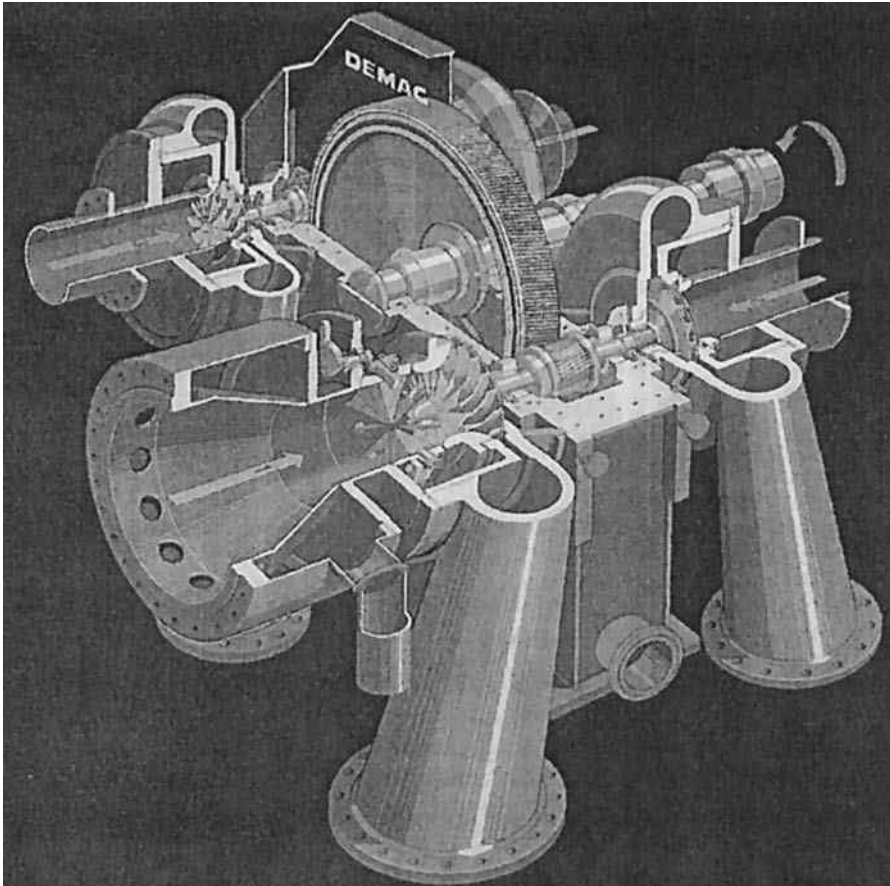


FIGURE 14.16 Typical integrally geared, centrifugal compressor. (Courtesy of Mannesmann Demag.)

The gas enters the compressor through the inlet nozzle to the first-stage impeller. The gas is discharged from the impeller into a diffuser. The gas is then discharged from the diffuser through a volute casing. The pressure of the gas increases in the diffuser and the volute casing. The gas is discharged from the volute casing through the first-stage discharge nozzle. It then enters an intercooler. The gas is then piped to the second stage. The gas leaving the second stage enters an intercooler. It is then carried by pipes to the third stage.

14.3.2 Axial Flow Compressors

Axial Horizontally Split Compressors

Figure 14.17 illustrates a typical axial compressor. It consists of the following:

1. A rotor shaft with a series of rotating blades
2. A tapered cylindrical casing with fixed stator vanes

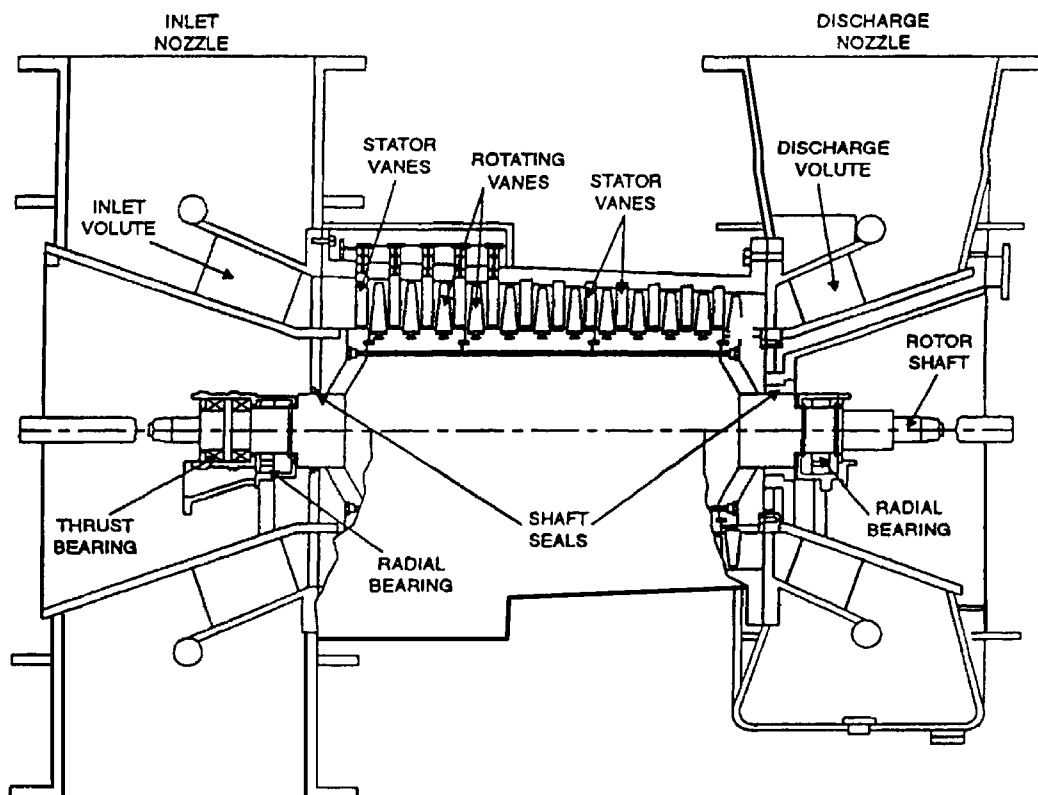


FIGURE 14.17 Typical axial compressor.

Each row of rotor blades is followed by a row of stator vanes. A compressor stage is defined as a row of rotating blades followed by a row of stationary blades. The gas enters through the inlet nozzle. This nozzle guides the gas to the inlet volute. The inlet volute directs and accelerates the gas flow into the first stage of stator vanes. These vanes direct the gas flow to align it properly with the rotating blades. These blades increase the velocity of the gas. The stator vanes act as a diffuser. They increase the pressure of the gas by decreasing its velocity. The stator vanes redirect the flow to the next row of rotating blades. This process of increasing the velocity of the gas by the rotating blades followed by increasing the pressure of the gas is repeated at each row. The gas flow exits from the compressor through the discharge volute and discharge nozzle.

14.4 Bibliography

- American Petroleum Institute, API Standard 618. *Reciprocating Compressors for Petroleum, Chemical and Gas Industry Service*, 4th ed., June 1995.
- Bloch, H. P., *A Practical Guide to Compressor Technology*, 2nd ed., John Wiley & Sons Inc., Hoboken, New Jersey, 2006.
- Forsthofer, W. E., *Forsthofer's Rotating Equipment Handbooks, Volume 3: Compressors*, 1st ed., Elsevier Ltd, Oxford, UK, 2006.

CHAPTER 15

Compressors

A compressor is a mechanical device that draws in gas and discharges it at higher pressure.

15.1 Compressor Types

Figure 15.1 provides a classification of the various types of compressors used in industry.

Positive displacement compressors use reciprocating or rotary action. The discharge flow produced by a reciprocating compressor is intermittent. A receiver is usually used in combination with it to absorb the resulting pulsation and stabilize the discharge pressure. The discharge flow from a rotary compressor is uniform. Figure 15.2 illustrates the operation of a positive displacement compressor. When the piston moves downward, it draws the intake valve open and pulls the gas into the cylinder (Fig. 15.2a). Then, the piston travels upward, and the intake valve is pushed closed (Fig. 15.2b). The gas is compressed until the pressure exceeds the discharge line pressure. Then, the discharge valve opens allowing the gas to enter the system.

15.2 Compressor Operation

The internal energy of a gas depends only on its temperature and is independent of the volume it occupies (Joules law). Therefore, the increase in the gas temperature represents the total energy expended to compress it. Most compressors use a heat exchanger to remove the heat from the discharge flow.

15.3 Gas Laws

Regardless of the type of compressor, all gases obey the laws of thermodynamics and the fundamental gas laws. The *general gas law* equation that closely approximates the behavior of gas is

$$\frac{P_1 \times V_1}{T_1} = \frac{P_2 \times V_2}{T_2}$$

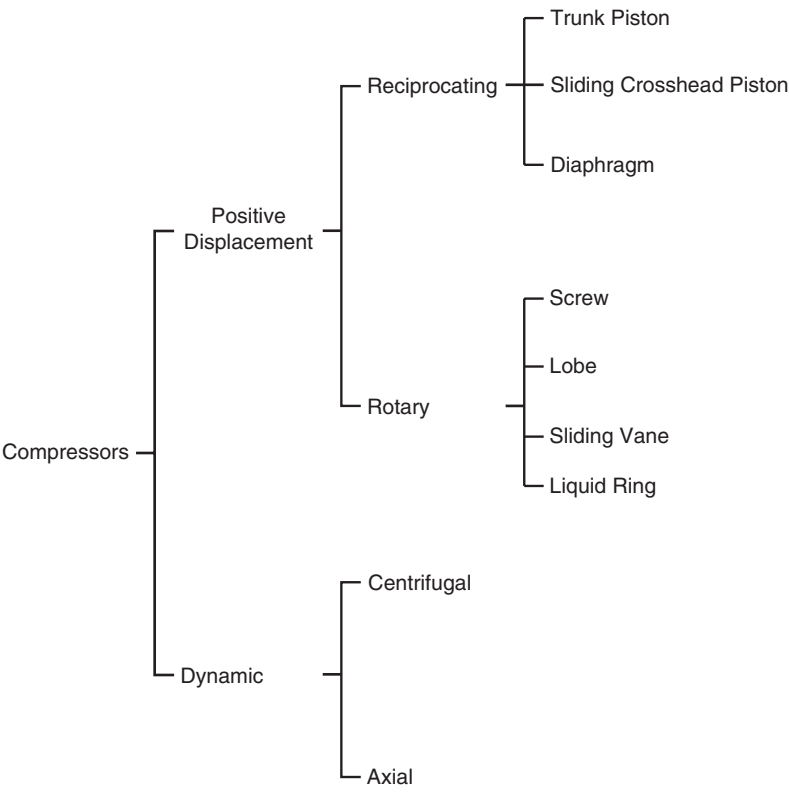


FIGURE 15.1 Classification of compressors. (Source: Reprinted with permission from National Resources Canada.)

where P_1, P_2 = initial and revised pressures [kPa_(absolute)]
 V_1, V_2 = initial and revised volumes (L)
 T_1, T_2 = initial and revised temperatures (K)
 $\text{kPa}_{(\text{absolute})} = \text{kPa}_{(\text{gauge})} + 101.325$
Kelvin units (K) = °C + 273.15

The general gas law stated in terms of the volumetric flow rate can be obtained by substituting the volume in the equation of the general gas law with the volumetric flow rate, f (L/s). For air, the equation is

$$\frac{P_1 \times f_{a1}}{T_1} = \frac{P_2 \times f_{a2}}{T_2}$$

15.4 Compressor Performance Measurement

The compressor performance is determined by calculations based on gas conditions at the inlet and outlet, and measured flow rate at the outlet. Manufacturers rate compressors normally by stating the equivalent *free* or *standard air flow rate*, at various discharge pressures based on free gas intake.

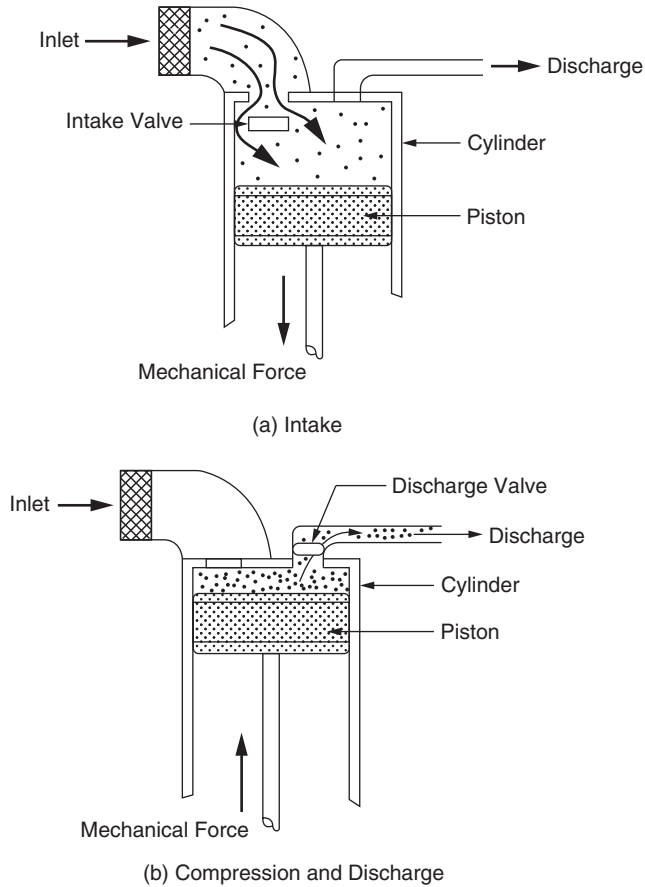


FIGURE 15.2 Positive displacement compressor. (Source: Reprinted with permission from National Resources Canada.)

The gas law equation is used to convert the measured flow rate at a particular pressure to the equivalent free gas flow rate. The flow rate from a positive displacement compressor is determined by measuring the time required to produce a given pressure rise in a closed pressure vessel, of a known volume. The initial and final pressures and temperatures in the receiver are measured and used in the gas law equation to calculate the equivalent volume of free gas delivered during the measured time period, and therefore the average free gas flow rate.

For example, a reciprocating-type compressor with a water-cooled heat exchanger on the discharge air operated 120 seconds to raise the pressure in a 275-L receiver from 651 kPa_(absolute) (P_1) to 790 kPa_(absolute) (P_2). The initial temperature (T_1) was measured to be 22°C (295.15 K) and the final temperature (T_2) was measured to be 29°C (302.15 K).

$$\begin{aligned} \text{Initial equivalent free air volume} &= \frac{P_1 \times V_1 \times 293.15}{T_1 \times 101.325} \\ &= \frac{651 \times 275 \times 293.15}{295.15 \times 101.325} = 1755 \text{ L} \end{aligned}$$

Similarly, the equivalent free air volume at the final conditions can be calculated.

$$\begin{aligned}\text{Final equivalent free air volume} &= \frac{P_2 \times V_2 \times 293.15}{T_2 \times 101.325} \\ &= \frac{790 \times 275 \times 293.15}{302.15 \times 101.325} = 2080 \text{ L}\end{aligned}$$

The average equivalent free air flow rate can then be calculated.

Equivalent free air flow rate

$$\begin{aligned}&= \frac{\text{Final equivalent free air volume} - \text{Initial equivalent free air volume}}{\text{Time}} \\ &= \frac{2080 - 1755}{120} = 2.71 \text{ L/s}\end{aligned}$$

15.4.1 Inlet Conditions

The compressor performance is directly affected by the temperature and pressure of the gas at the inlet. When the density of the gas entering the compressor increases, a greater quantity of gas is compressed for a given volume.

Intake screens, filters, and ductwork should be selected to minimize the pressure drop across them (flow resistance). This is done to minimize the energy used by the compressor.

15.4.2 Compressor Performance

The compressor-operating characteristics are provided by the manufacturer. The performance tables usually provide the equivalent free gas flow rates at certain operating speeds and discharge pressures.

The capacity of some positive displacement compressors depends on the working pressure. It is also affected by internal leakage and the *clearance volume* which is the volume of gas that remains in the compressor on each compressor cycle. The capacity of a dynamic compressor varies with the working pressure. Figure 15.3 illustrates a typical performance characteristics for positive displacement, centrifugal, and axial compressors operating at constant speed.

15.4.3 Energy Available for Recovery

All compressor systems cool the discharge gas because all the energy expended by the compressor is represented entirely by an increase in gas temperature

Small compressors usually use heat-dissipating fins on the compressor casing to enhance the heat transfer from the discharge gas. Intercoolers are used in multistage compressors (Fig. 15.4), and many systems use an aftercooler to cool the gas after the final stage.

The specific volume of the gas decreases when it is cooled. This results in the use of smaller pipes and/or reduction in the friction losses. Most of the energy expended in the compression process is recoverable in the intercooler and aftercooler. It is used sometimes for other useful purposes.

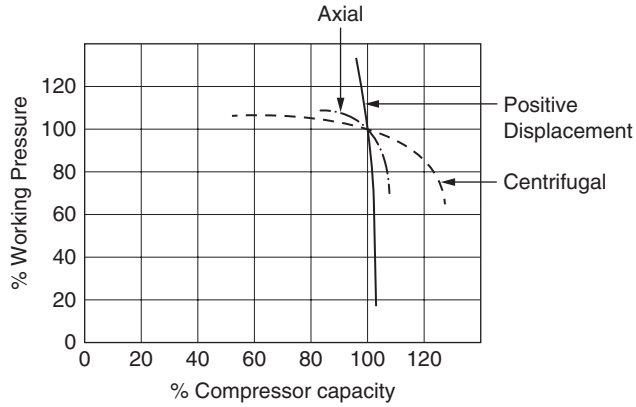


FIGURE 15.3 Typical compressor performance characteristics.

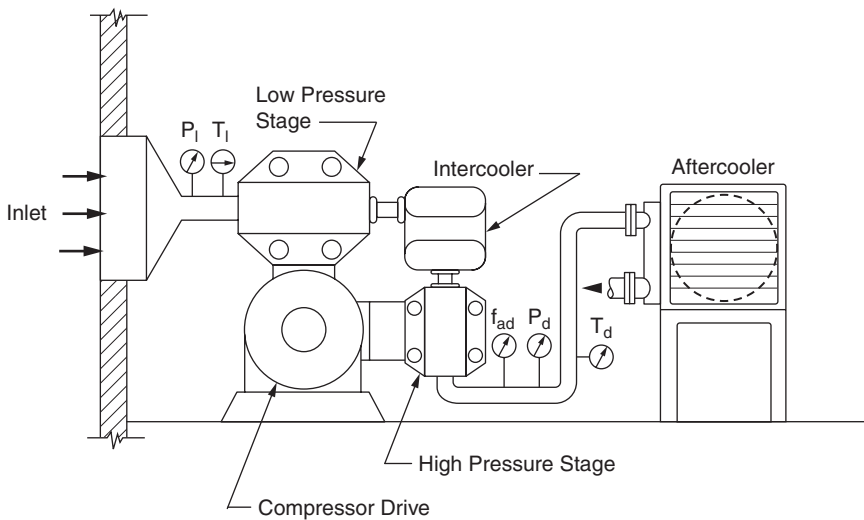


FIGURE 15.4 Arrangement of compressor components. (Source: Reprinted with permission from National Resources Canada.)

15.4.4 Positive Displacement Compressors

The volume of gas being compressed in positive displacement compressors is reduced in a cylinder or rotor. The types of positive displacement compressors are reciprocating and rotary.

15.4.5 Reciprocating Compressors

The most common types of positive displacement compressors are reciprocating compressors. The common characteristics of these units are listed in Table 15.1.

Compressor	Maximum Capacity (L/s)	Maximum Pressure [kPa _(gauge)]	Maximum Power (kW)
Trunk type			
Single stage	20	1040	11
Multistage	320	1724	93
Sliding crosshead			
V type	755	860	120
L and horizontal	4700	3500	3700
Diaphragm	2	420	1

TABLE 15.1 Common Characteristics of Reciprocating Compressors

Reciprocating compressors are used widely because they are simple to operate and maintain, compact, and have low capital cost. Their operation speed is low, they are easily regulated, and have good efficiency over the capacity range. However, they generate internal heat due to friction. Vibration is generated by the reciprocating action. This necessitates more robust foundation than other compressors.

15.4.6 Trunk Piston Compressors

Figure 15.5 illustrates a trunk piston compressor. Their design limits them to single-acting and lubricated applications.

A connecting rod connects the piston to the crankshaft, and the bottom of the cylinder is open to the crankcase. Compression is restricted to the top of the piston because the bottom of the cylinder cannot be sealed. This makes the unit single acting).

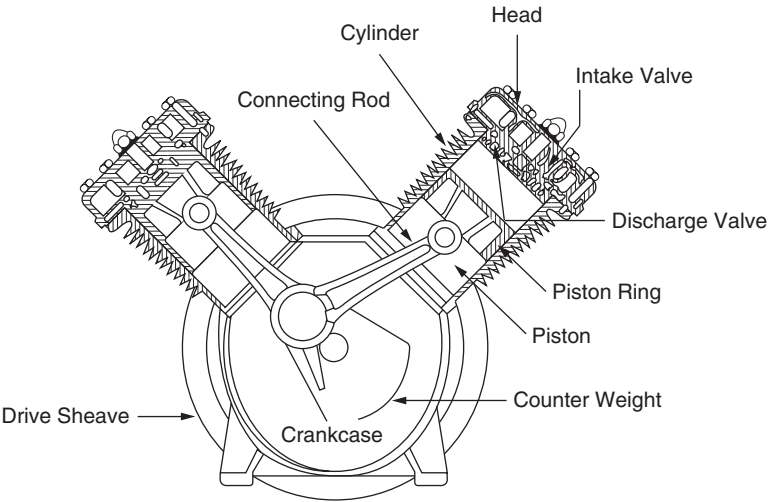


FIGURE 15.5 Trunk piston compressor. (Source: Reprinted with permission from National Resources Canada.)

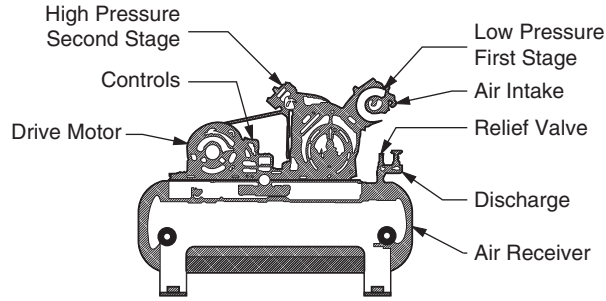


FIGURE 15.6 Small compressor package. (Source: Reprinted with permission from National Resources Canada.)

The lubricating oil in the crankcase splashes on the exposed cylinder walls. Due to this method of lubrication, the exhaust gas sets contaminated with oil. Two or more cylinders can be used in single-stage units. They are usually air cooled. Two-stage machines are larger and cooled by air and water. Higher-capacity units usually use water as a coolant.

Figure 15.6 illustrates a small compressor package where the compressor is mounted directly on an air-receiving tank. It forms a package system complete with piping, controls, relief valve, and motor.

15.4.7 Sliding Crosshead Piston Compressors

Figure 15.7 illustrates a sliding crosshead piston compressor. The piston is connected to the crosshead by a rigid rod. A seal around the rigid rod allows compression on both sides of the piston (double acting).

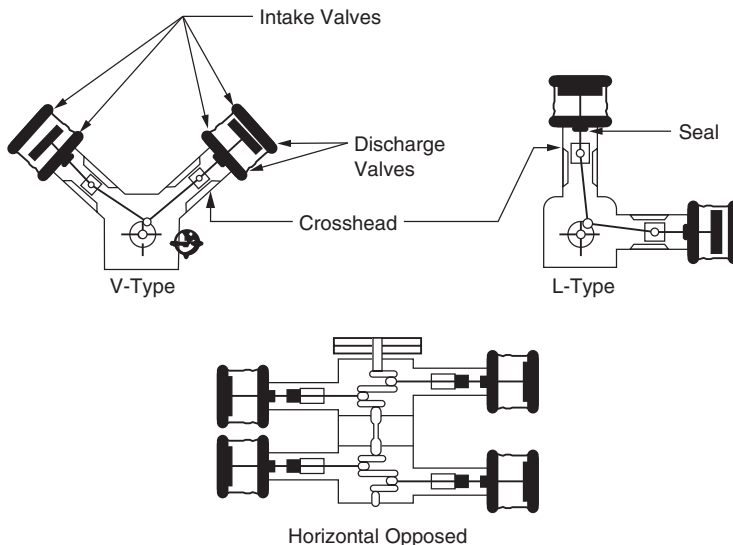


FIGURE 15.7 Sliding crosshead piston compressors. (Source: Reprinted with permission from National Resources Canada.)

The cylinder is isolated from the crankcase by the seal. These compressors are always double acting but can be single or multistage.

These machines can be oil-lubricated or oil free because the piston is isolated. A special self-lubricating materials or coatings on the piston, cylinder walls, and piston rings are used in the oil-free design. Special industrial applications, instrument air, and industrial medical air require oil-free design.

These compressors are available in the following configurations: vertical, vertical V, L-type, and horizontal. They are also available in various capacities and pressure ratings. The discharge pressure of some multicylinder horizontal units can reach 60 MPa. A labyrinth design is used in a modified version of the sliding crosshead compressor. Concentric grooves are used on the piston and on the cylinder walls instead of sealing rings. The gas held in the grooves separates the surfaces, resulting in minimal piston friction. However, the gas leakage past the piston is high. The discharged gas is contaminant free but at reduced efficiency.

15.4.8 Diaphragm Compressors

Figure 15.8 illustrates a diaphragm compressor. A flexible membrane, or diaphragm, is used to perform the same function as the piston.

The connecting rod is pinned to a block, or piston. The diaphragm is attached to the piston. A different design has the piston separated from the diaphragm by a chamber filled with fluid that is used to move the diaphragm hydraulically. These compressors provide oil-free gas because the diaphragm isolates the gas from the working parts of the compressor.

The main features of diaphragm compressors are: inexpensive, light, compact, simple, easily maintained. Due to low friction, the discharge gas is cool and delivered at low noise. Also, these compressors can withstand impurities in the gas because the gas does not come in contact with the working parts.

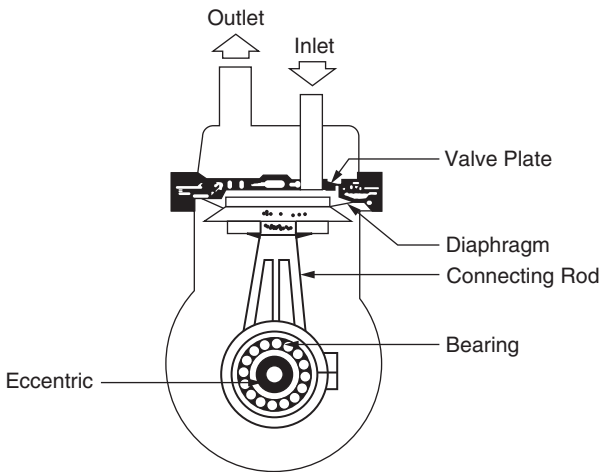
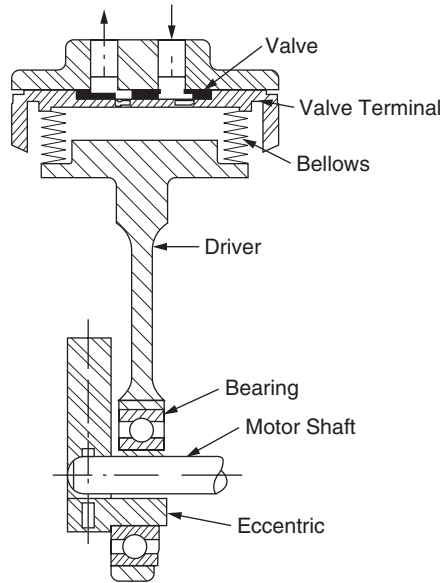


FIGURE 15.8 Diaphragm compressor. (Source: Reprinted with permission from National Resources Canada.)

FIGURE 15.9 Bellows compressor.
(Source: Reprinted with permission from National Resources Canada.)



15.4.9 Bellows Compressors

Bellows compressors (Fig. 15.9) are similar to diaphragm compressors except that the diaphragm has been replaced with metal bellows (usually stainless steel). These compressors usually have two heads that can be operated in series or in parallel, depending on the system requirements.

They have the same advantages as diaphragm compressors. In addition, they have a longer lifetime than diaphragm compressors because their flexible components (bellows) are designed to withstand a very large number of cycles without experiencing failures.

Figure 15.10 illustrates a pressure-flow characteristics of a bellows compressor. The capacity is dependent on the pressure rise across the compressor.

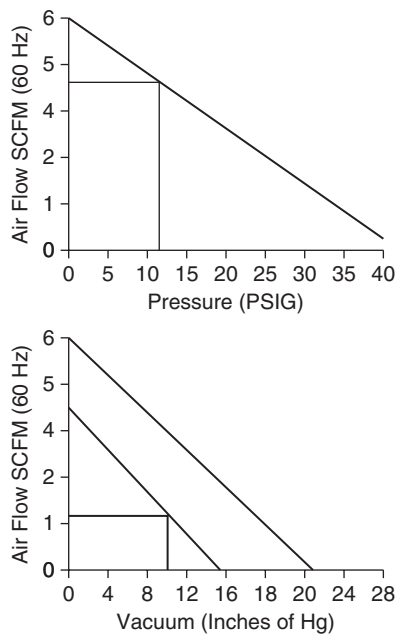
This is due to the fact that as the driver reaches the top dead center, the pressure in the head is the same as the downstream pressure. As the driver moves downward, the remaining gas has to expand until the pressure reaches the suction pressure. At this point, the inlet valve opens, and gas begins to enter the head. During the remaining portion of the downward stroke, gas is admitted into the head. As the downstream pressure increases, a larger portion of the downward stroke is spent to expand the gas until it reaches the suction pressure. The remaining portion of the downward stroke (when gas is admitted) is reduced.

15.4.10 Rotary Compressors

Certain applications that use to be dominated by reciprocating machines are now using rotary compressors, sometimes called rotaries.

Rotary compressors can be directly coupled to a motor and operated at high speeds. In general, their overall weight, size, and capital cost are less than the equivalent

FIGURE 15.10 Pressure-flow characteristics of a bellows compressor. (Source: Reprinted with permission from National Resources Canada.)



Compressor	Maximum Capacity (L/s)	Maximum Pressure [kPa _(gauge)]	Maximum Power (kW)
Rotary screw			
Small	78	860	30
Large	9400	1030	600
Lobe	10856	100	697
Vane			
Single stage	850	310	190
Double stage	2800	1030	300
Liquid ring	4720	103	746

TABLE 15.2 Common Characteristics of Rotary Compressors

reciprocating compressors. This is possible because rotaries have no intake, or outlet valves and their mechanical forces are balanced.

Their efficiency at full load is good. However, their part-load efficiency is poor because of leakage between the mating surfaces. Table 15.2 lists the common capacities, pressures, and drive sizes.

15.4.11 Rotary Screw Compressors

Figure 15.11 illustrates a rotary screw compressor with four spiral starts and lobes. The driven rotor has six starts and cavities. The lobes fits into the cavities.

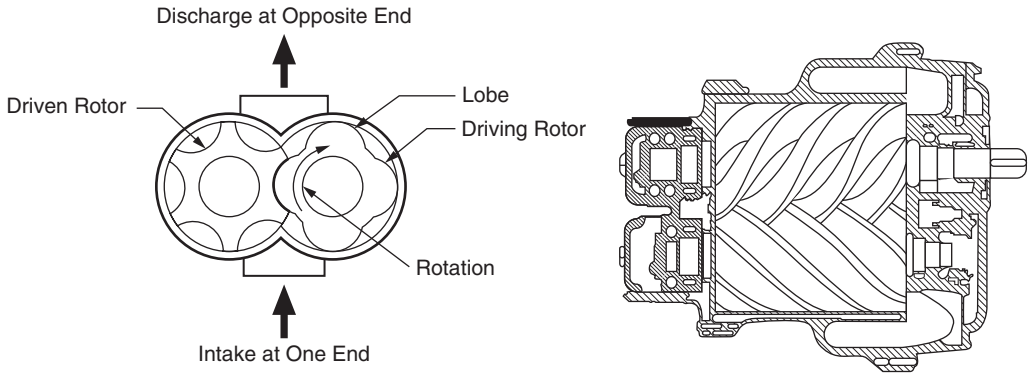


FIGURE 15.11 Rotary screw compressor.

The gas is trapped into the moving cavities between the rotor and the casing. Compression is achieved at the discharge port by the termination of the cavities. These compressors are commonly used in the capacity range of 150 to 1000 L/s. They use water or air as a coolant. They have high reliability, low overall capital and operating costs, relatively high full-load efficiency, and they can tolerate contamination in the gas.

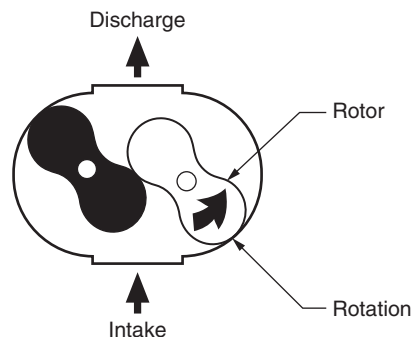
15.4.12 Lobe-Type Air Compressors

Figure 15.12 illustrates a lobe-type air compressor, sometimes called blowers. These units are used in low-pressure applications. The gear-driven rotors compress the gas to the discharge line pressure.

Their advantages are: high volumetric efficiency, low mechanical friction, and low vibration. They are suitable for flow measurements because of their positive displacement characteristics.

There is no contact between the rotors because they are gear driven. Lubrication and cooling are not required. This allows oil-free operation. These compressors are usually used for aeration, and as air pressure booster for internal combustion engines.

FIGURE 15.12 Lobe compressor.



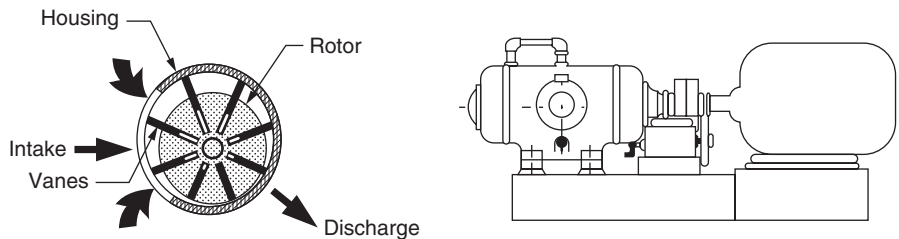


FIGURE 15.13 Sliding vane compressor.

15.4.13 Sliding Vane Compressors

Figure 15.13 illustrates a sliding vane compressor. The solid rotor is mounted eccentrically within a circular housing. When the rotor rotates, thin blades or vanes move outward from slots in the rotor to maintain a seal against the interior of the housing. Gas is drawn into the chamber due to the eccentric location of the rotor. The gas is compressed and then discharged by the changing volume between the vanes and the housing.

These units are cooled by air, water, or oil. In oil-cooled compressors, the lubricating oil is circulated to air- or water-cooled heat exchanger. Compressors rated below 1 kW can be operated oil-free with self-lubricating carbon or fibrous vanes. Their advantages are their ability to be direct driven, and they have no unbalanced forces, which results in low vibration.

Their disadvantages are low efficiency, high lubricating oil consumption, and poor single-stage capacity control. The applications of the sliding vane compressors are in the general air service and instrumentation air.

15.4.14 Liquid Ring Compressors

Figure 15.14 illustrates a liquid ring compressor. It has a multiblade rotor that is mounted eccentrically within a circular housing which is partially filled with liquid.

When the rotor spins, centrifugal force throws the liquid in the housing outward, forming a liquid ring. Due to the eccentric position of the liquid ring, the spaces in the

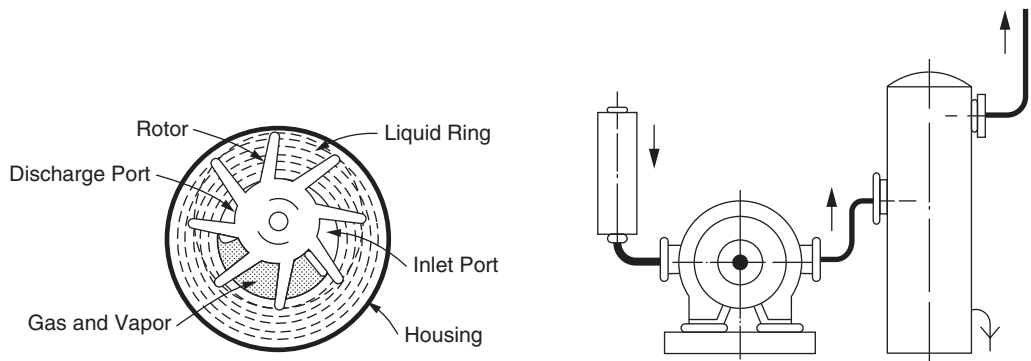


FIGURE 15.14 Liquid ring compressor.

rotor fill with liquid and then empty as the blades rotate. The continuous filling and emptying creates a pumping effect that causes suction and discharge of the gas. Their main applications are with central vacuum, and compressed air systems in hospitals, laboratories, and other industrial establishments. These machines have the ability to clean dust and bacteria from the air without adding oil and other contaminants. During compression, the liquid is recirculated outside the housing to cool the air.

The efficiency of these compressors is low and they are limited to low discharge pressure as shown in Table 15.2. They must also have all the accessories to handle the sealing liquid.

15.4.15 Dynamic Compressors

Dynamic compressors are normally used in large industrial applications and for specialized applications such as gas turbines. Their common characteristics are given in Table 15.3.

15.4.16 Centrifugal Compressors

Figure 15.15 illustrates a centrifugal compressor. The gas is thrown radially by the impellers. It passes through diffusers to convert its kinetic energy to an increase in pressure.

Compressor	Maximum Capacity (L/s)	Maximum Pressure [kPa _(gauge)]	Maximum Power (kW)
Centrifugal	71,000	1034	10,000
Axial	80,000	1034	11,000

TABLE 15.3 Common Characteristics of Dynamic Compressors

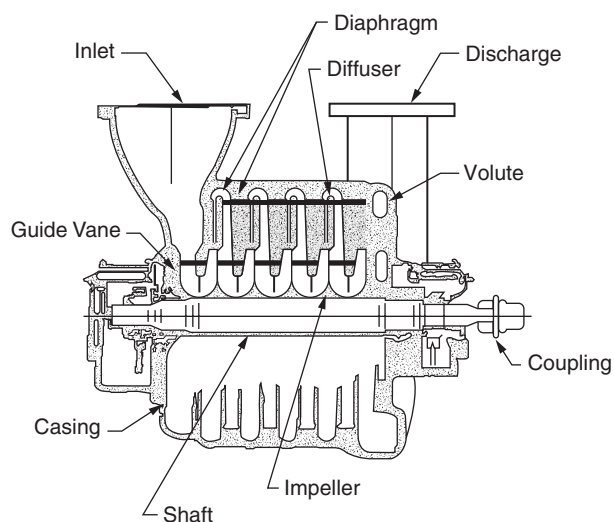


FIGURE 15.15 Centrifugal compressor.

The gas can be cooled effectively between the stages by cooling the housing. This results in near-ideal (isentropic) compression.

Except in large sizes, the overall efficiency of centrifugal compressors is usually less than that of positive displacement machines. This is mainly caused by the energy lost in the diffusers.

Their discharge pressure is stable, and they can provide a wide range of flow rate. Cooling water is circulated in the casing to cool the gas between the stages. Most units that have a discharge pressure less than 400 kPa_(g) do not require cooling.

Most commercial units operate at 20,000 rpm while speeds of 100,000 rpm are reached in the aircraft and space industries.

Their advantages become significant when the flow rate exceeds 1200 L/s. The major advantages are large capacity, low vibration, compact construction, oil-free gas discharge, and self-limiting capacity.

Their main disadvantages are the need for a speed increaser (unless turbine driven), close running clearances, and high maintenance cost.

15.4.17 Axial Compressors

Figure 15.16 illustrates an axial-flow compressor. The gas moves through a series of stationary and rotating blades.

The pressure rise through a stage (one row of stationary and one row of rotating blades) is limited due to the difficulty of cooling the gas within the casing.

The speed of an axial-flow compressor should be 25% higher than that of a centrifugal compressor to discharge gas at similar conditions. Their advantages are lower capital and operating costs for capacities higher than 65,000 L/s, and oil-free discharge air. Their main disadvantage is that, for stable operation the gas flow must be relatively constant and close to the operating range (to prevent surge). Their main applications are in gas turbines and in blast furnace air in the steel industry.

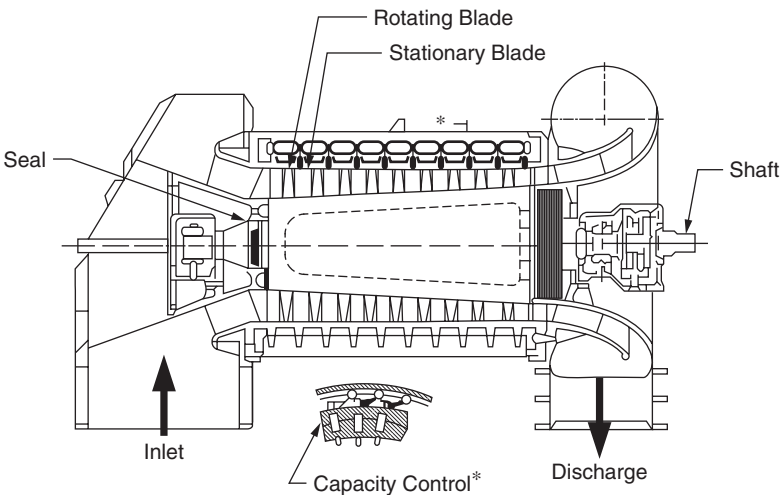


FIGURE 15.16 Axial compressor.

15.4.18 Air Receivers

Air receivers are useful in plants where the air demand is not constant and the compressor is not used at maximum capacity at all times.

The compressor can be sized for average plant load using an air receiver. The stored air supplies the peak demand load. Short cycle loading and unloading of the compressor are prevented. This allows the compressor to operate at higher efficiency for longer periods of time. The air receiver dampens out pressure surges caused by load changes, or pulsations in air lines caused by reciprocating compressors.

If the plant has long run of pipes, or if the air usage is large or irregular, more than one receiver is used. Partial cooling of the compressed air in the receiver results in water condensate and water droplets. The compressor and air receiver system illustrated in Fig. 15.16 permits the use of an inexpensive fixed capacity compressor for a system with varying air consumption. In a multiple compressor installation, an air receiver will permit the reduction in the number of compressors being operated at any one time. This results in saving in operating and peak power demand cost.

15.5 Compressor Control

One of the following methods is used to control the capacity of a compressor when its full load is not required:

- *Constant speed control* has the compressor runs continuously while the capacity of one of several *compressor unloading systems* is varied. This method is used mainly by large compressors because of the inability of large electric motors to withstand numerous starts.
- *Start-stop control* has the compressor starting and stopping by using a pressure sensing switch. Most small compressors use this method when delivering air to a receiver. Start-stop devices control some large compressors when their capacity exceeds twice the gas demand.
- *Dual control* is a combination of the previous two methods that allows selection of the appropriate method, depending on the operating conditions. The selection can be made automatically or manually.

15.6 Compressor Unloading System

The output of a compressor is reduced by the compressor unloading systems. This is done by using devices that reduce the internal volumetric capacity of the compressor. The motor control is combined with these systems to provide volume control and conservation of energy.

Suction valve unloading is illustrated in Fig. 15.17. When the demand for air is reduced, a claw cage is used to push open the plates of the suction valve.

A reciprocating compressor stage with two suction valves can be unloaded progressively to provide three-step control: full load, half load, and no load. The consumption of energy with this type of unloading is approximately proportional to the reduced load.

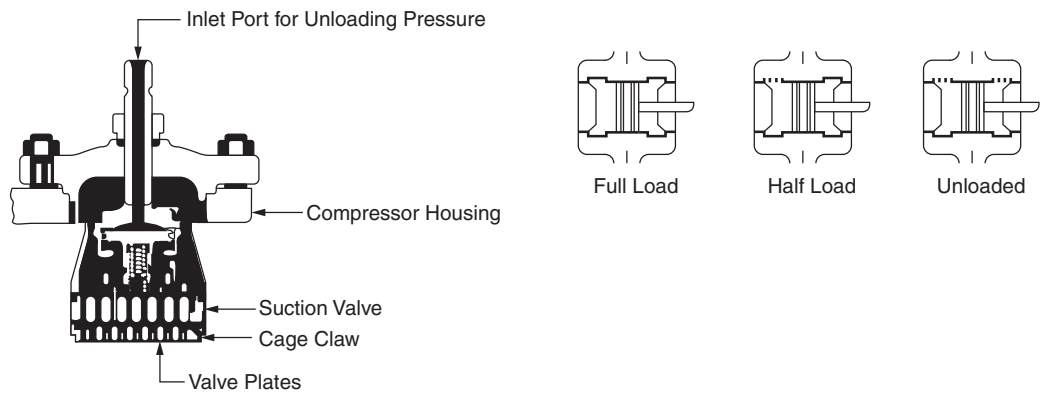


FIGURE 15.17 Suction valve unloading.

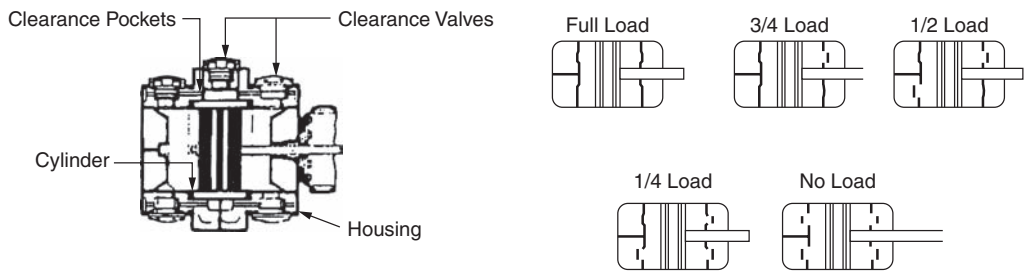


FIGURE 15.18 Clearance pocket unloading.

Compressor pocket unloading is illustrated in Fig. 15.18. The cylinders of the compressors are interconnected to clearance pockets within the housing using single or multiple valves.

When the valves are opened, the volume of the cylinder increases. This reduces the volumetric efficiency and the ability of the compressor to deliver gas. The energy consumption during this unloading method remains relatively high.

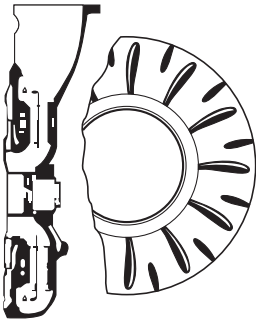
Some *inlet throttling unloading* devices are illustrated in Fig. 15.19. A continuously variable valve or inlet guide vanes are utilized on the compressor inlet. When the valve closes, the gas flow rate is reduced.

Bypass control and Blow-off control unloading systems allow the compressor to continue delivering gas, but the excess gas is vented at the discharge or bypassed back to the inlet.

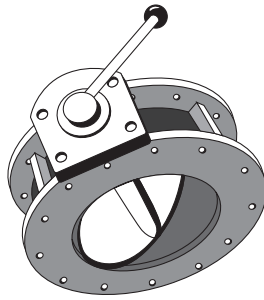
There is no reduction in the energy consumption using these systems under part-load operation. Also, more heat energy must be removed from the gas using the bypass system.

15.7 Intercooler and Aftercoolers

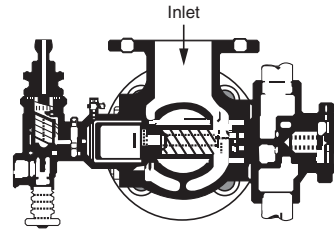
The compressor performance is improved by cooling the gas in intercoolers between the stages of multiple-stage compressors.



Adjustable Inlet Guide Vanes



Hand Operated Butterfly Valve



Inlet Unloader For Sliding Vane Compressor

FIGURE 15.19 Compressor unloading devices.

The compressed gas temperature is lowered in aftercoolers at the compressor outlet to reduce the amount of moisture in the gas (Fig. 15.4). The volumetric flow rate is decreased (due to an increase in density), resulting in reduction in pressure drop in the piping system.

The intercoolers and aftercoolers can be air- or water-cooled. Refrigeration of aftercoolers is used in some systems that require very dry compressed air.

15.8 Filters and Air Intake Screens

During service, the performance characteristics of filters and air intake screens vary significantly due to obstruction by dust and other airborne particles. Therefore, these components should be considered when evaluating the performance of a compressor system.

15.9 Preventive Maintenance and Housekeeping

Preventive maintenance and housekeeping is the term used to describe the actions that are implemented on a regular basis to reduce the failure rate and use energy wisely to reduce the compressor operating cost. The frequency of these actions depends on the system.

1. Check the gas system for leaks. This will identify sources of leaks and results in reduction in operating cost upon leak repairs. These checks must be performed more frequently if the gas is toxic.
2. Check the operation of the coolers and clean the heat exchange surfaces regularly. Fouling can occur if the gas or the service water is contaminated. Since fouling reduces the heat transfer coefficient significantly, cleaning the surfaces will decrease the operating cost. The following actions are recommended to reduce fouling:
 - a. The filters should be cleaned or replaced as specified by the manufacturer, or when the pressure drop across it exceeds the acceptable limit.
 - b. Test the cooling water for contamination.
 - c. Measure the operating parameters of the unit (flow rate, pressure increase, etc.) and compare them with the design conditions.

3. Provide the coolest possible gas to the compressor.
4. Implement a preventive maintenance program to minimize the failure of the compressor components.
5. Monitor for noise and vibration to ensure smooth and efficient operation.
6. Perform regular checks and adjustments on the drives regularly to maintain proper belt tension and coupling alignment.
7. Ensure the compressors are shut down when compressed gas is not required.
8. Clean the filters regularly to prevent excessive wear of moving components and restriction of flow.
9. Adjust the controls to operate at the lowest acceptable pressure levels and flow rates.

15.10 Bibliography

Energy Management Series for *Compressors and Turbines, Industry and Commerce and Institutions*, (14), Canadian Government Publishing Center, 1987.

CHAPTER 16

Performance of Positive Displacement Compressors

16.1 Compressor Performance

16.1.1 Positive Displacement Compressors

The amount of losses varies depending on the type of compressor. For example, a lubricated reciprocating compressor having good piston rings will have low leakage losses. However, a dry screw compressor will have significant leakage losses. These losses will be especially high if the pressure increase across the compressor is high and rotational speed low.

Some compressors are designed to operate at a specific pressure ratio ($P_{\text{discharge}}/P_{\text{suction}}$). The compressor efficiency drops when the actual pressure ratio deviates from the design pressure ratio. Other compressors use pressure actuated valves. This feature optimizes the compressor efficiency at any pressure ratio. The following sections cover the application of reciprocating compressors, however, similar considerations apply to other types of compressors.

16.1.2 Reciprocating Compressor Rating

There is a design limit for each component in the compressor. A design rating is specified for each of these components to ensure that the design limits are not exceeded during operation.

16.1.3 Reciprocating Compressor Sizing

The compressor can be selected when the following parameters are determined:

- Suction pressure
- Discharge pressure
- Suction gas temperature
- Required flow rate
- Gas composition

The selection of the compressor depends on the relative importance of the following three variables:

1. Compressor efficiency
2. Compressor system reliability
3. Compressor cost

The piston speed remains almost constant in reciprocating compressors over a wide range of applications. Thus, compressors having a short stroke run faster than those with a long stroke. Short-stroke compressors normally have a lighter construction and lower allowable loads. The highest efficiency and reliability are achieved by selecting a piston speed at the low end of the normal range. These compressors normally have a higher cost. The compressor horsepower requirement is used to determine its speed and stroke. Low-horsepower applications require a compressor having the following three characteristics:

1. Light
2. Low stroke
3. High speed

High-horsepower applications require a compressor having the following three characteristics:

1. Heavy
2. Low stroke
3. Low speed

Larger compressors are normally directly coupled to the driver whenever possible. Consideration to the following parameters should be made when selecting the number of stages:

- Allowable discharge temperature.
- Pressure ratio capability of the available cylinders as determined by their fixed clearance (the volume of the cylinder when the piston has reached the end of its travel. This term is also known as clearance volume).
- Efficiency.

The isentropic discharge temperature can be used during the preliminary sizing calculations. However, the discharge temperature should be calculated more accurately if a certain number of stages creates a marginal situation. The compressor will require additional stages if the calculated discharge temperature using one stage is too high. Intercooling is used between the stages to lower the discharge temperature. The increase in the number of stages, up to a limit, will increase the efficiency of the compressor. This is because the power requirement for a compressor stage drops when the gas inlet temperature is reduced. Figure 16.1 illustrates this fact. The area of the pressure versus volume diagram represents the work required to compress the gas ($W = -\int V_{dp}$; where W is the work required by the compressor, V is the volume, dp is an

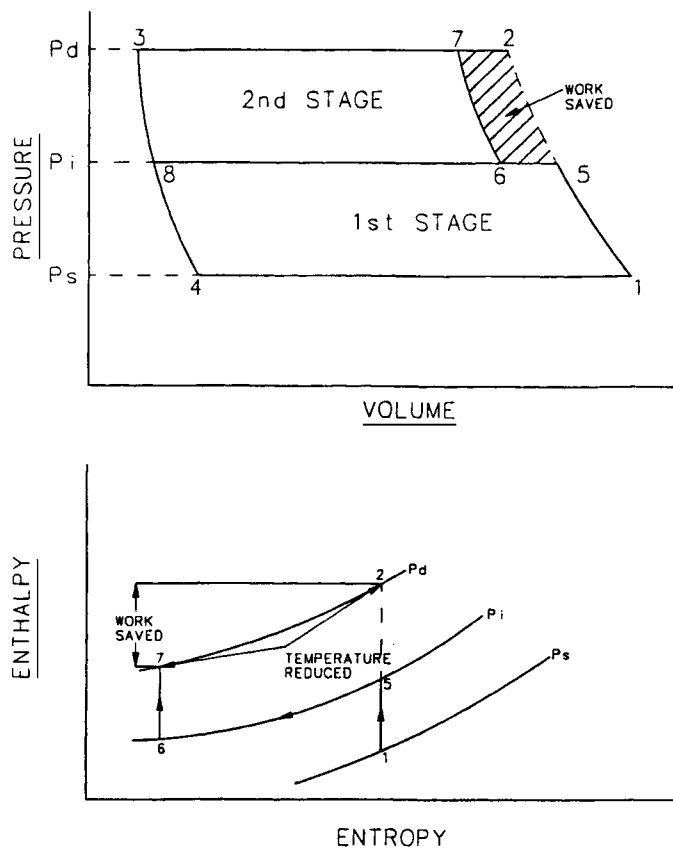


FIGURE 16.1 Effect of multistaging.

elementary change in pressure). Figure 16.1 illustrates a single-stage compression process for a given application. The diagram for this process is 1-2-3-4-1. Figure 16.1 illustrates also a two-stage compression process for the same application. The diagram for this process is 1-5-6-7-3-8-4-1. A comparison between these two diagrams shows that the two-stage compression process requires less power than the single-stage process. Intercoolers also condense out any water vapor (or other liquid vapors) that was entrained with the gas. The condensed liquids are separated from the gas in a separator. Thus, the mass of the gas compressed at the stages located downstream of the intercooler will be reduced. This causes a further reduction in the power required by the compressor. However, the number of intercooling stages should be limited. This is because the pressure losses across the valves and piping will increase with the number of stages. This will offset the gains obtained from intercooling. The cost of a compressor also increases with the number of stages.

Some applications have side streams. Gas enters or leaves the process at fixed pressures from these side streams. These requirements may help in determining the inter-stage pressures selected for the application.

The cylinders for each compressor stage can be selected after determining the number of stages required. The required cylinder bore can be estimated when the following parameters are selected:

- Inlet conditions
- Required capacity
- Speed and stroke of the piston

The cylinder is normally selected from the available designs. The following parameters must be checked before selecting the cylinders:

- The cylinder pressure rating must be adequate at the design and any upset conditions.
- The cylinder pressure rating should be higher than the setting of the relief valve.
- The frame load and rod load must be lower than the rating of these components.
- The calculated cylinder capacity when taking all losses into consideration must meet the requirements.
- The power rating per throw of the frame components must be higher than the power requirement of the cylinder.

The following factors should also be considered before finalizing the design:

- Out-of-balance force transmitted from the compressor to the foundation
- Potential for significant torsional vibration in the crankshaft and drive train
- Optimization of the layout, efficiency, and cost of the compressor

16.1.4 Capacity Control

Many applications require the reduction of the compressor capacity to meet varying process demands. The following methods are used to accomplish this task.

A simple start/stop control system is used in some applications. The compressor fills the receiver to a greater pressure than the required pressure. It is then stopped. The compressor starts again when the receiver pressure drops to the minimum value. This system causes large variations in receiver pressure. The frequent starts and stops have the following affects:

- They are stressful for the compressor and motor.
- Potential significant increase of the power cost. This is due to the frequent requirement of inrush current. This current is required when the motor is started. It is around 6 to 8 times higher than the normal operating current. This current affects the power factor of the industry adversely. The drop in power factor may lead to a significant increase in the cost of power in some applications.

The compressor capacity can also be varied using a variable-speed drive. However, this option is suitable for most applications employing positive displacement compressors. This is because these compressors do not experience a significant drop in pressure when the speed is decreased. This option is not suitable for dynamic compressors. This is because

their discharge pressure drops significantly when the speed is reduced. Unloading methods are used to give step changes in flow rate. The speed control method can be used in conjunction with an unloading method to vary the flow rate between these steps.

A bypass is also used to control the output from the compressor. This method allows a part of the discharge flow to return to the suction. This method reduces the efficiency of the compressor. The gas flowing in the bypass line may also have to be cooled. Otherwise, the increase in inlet gas temperature to the compressor will reduce its efficiency further. However, this method is simple, reliable, and inexpensive. It is suitable for unloading the compressor during start-up and shutdown. The flow through the compressor can also be adjusted by throttling the suction valve. This method reduces the efficiency of the compressor. It should be noted that this method cannot be used with dynamic compressors. These compressors could face the following problems if their inlet flow is reduced:

- Surge problems (this phenomenon will be described in detail in the chapters dealing with dynamic compressors. It causes flow reversal, significant vibrations, and overheating in the compressor.)
- A reduction in compressor efficiency

Positive displacement compressors can be unloaded. The variation of their efficiency is minimal when they are unloaded. Some screw compressors use slides to change the inlet port timing. The compressor flow can be varied within a wide range using this method.

Unloaders are used with reciprocating compressors. They provide a single cylinder per stage, double-acting compressor with a three-step control. The compressor can be operated at 0%, 50%, or 100% capacity. Some compressors have two identical double-acting cylinders per stage. Five-step control can be accomplished if each stage has two identical double-acting cylinders. These compressors can operate at 0%, 25%, 50%, 75%, or 100% capacity. Additional steps can be arranged if the cylinders have different clearance volumes (at the ends). The first stage of a multistage compressor normally controls the capacity. The inter stage pressures will change significantly if only the first stage is unloaded. The discharge pressure of a higher-stage cylinder will probably exceed its design limit. All stages should normally be unloaded together. There could also be limitations on the time period when a cylinder is operated completely unloaded. This is due to the heat build up that occurs in the cylinder when it operates in this mode. There could also be problems with the build up of lubricating oil or process liquids in the cylinder when the unit operates completely unloaded.

There are three common methods to unload a cylinder. They all involve connecting the cylinder to the suction passage. This allows the gas to flow back and forth between the cylinder and the suction passage. The gas flowing in this line will not be compressed. The cross-sectional area of this connection should be adequate. Otherwise, the frictional losses will be high. The cylinder will overheat due to the increase in frictional losses. The compressor efficiency will also drop due to the increase in the power required to compress the gas. Unloaders are used to make a step-change in load while the compressor is operating. They prevent the cylinder from compressing the gas by maintaining an open flow path between the cylinder bore and its suction chamber. The unloader allows the gas to be admitted into the cylinder. However, they allow the cylinder to pump the gas which is still at suction pressure back into the suction chamber

through this open pathway. The compressor will only require enough power to overcome the friction losses in this mode of operation. The following are the types of unloaders:

- Finger unloaders (valve depressors)
- Plug or port unloaders

Figures 16.2 and 16.3 illustrate finger-type unloaders. They consist of a series of small fingers that are housed in the valve. They are actuated by a push rod from an outside actuator. The fingers are lowered to unload the valve. They depress the valve sealing components. This keeps the valve in the open position. A pathway is now established between the cylinder bore and the gas passage through these open suction valves. Finger-type unloaders are normally mounted on each suction valve. This maximizes the flowpath of the unloaded pathway. Finger-type unloaders can be actuated manually using a handwheel and a screw or lever arrangement. They can also be actuated pneumatically using a small air cylinder located on top of the unloader stem. The main problem with this type of unloader is the potential for damaging the valve sealing elements with the fingers. This could lead to premature valve failure.

Figures 16.4 and 16.5 illustrate plug-type unloaders. These unloaders have a passageway bored in the middle of the valve. The plug seals the passageway during normal operation. The cylinder is unloaded by removing this plug. This allows the gas to flow in and out through this passageway. The gas will not be compressed in this configuration. Plug-type unloaders have an advantage over finger-type unloaders. This is because plug-type unloaders do not act on the valve sealing components like finger-type unloaders. However, the normal effective flow area of the valve is reduced due to the presence of a passageway through the centre of the valve. This will increase the horsepower requirement of the compressor due to the increase in

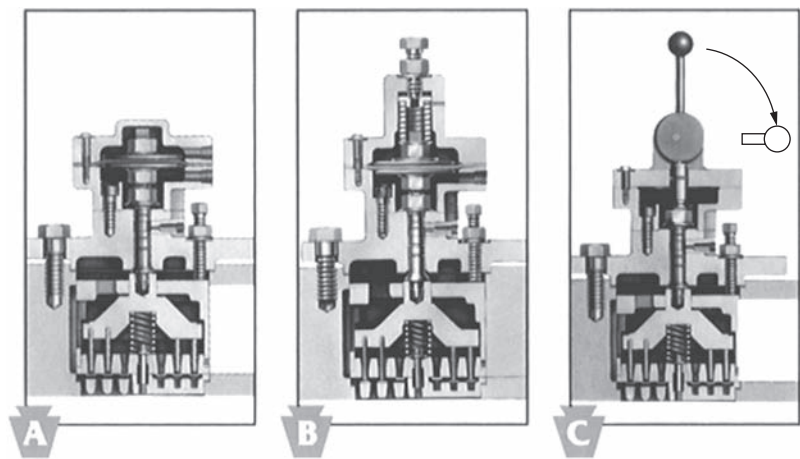
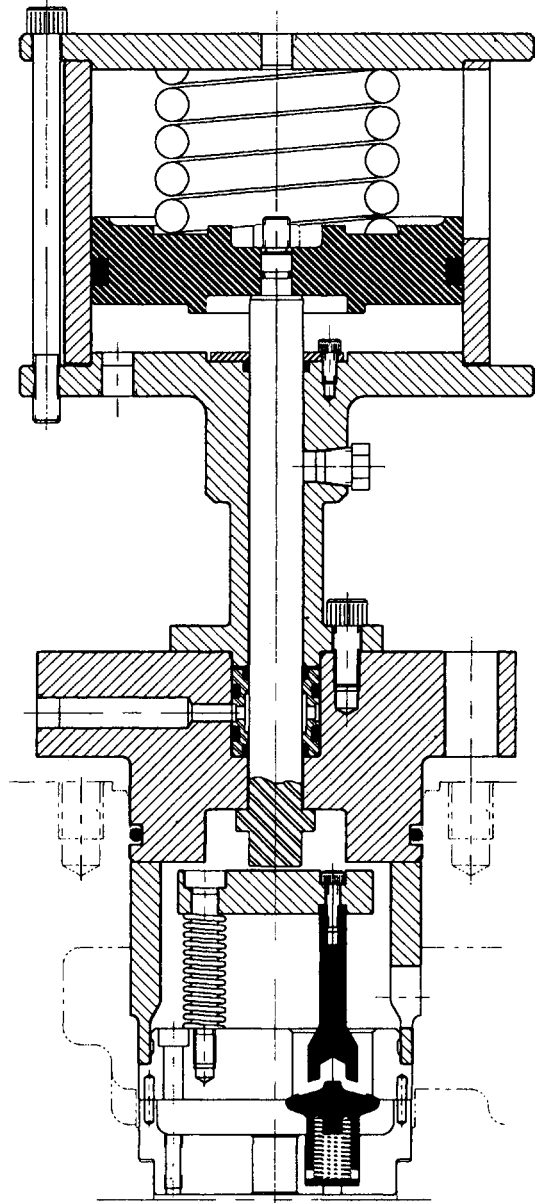


FIGURE 16.2 Finger-type unloaders, pneumatically operated: (a) direct-acting (air-to-unload); (b) reverse-acting or fail-safe (air-to-unload) which automatically unloads the compressor in the event of control air failure; (c) manual operation. (Source: Reprinted with permission from Cameron Corporation, Texas.)

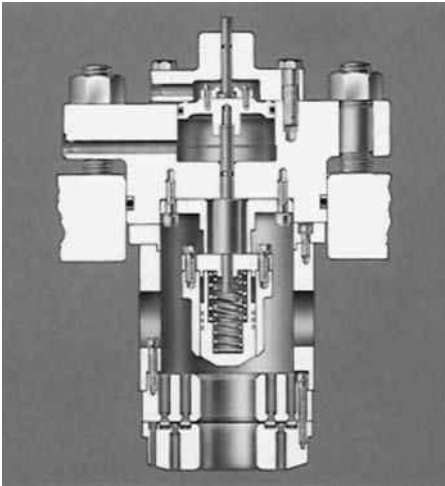
FIGURE 16.3 Finger-type unloader.



pressure drop across the valve. A plug-type unloader is normally used with each suction valve.

Figures 16.6 and 16.7 illustrate port-type unloaders. These unloaders are also known as passage-type unloaders. They also use a plug to seal the unloader passageway. However, this type of unloader does not use a passageway through the valve like the plug-type unloaders. Port-type unloaders use a separate port between the cylinder bore and the gas passage. This port is created by removing one suction valve from each

FIGURE 16.4 Outside operated plug-type unloader. Actuating air cannot mix with the gas being compressed. (Source: Dresser-Rand Company, Painted Post, N.Y.)



cylinder end. The suction valve is replaced with a large plug assembly. Port-type unloaders have several advantages. The flow area of the passageway is the same as the flow area of a normal valve. The plug will normally lift 2.54 to 5.08 cm (1–2 in) off its seat. Since only one unloader is required for each cylinder end, the compressor uses fewer components. The removal of these unloaders is not required for regular valve maintenance. This is because this type of unloaders does not work on an active valve. Applications involving low-molecular-weight gases use this type of unloader normally. This is because the pressure drop across the active valves is reduced due to the reduction in the total number of suction valves.

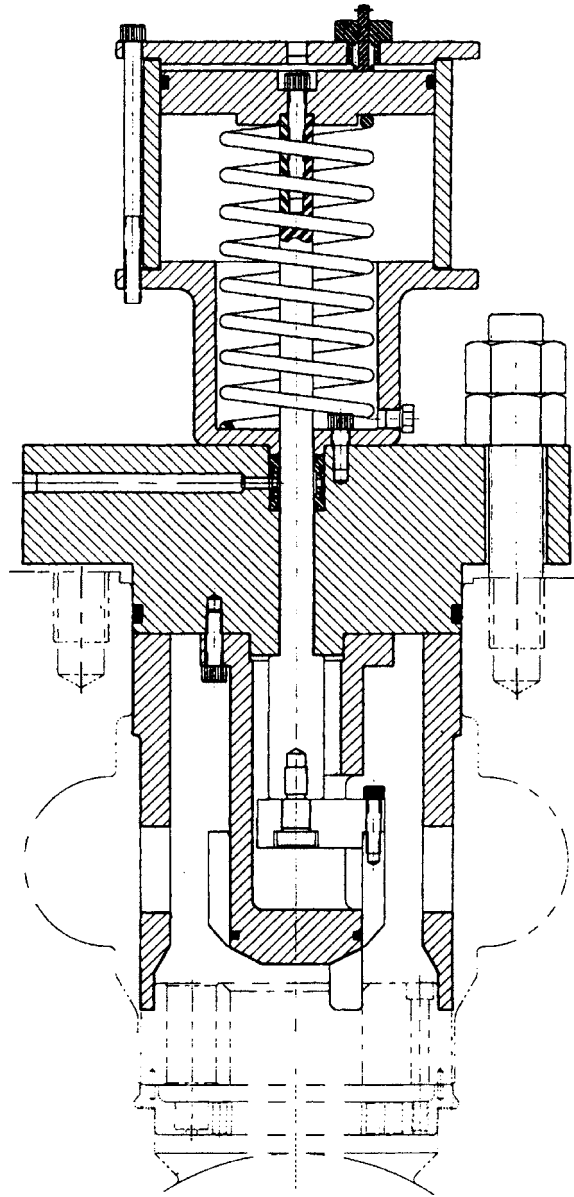
Suction valve unloaders control the capacity in discrete steps. However, some manufacturers (e.g., Hoerbiger compressor controls etc.) provide step-less capacity control systems. These systems use pneumatically actuated finger-type unloaders. They unload the valve for only a portion of the stroke (conventional unloaders keep the valve unloaded continuously). This feature provides partial flow (conventional unloaders provide full flow or no flow for a fully unloaded cylinder). A specially designed control panel actuate these unloaders. This control panel monitors the flow requirements of the process. It unloads the compressor as necessary.

Overheating will occur in the inlet passage and cylinders when the compressor is operated unloaded for an extended period of time. This problem can be prevented by operating the compressor fully loaded periodically. This allows the heated gas to be pumped to the process. The flow of cooler inlet gas will normalize the temperatures. This cyclic loading should be done for 10 minutes out of every hour of unloaded operation. This overheating problem does not occur when the compressor operates at 50% loads.

Clearance Pockets

Figures 16.8 and 16.9 illustrate clearance pockets. They are commonly used to reduce the capacity of the compressor. This is achieved by connecting the cylinder end to an additional volume. The cylinder will compress a reduced amount of gas when it is in this configuration. Clearance pockets are controlled manually or pneumatically. Figure 16.10

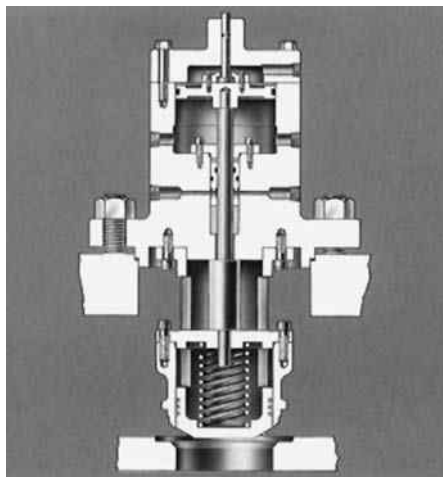
FIGURE 16.5 Plug-type unloader.



illustrates a variable-volume clearance pocket. This allows the compressor to deliver any capacity within the range of the unloader. The required volume of the clearance pockets depend on the following parameters:

- Flow required
- Cylinder size
- Pressure ratio ($P_{\text{discharge}}/P_{\text{inlet}}$)

FIGURE 16.6 An outside-operated port-type unloader requires the use of only one unloading device per cylinder end and is typically used for lower-molecular-weight gases. Air cannot mix with the gas being compressed. (Source: Dresser-Rand Company, Painted Post, N.Y.)



Applications having low-pressure ratios require a very large pocket to give a slight reduction in capacity. Large cylinders can have several fixed-volume clearance pockets. This allows the compressor to deliver different flow rates. The compressor operation should be checked carefully at every possible unloaded condition. This is required to confirm that all cylinders are operating within design limits. The rod loads and discharge pressures must be within the acceptable ranges.

16.1.5 Compressor Performance

The compressor performance depends on several parameters. These parameters are discussed in the following sections with reference to reciprocating compressors. They are then discussed with reference to screw compressors. However, the losses that occur in other types of positive displacement compressors are similar to those discussed in this chapter.

16.2 Reciprocating Compressors

16.2.1 Compressor Valves

Reciprocating compressor valves have a significant effect on the efficiency (horsepower and capacity) and reliability of the compressor. These valves are simply check valves. They must meet the following requirements:

- They must operate reliably for a billion cycles.
- Their opening and closing times must be within milliseconds.
- They must not leak in the reverse flow direction.
- The pressure drop across them must be low.
- They must operate adequately in highly corrosive, dirty gas environment and while covered with sticky deposits.

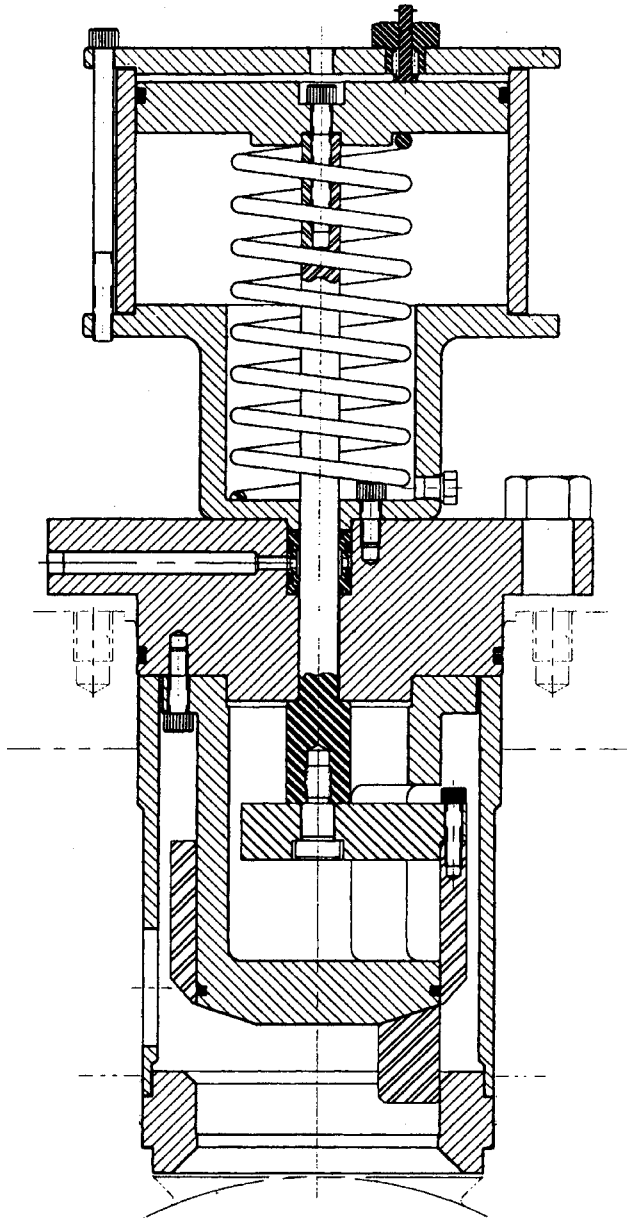


FIGURE 16.7 Port-type unloader.

Compressor valves have a significant effect on performance due to the following reasons:

- The pressure drop across the valves.
- The leakage from the compressor through the valves.
- The valve may not operate exactly like an ideal valve.

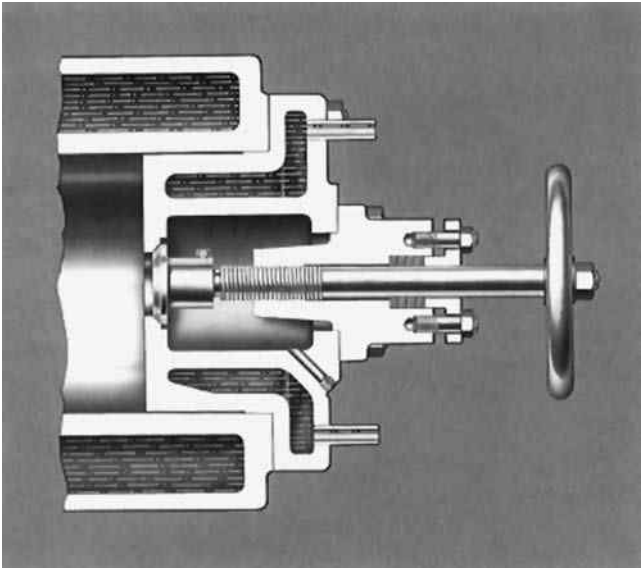


FIGURE 16.8 Manual fixed-clearance pocket valve is generally located in the outer head of a cylinder, as shown. This type of control is used for applications that require limited and infrequent capacity changes. (Source: Dresser-Rand Company, Painted Post, N.Y.)

Figure 16.11 illustrates a typical valve dynamics diagram. The following comments can be made about the diagram:

- The valve does not open instantaneously. This is due to its inertia.
- The valve does not stay at full lift during the period when it is open. This is due to the springing effect.
- The valve does not close exactly at the dead center.

All these factors have a significant effect on the capacity and power requirement of the compressor. These valves should be inspected and replaced regularly to prevent deterioration of the compressor performance.

16.2.2 Reciprocating Compressors Leakage

All compressors use sliding seals between the high- and low-pressure zones. There is always a leakage through these seals. This leakage reduces the efficiency of the compressor. Reciprocating compressors have the following usual leakage paths:

- Through the piston rings
- The rod packing of double-acting compressors
- Valves that do not provide adequate seals against reverse flow

These seals should be maintained in the best possible condition to reduce the drop in compressor efficiency.

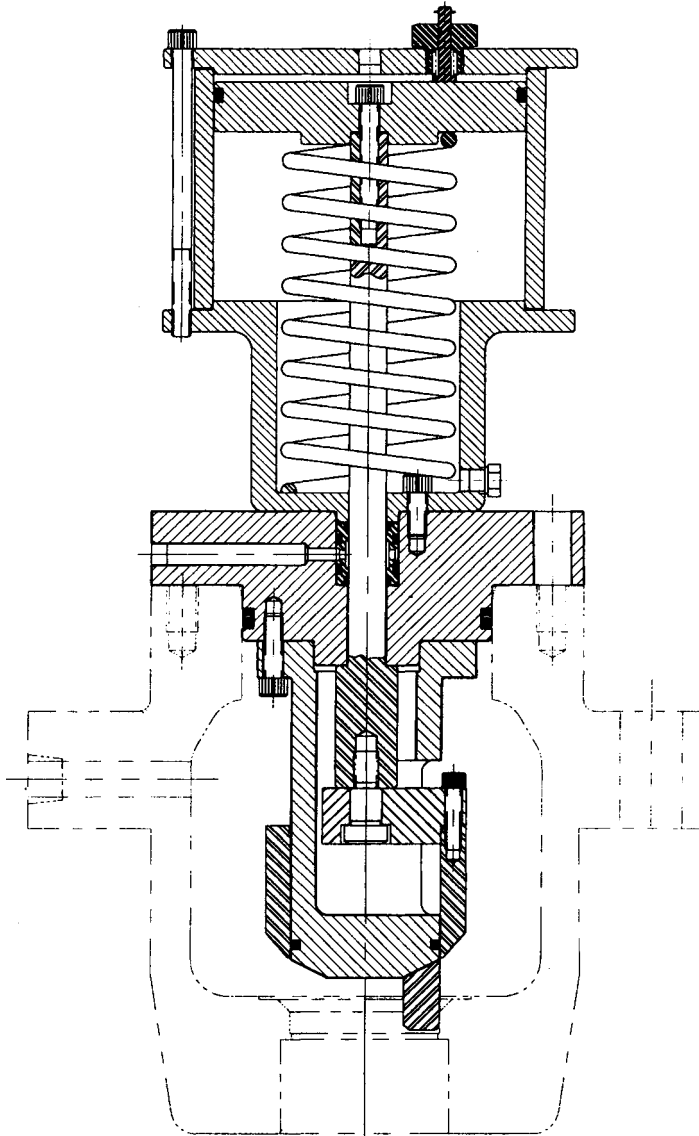


FIGURE 16.9 Clearance pocket.

16.2.3 Screw Compressors Leakage

Screw compressors have different leakage paths than reciprocating compressors. Screw compressor sealing relies only on tight running clearances. The main leakage paths in screw compressors occur at the following locations:

- Between the meshing rotors
- The clearances between the rotors and the compressor casing

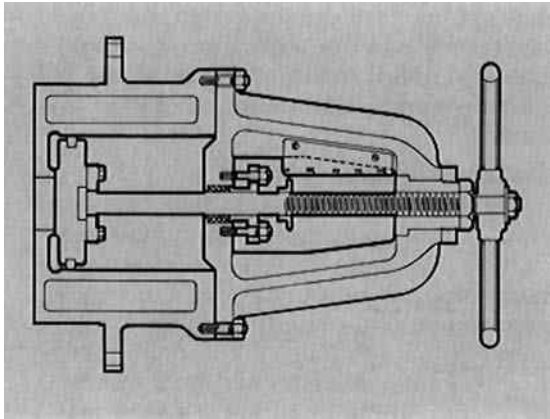


FIGURE 16.10 Manually controlled variable-volume clearance pocket. This clearance pocket provides capacity reduction in an infinite number of steps over a given range. This pocket can also be automatically actuated by a hydraulic system that varies the position of the piston in the pocket. (*Source:* Dresser-Rand Company, Painted Post, N.Y.)

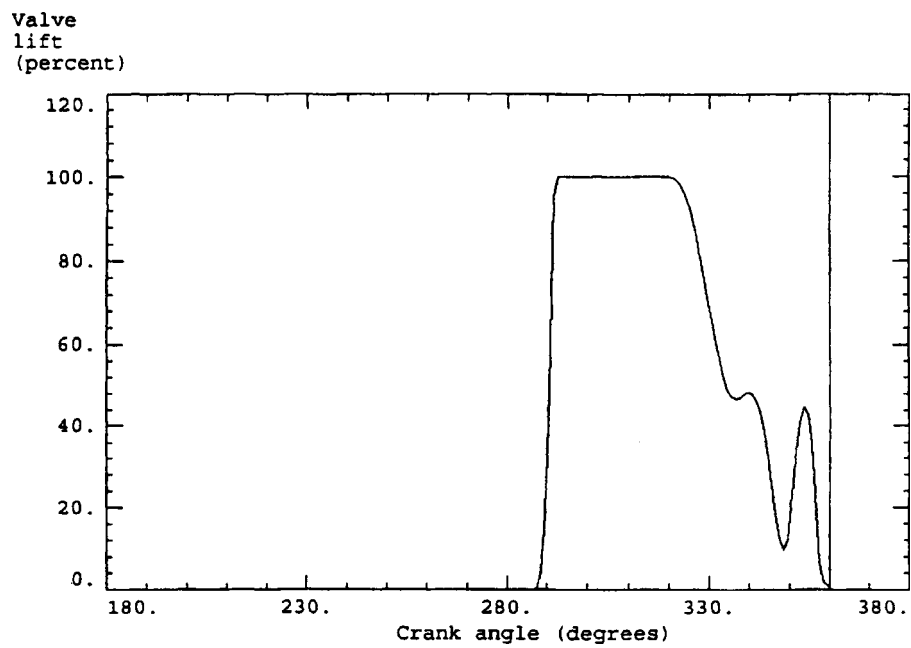


FIGURE 16.11 Typical valve dynamics diagram.

The sealing effect of the liquid in an oil- or water-flooded compressor reduces the leakage. The drop in efficiency should be monitored carefully during operation. The compressor should be refurbished at the earliest opportunity if the drop in efficiency becomes significant.

16.3 Bibliography

- Block, H. P., *A Practical Guide to Compressor Technology*, 2nd ed., John Wiley & Sons Inc., Hoboken, New Jersey, 2006.
- Hanlon, P. C., *Compressor Handbook*, McGraw-Hill, New York, 2001.

This page intentionally left blank

Reciprocating Compressors

17.1 Introduction

Reciprocating compressors are very efficient and reliable. They compress any gas mixture from vacuum to 300 MPa. The power ratings of these compressors vary up to 18 MW. Their capacity varies up to 35,000 m³/h at compressor inlet conditions. Reciprocating compressors are also capable of compressing a wide range of gas densities. They are used for compressing hydrogen which has a molecular weight of 2. They are also used for compressing heavier gases such as chlorine. This gas has a molecular weight of 70.

The pressure ratio per stage of a reciprocating compressor can vary from 1.1 to 5. However, the typical compression ratio per stage is around 3. This is done to limit the discharge temperature to around 150 to 175°C (300–350°F). Some reciprocating compressors have six stages. These compressors provide a total compression ratio over 300.

Most reciprocating compressors operate continuously for many years. They are shutdown occasionally for maintenance. Problems occur in some applications due to the following:

1. The gas flowing through the compressor is corrosive.
2. Liquid and/or foreign abrasive particles are entrained with the gas.

These compressors operate within the following three ranges:

1. Rotational speed from 275 to 600 rpm
2. Piston speed from 3 to 5 m/s (600–1000 ft/min)
3. Compressor stroke from 150 to 460 mm (6–18 in)

Longer strokes and slower speeds are used for compressors rated for higher power. Lower rotational and piston speeds are used in non-lubricated applications. This is done to increase the life of the piston and packing rings.

Figures 17.1 through 17.3 illustrate balanced opposed reciprocating compressors. This is the most commonly used type today. The unbalanced forces and moments are minimized in this design. This maximizes the operating life of larger units. They normally employ 2 to 10 cylinders. All reciprocating and rotating weights are almost balanced in these compressors. Single-cylinder units can also be designed with opposed balanced weight crossheads. Figures 17.4 through 17.8 illustrate Y-shaped and similar vertically arranged reciprocating compressors.

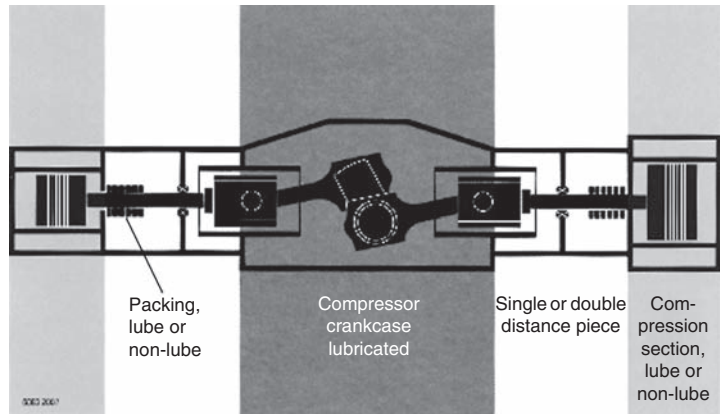


FIGURE 17.1 Principle of a balanced-opposed reciprocating process compressor. (Source: Reprinted with permissions from MAN Diesel & Turbo SE.)

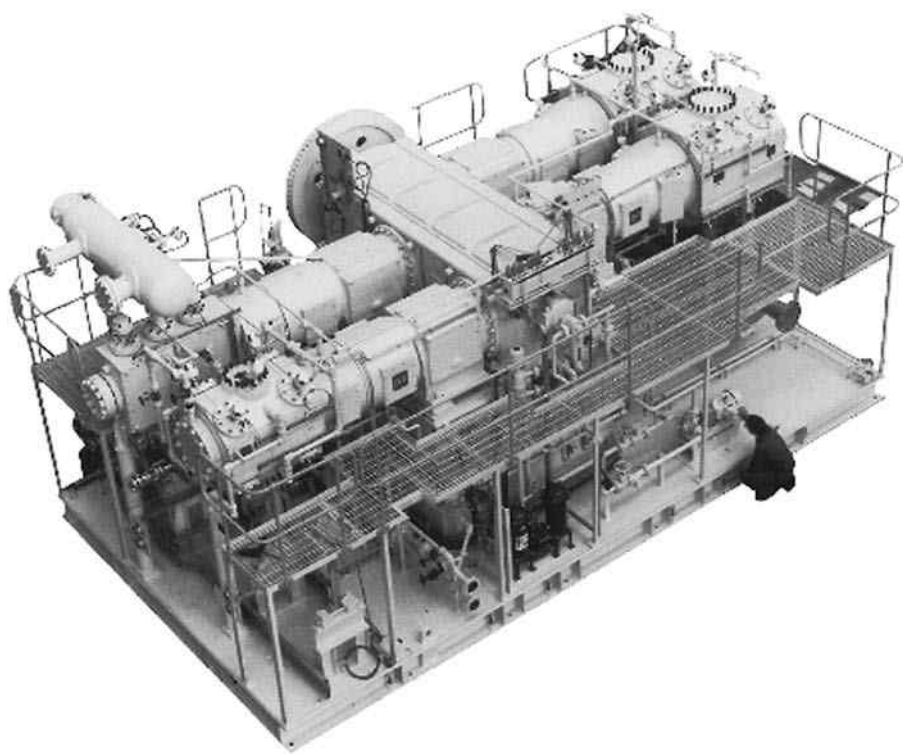


FIGURE 17.2 Balanced-opposed reciprocating compressor package. (Source: Dresser-Rand Company, Painted Post, N.Y.)

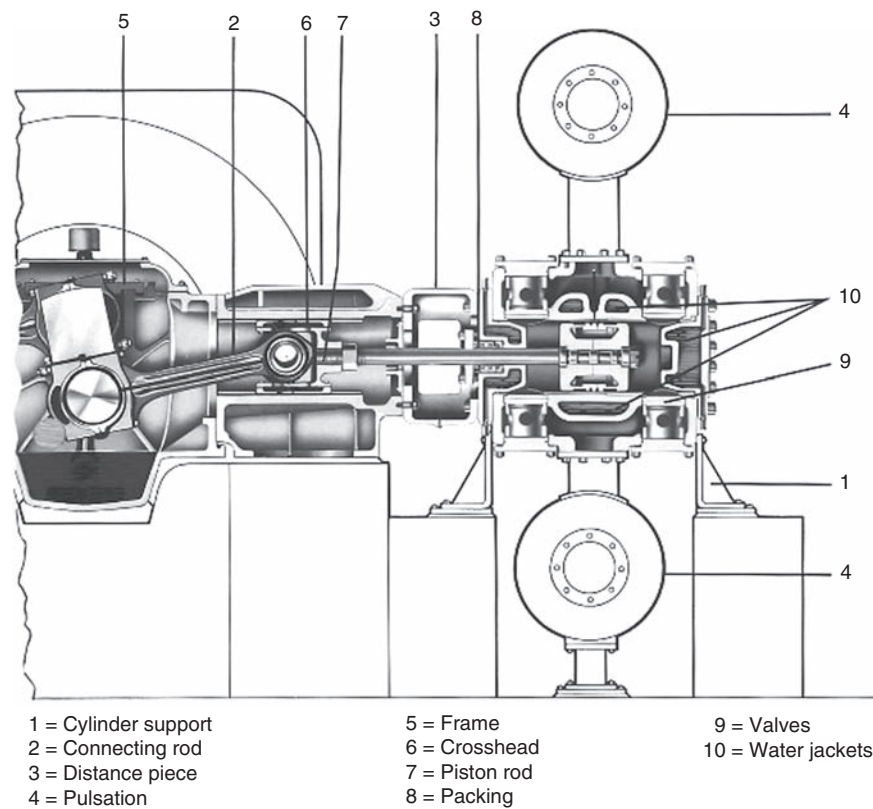


FIGURE 17.3 Right-hand portion of a balanced-opposed reciprocating compressor. (Source: Dresser-Rand Company, Painted Post, N.Y.)

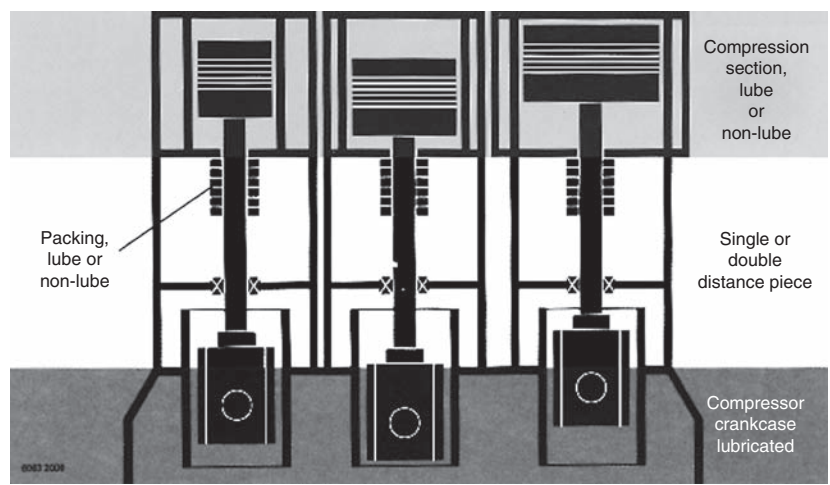


FIGURE 17.4 Vertically arranged compressor cylinders. (Source: Dresser-Rand Company, Painted Post, N.Y.)

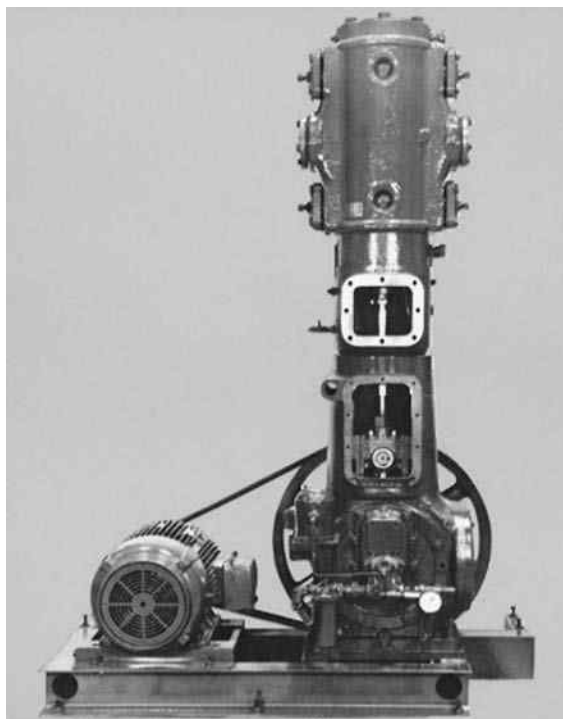


FIGURE 17.5 Vertically oriented reciprocating compressor. (Source: Reprinted with permissions from Cameron Corporation, Houston, TX.)

17.2 Crankshaft Design

Some large reciprocating compressors have up to 10 crankthrows. The cranks are arranged to have equal angles between them (see Fig. 17.9). This is done to optimize the unbalanced forces (see Fig. 17.10). Even-number crankthrow units are arranged to have 180° opposed pairs of cranks. This is done to cancel out inertia forces. Odd-number crankthrow units use dummy crossheads in some applications to balance the forces (Fig. 17.11).

The movement of the pistons creates pulsating compression forces and vibratory torque on the shaft. The peaks of this torque can exceed the average compressor horsepower torque by up to 5 times. The crankshaft must be designed to withstand these vibrating stresses. It is made from forged steel in compressors rated over 150 kW per crank.

The American Petroleum Institute specification (API 618) requires the following:

- The crankshafts should be made from forged steel.
- The crankshafts should be heat-treated with ground-bearing surfaces.

Some experienced manufacturers further require the cranks to be upset forged from the steel billet. This is done to provide stronger grain flow through the crank

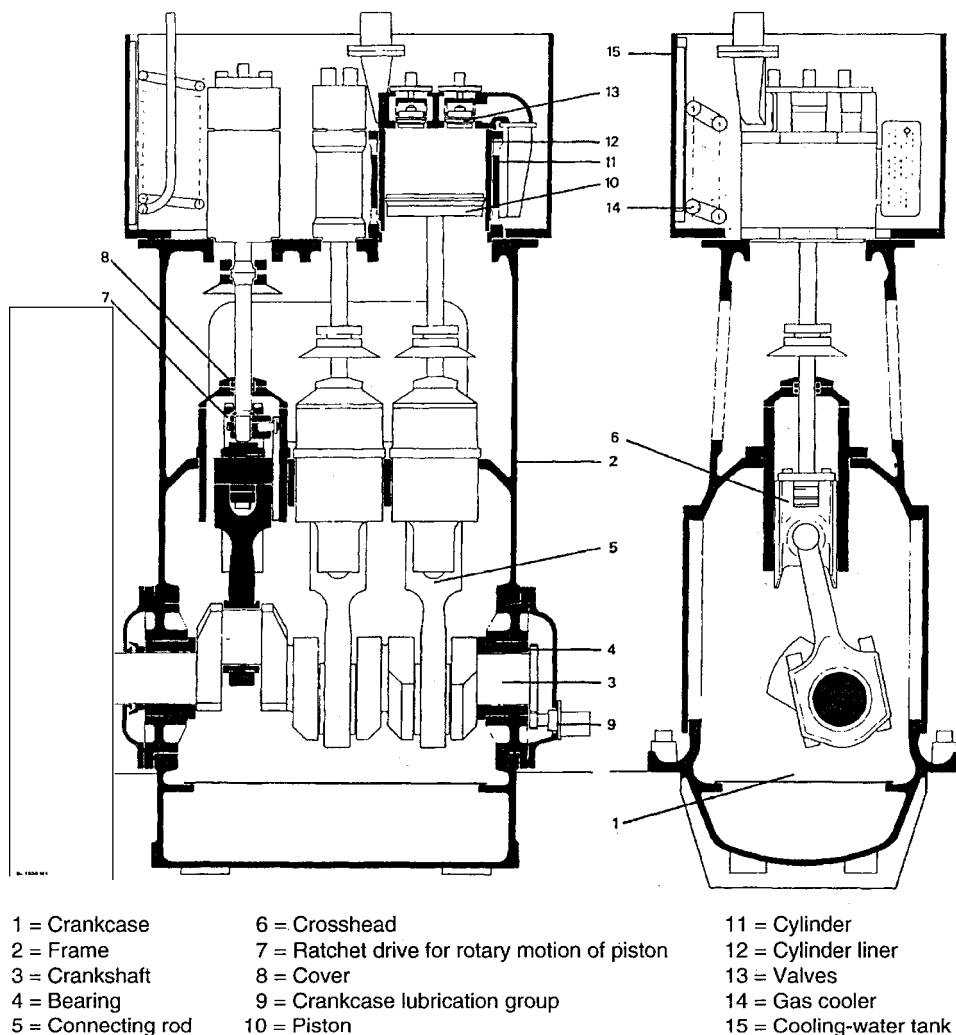


FIGURE 17.6 Sectional drawing of a water-lubricated vertical reciprocating compressor used in oxygen service. (Source: Reprinted with permissions from MAN Diesel & Turbo SE.)

webs. This technique has proved to produce stronger cranks than machining them from a billet.

The materials used to manufacture the cranks are alloy steel AISI 1045 or AISI 4140. The crankshafts (Fig. 17.12) are ultrasonically inspected by the supplier. They are also purchased completely finished from the supplier. Special facilities at the supplier are used to upset forge and grind the journals and crankpins. Oil passages (Fig. 17.13) are drilled in the crankshaft to allow the oil flow from the journals to the crankpins. Stress concentration points are prevented at the intersection of the oil holes. This is done by radiusing and polishing these holes. Bolted-on or integral counterweights are mounted on the crankshaft to offset the unbalanced forces and moments.

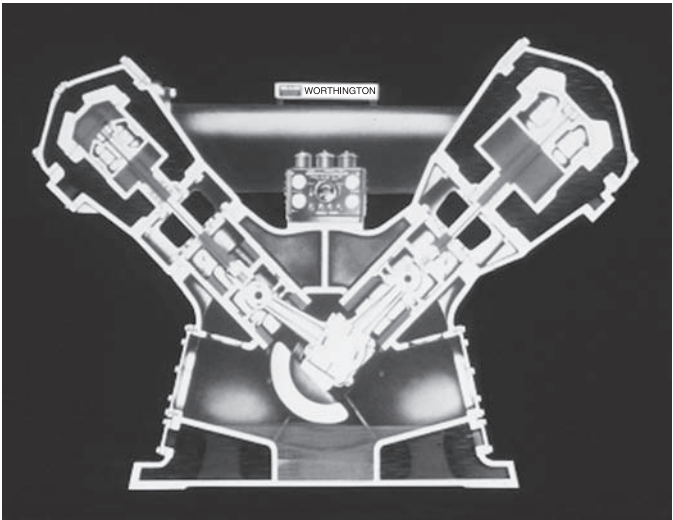
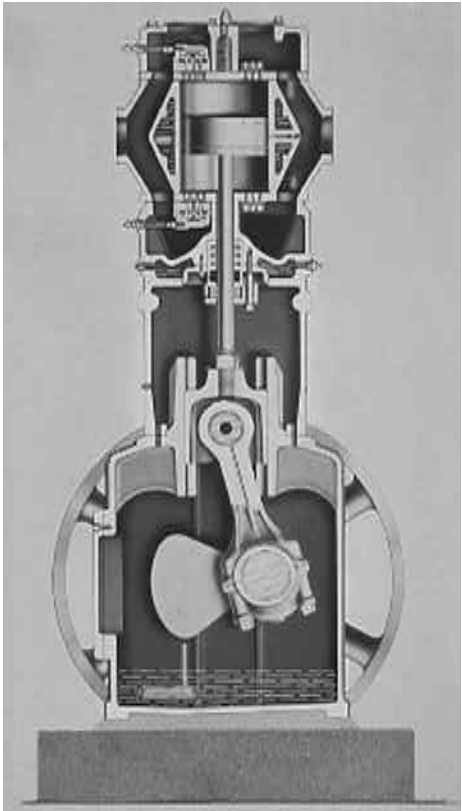


FIGURE 17.7 Reciprocating compressor with Y arrangement of cylinders. (Source: Dresser-Rand Company, Painted Post, N.Y.)

FIGURE 17.8
Reciprocating
compressor with
vertically arranged
cylinders. (Source:
Dresser-Rand Company,
Painted Post, N.Y.)



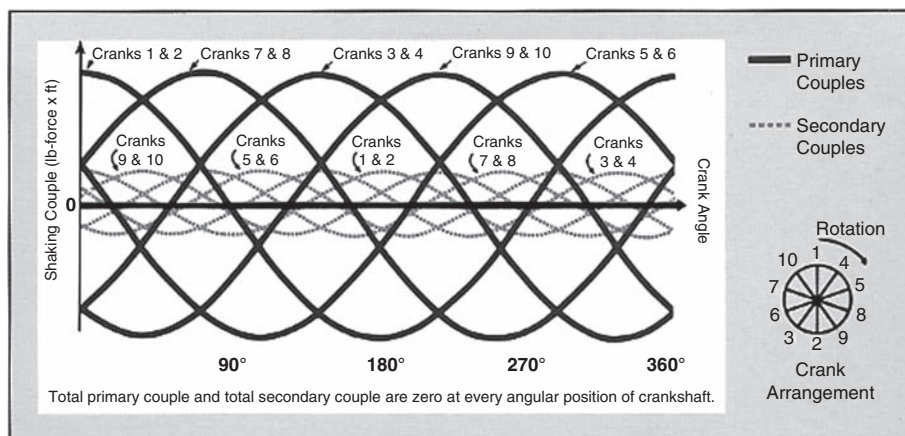


FIGURE 17.9 Preferred crank arrangement and resulting couples for a 10-throw compressor. Pairs of cranks are uniformly displayed at 36° . Therefore, reciprocating and rotating weights in opposing cylinders will not normally be equal. A variety of techniques are used to add weights to reciprocating parts to achieve the desired balance. (Source: Dresser-Rand Company, Painted Post, N.Y.)

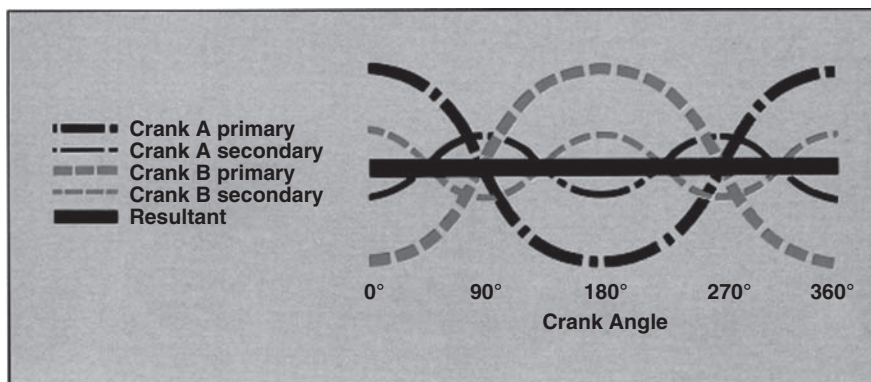


FIGURE 17.10 Crank-angle diagram demonstrating how primary and secondary forces balance each other out by acting in opposite directions. (Source: Dresser-Rand Company, Painted Post, N.Y.)

17.3 Bearings and Lubrication Systems

Most reciprocating compressors have replaceable precision-bored sleeve-type aluminum alloy crankpin and bearings. These components do not require field fit-up or adjustments. The bearing load capability of aluminum alloy is high. This material will not likely score the crankshaft surface when a bearing failure occurs. The following materials are also used in bearings:

- Steel-backed aluminum
- Steel or bronze-backed
- Babbitt-lined
- Tri-metal (steel-bronze-babbitt)

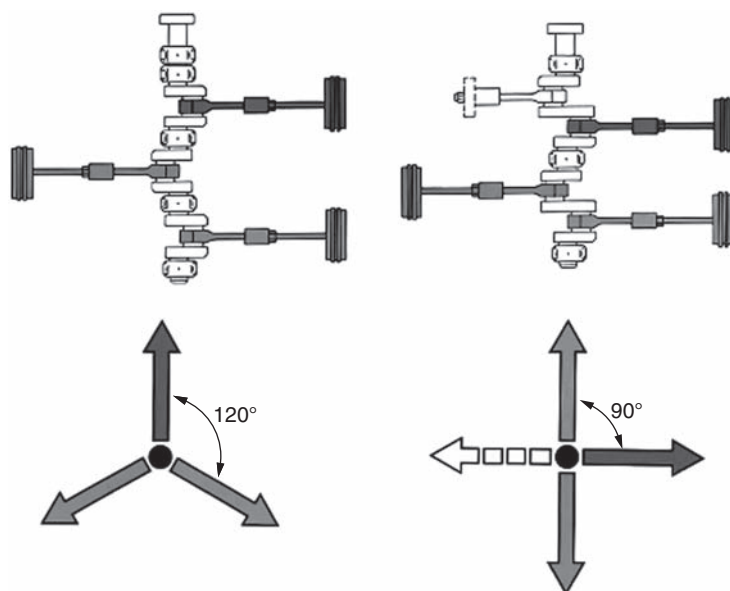


FIGURE 17.11 Three-throw crank angles at 120° (left) vs. 90° (right). Note the dummy crosshead required on the right. (Source: Dresser-Rand Company, Painted Post, N.Y.)



FIGURE 17.12 Two-throw crankshaft. (Source: Dresser-Rand Company, Painted Post, N.Y.)

The oil piping and filters should be maintained clean to increase the life of the bearing systems and the reliability of the compressor. The oil should also be maintained at the pressure and temperature recommended by the supplier.

Figure 17.14 illustrates the force-feed lubrication system used for reciprocating compressors. It includes an oil pump, oil cooler, and oil filter. The criticality of the process governs the redundancy and instrumentation requirements of this system. The API specification API 618 provides the available options of this system.

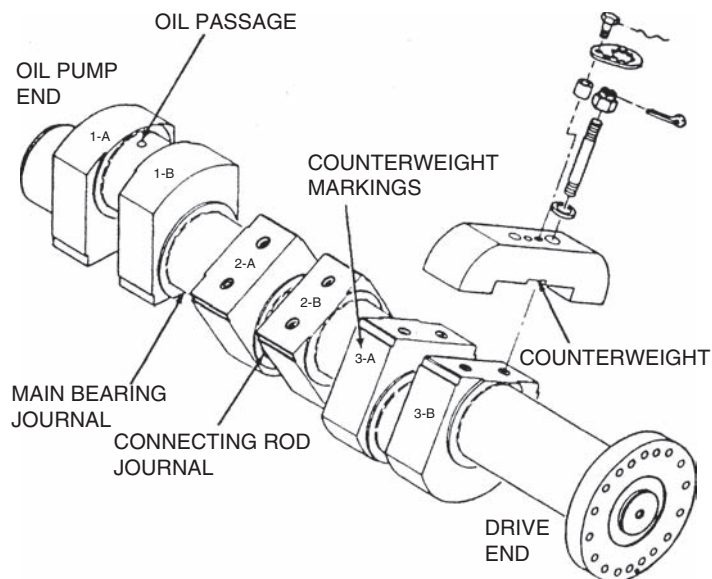


FIGURE 17.13 Three-throw crankshaft showing oil passages and counterweights. (Source: Dresser-Rand Company, Painted Post, N.Y.)

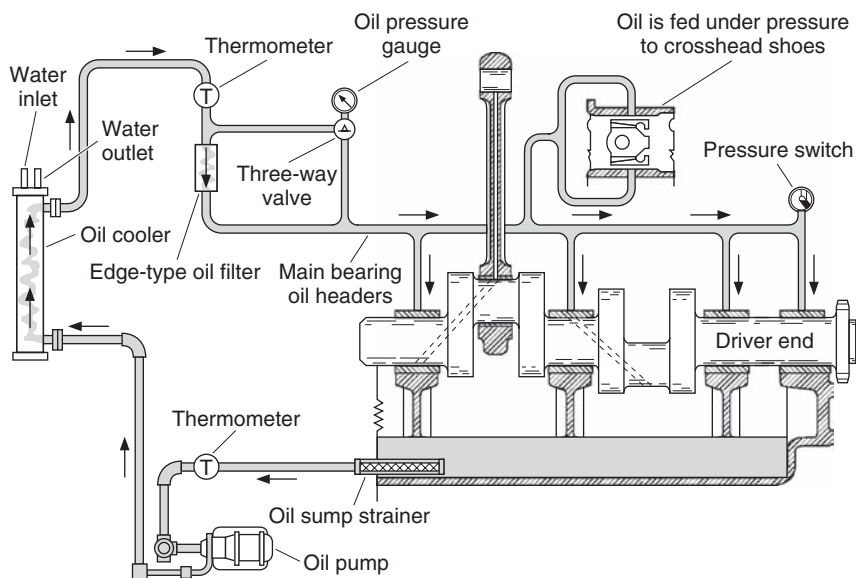


FIGURE 17.14 Force-feed lubrication system for reciprocating compressor. (Source: Dresser-Rand Company, Painted Post, N.Y.)

17.4 Connecting Rods

Figures 17.15 and 17.16 illustrate connecting rods used in reciprocating compressors. They are made from forged steel. They are also manufactured with a closed die to provide good grain flow throughout the material. Forced lubrication oil passages are drilled along the length of the rod. They permit the oil flow from the crankpin to the



FIGURE 17.15 Typical connecting rod. (Source: Dresser-Rand Company, Painted Post, N.Y.)

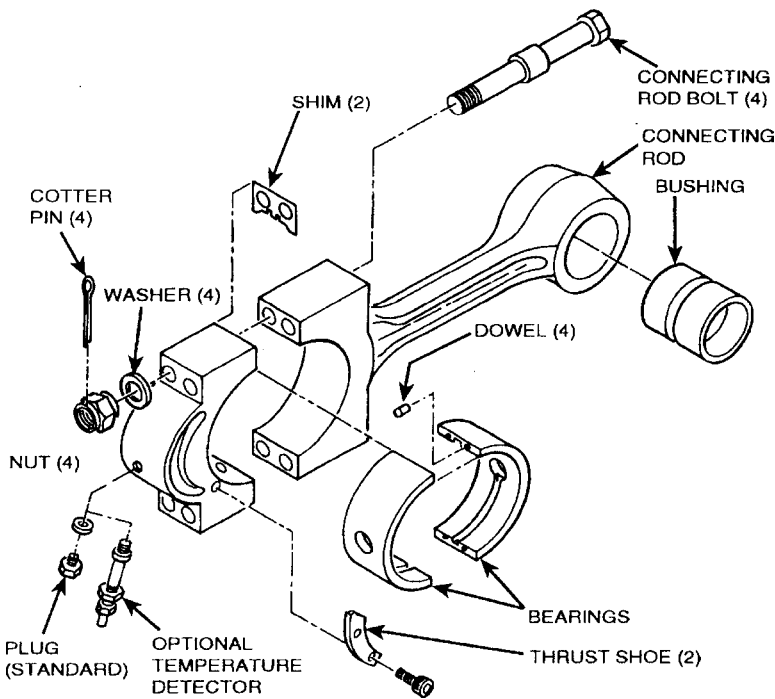


FIGURE 17.16 Component parts of a typical connecting rod. (Source: Dresser-Rand Company, Painted Post, N.Y.)

crosshead pin bushing. The crosshead bushings are made from replaceable bronze. The bolts of the connecting rod are special forgings. The larger ones have rolled threads to maximize the strength.

17.5 Crossheads

Figures 17.17 and 17.18 illustrate a crosshead used in a reciprocating compressor. The crosshead is a sliding component. It is manufactured typically from cast steel, or cast or ductile iron. There are options for the cast steel to meet the specification API 618.



FIGURE 17.17 Crosshead for a reciprocating compressor. (Source: Dresser-Rand Company, Painted Post, N.Y.)

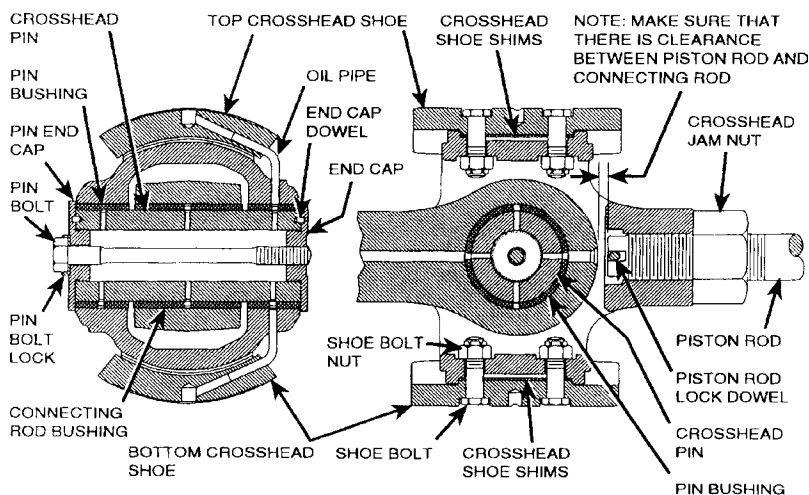


FIGURE 17.18 Cross-sectional views of crosshead components. (Source: Dresser-Rand Company, Painted Post, N.Y.)

The replaceable shim-adjusted top and bottom crosshead shoes are supplied for units rated over 150 kW. Most crossheads have a floating-pin design. However, a fixed-pin design is used on some larger units. Both types provide a reliable long-term application.

17.6 Frames and Cylinders

Figure 17.19 illustrates a cast iron compressor frame assembly. A few manufacturers provide fabricated, or welded, equipment frames. However, the majority of compressor frames are made of cast iron. The frames have suitable ribbed bearing supports. This is done for the following reasons:

- To eliminate the deflection of the frame
- To maintain the crankshaft alignment under all operation conditions

The frames rated over 750 kW have either a tie-rod or a tie-bar over each main bearing. This is done to prevent deflections from the inherently high horizontal gas and inertia forces. The frames are totally enclosed. They are designed to withstand outdoor conditions. They also have large maintenance access covers.

Figures 17.20 through 17.25 illustrate compressor cylinders. These cylinders have different pressure rating. They are manufactured from any of the following materials:

- Cast iron
- Ductile or modular iron
- Cast steel
- Forged steel
- Carbon and stainless steel

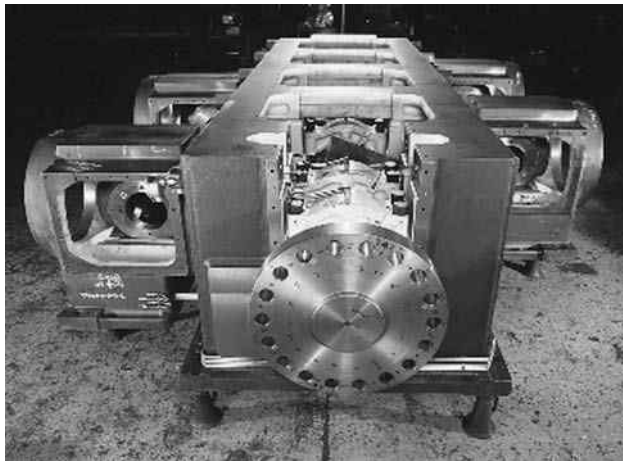


FIGURE 17.19 Cast iron compressor frame assembly showing double bearings at the drive end. (Source: Dresser-Rand Company, Painted Post, N.Y.)

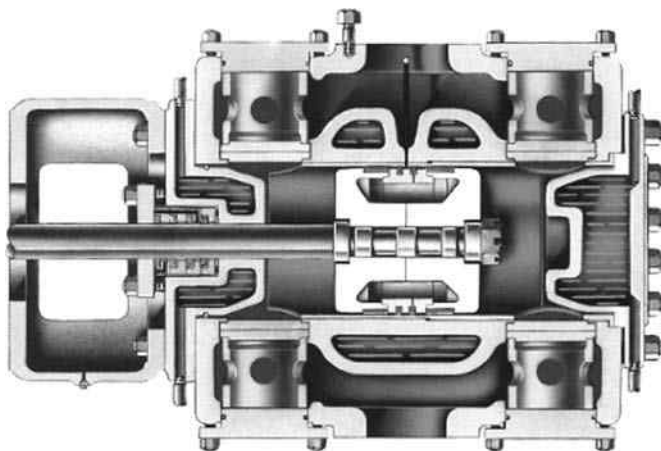


FIGURE 17.20 Low- or medium-pressure double-acting cylinder with a flanged liner and a three-piece piston. Liberally sized jackets reduce thermal stresses and aid in heat dissipation. (Source: Dresser-Rand Company, Painted Post, N.Y.)

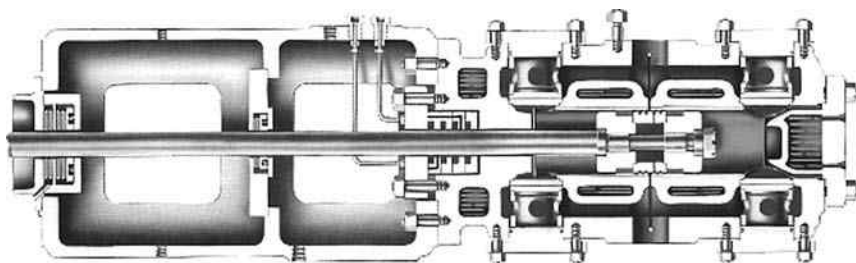


FIGURE 17.21 Medium- or high-pressure double-acting cylinder with a flanged liner. The liner is locked in place by a flange between the head and the cylinder barrel. A two-compartment distance piece designed to contain flammable, hazardous, or toxic gases is illustrated. (Source: Dresser-Rand Company, Painted Post, N.Y.)

Tandem cylinders (Fig. 17.26) are used in applications having limited space and cost. Figure 17.27 illustrates step or truncated cylinders. These cylinders are used in applications having the same considerations as tandem cylinders. Each cylinder has normally a separate forced-feed lubrication system. However, some applications use non-lubricated cylinders. These applications require very clean gas. This is achieved by using a suction filtration of $2\text{ }\mu\text{m}$ if necessary. The piston speed is reduced in these applications to below 4 m/s (700 ft/min). This is done to increase the longevity of the piston, rider, and packing rings.

Most cylinders used in reciprocating compressors are double-acting. This indicates that the gas is compressed in both halves of the cylinder. This occurs when the piston moves toward the cylinder head in one-half of the cylinder. The gas is also compressed in the second half of the cylinder when the piston moves toward the crank end of the

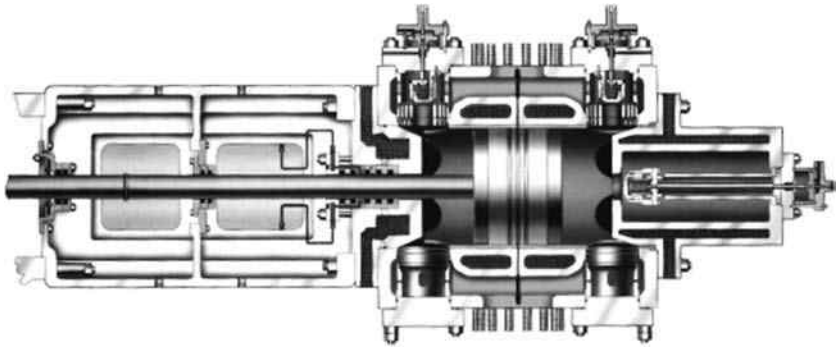
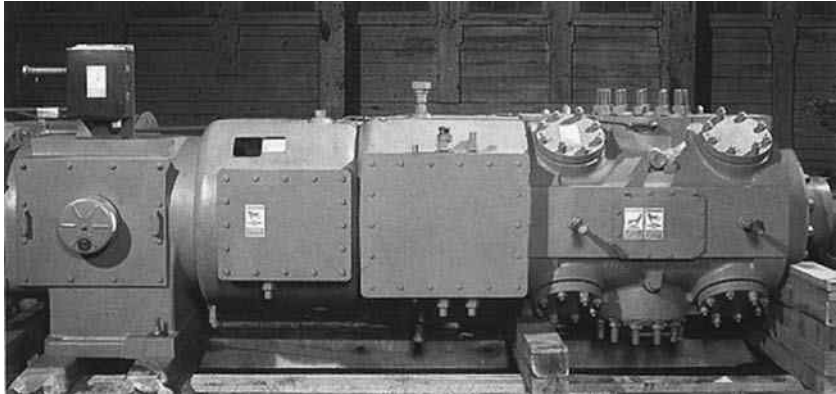


FIGURE 17.22 Cast iron or nodular iron cylinders shown with a two-compartment distance piece and frame extension. Pressures to 10.2 MPa (1500 psi) are typical. (Source: Dresser-Rand Company, Painted Post, N.Y.)

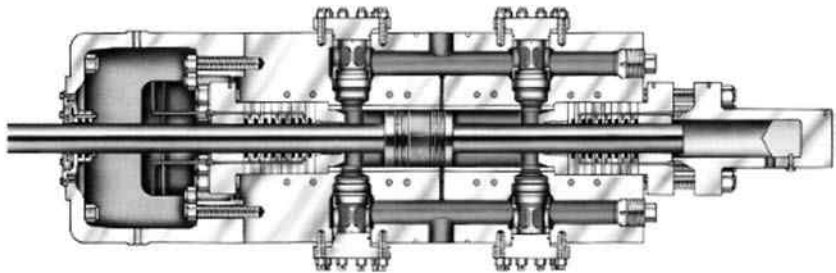


FIGURE 17.23 Forged steel cylinder with a tailored design for pressures to 51 MPa (7500 psi). Tailored construction is used to pressure-balance a piston or to achieve rod load reversals. This load reversal may be needed to properly lubricate the crosshead pin bearing. (Source: Dresser-Rand Company, Painted Post, N.Y.)

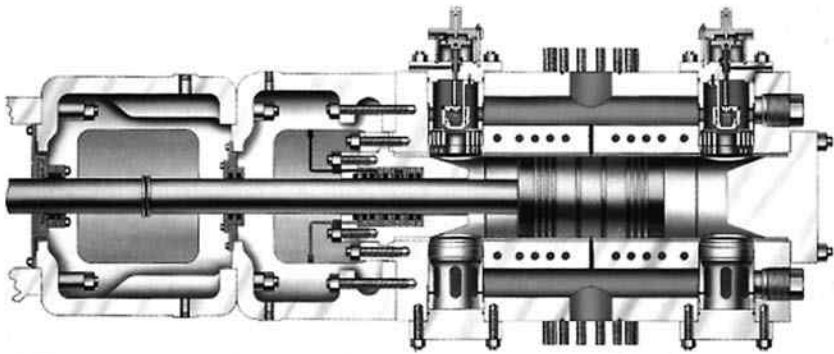
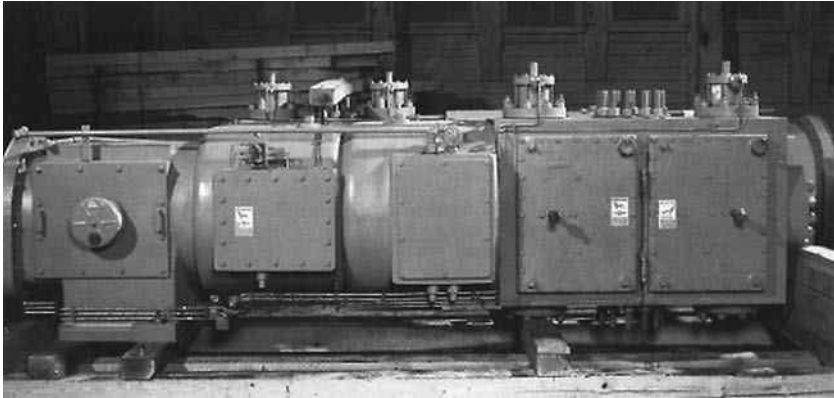


FIGURE 17.24 Forged steel cylinder with two-compartment distance pieces and a frame extension for 20.4 MPa (3000 psi) refinery service. (Source: Dresser-Rand Company, Painted Post, N.Y.)

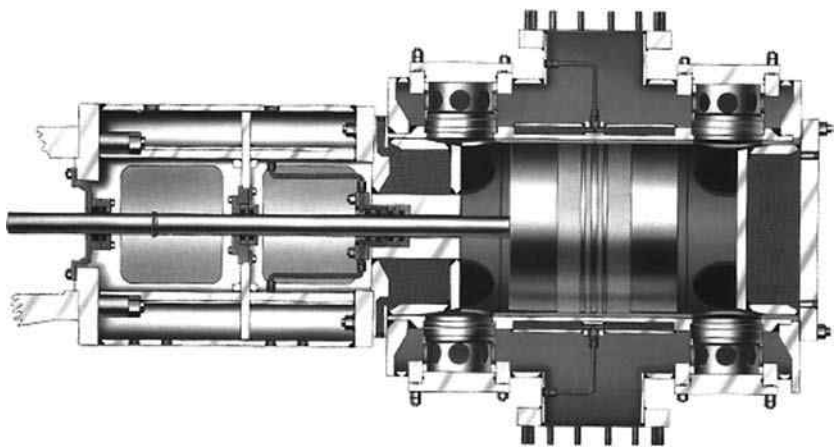


FIGURE 17.25 Fabricated carbon or stainless steel cylinders for special applications. (Source: Dresser-Rand Company, Painted Post, N.Y.)

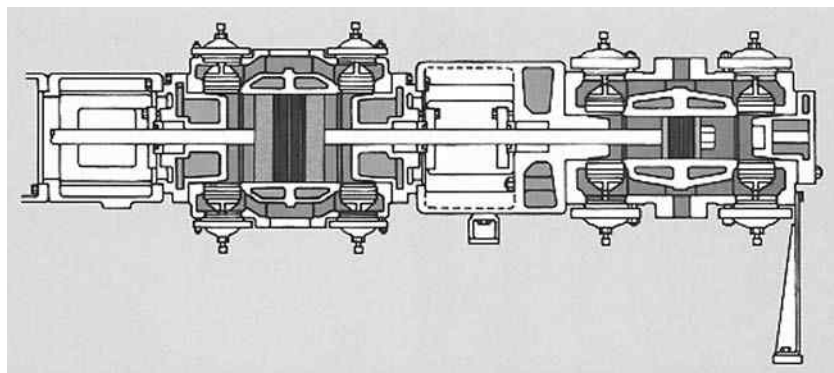


FIGURE 17.26 Tandem cylinders are furnished with a second piston connected inline with the first piston. (Source: Dresser-Rand Company, Painted Post, N.Y.)

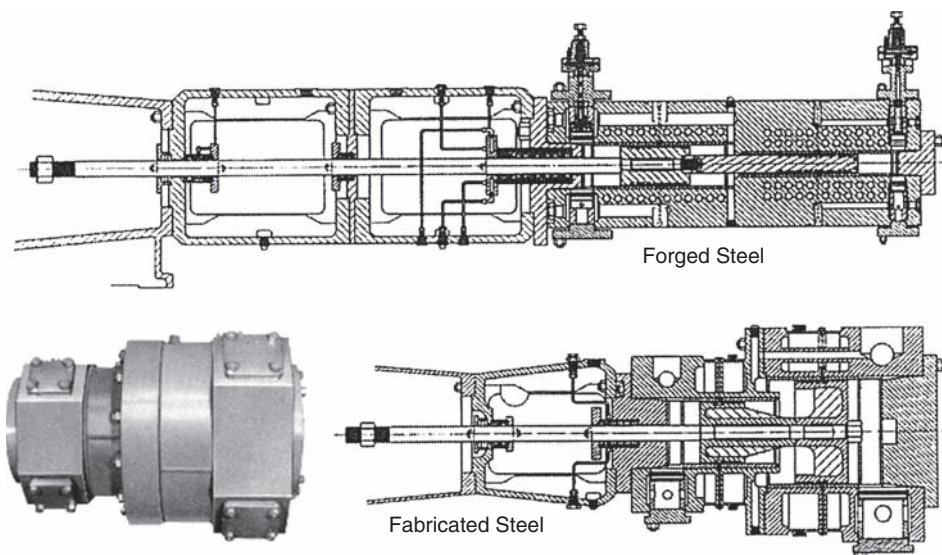


FIGURE 17.27 Truncated or step cylinders allow for space-saving multistaging. (Source: Reprinted with permissions from Cameron Corporation, Houston, TX.)

machine. However, some cylinders are single-acting. These cylinders are employed in high-pressure applications. These applications include automotive fuel gas compression. A small piston displacement is required in these applications. Figure 17.28 illustrates conventional and tandem cylinders. These cylinders are used in a trunk piston compressor. These compressors resemble automobile engines. This is because they do not employ crossheads. The flatness of the discharge pressure curve versus suction volume shown in Fig. 17.28 should be noted. The discharge pressure of the compressor varies only between $150 \text{ bar}_{(\text{abs})}$ (2000 psia) and $325 \text{ bar}_{(\text{abs})}$ (4770 psia) when the suction

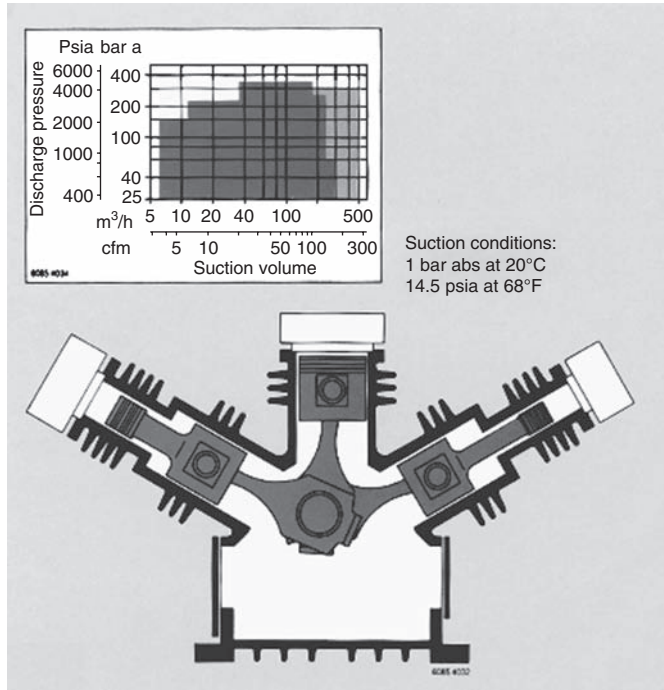


FIGURE 17.28 Trunk piston compressor with conventional stage 1 and step-type higher-stage pistons. (Source: Reprinted with permissions from MAN Diesel & Turbo SE.)

volume varies between 7 m³/h (4 cfm) and 230 m³/h (136 cfm). This is the characteristic of reciprocating compressors. It is different from dynamic compressors. These compressors experience a steady drop in discharge pressure as the suction volume increases.

Replaceable full-length liners are installed in most large double-acting compressor cylinders (Fig. 17.29). These liners are fixed in place to prevent their movement or rotation.

Steel cylinders are always equipped with liners. These liners are manufactured from any of the following materials:

- Centrifugal cast iron
- Nickel-resist (Ni-resist)

The latter material is used for special applications.

17.7 Compressor Cooling

Forced cooling through the cylinder barrel and heads is used commonly on large compressors. Figure 17.30 illustrates the cooling system for large compressor cylinders. Water is used commonly as a coolant. This water should be cleaned and treated. Untreated lake or river water cannot be used as a coolant. Untreated water will cause excessive deposits and fouling buildup in the cylinder jackets. This results in a serious

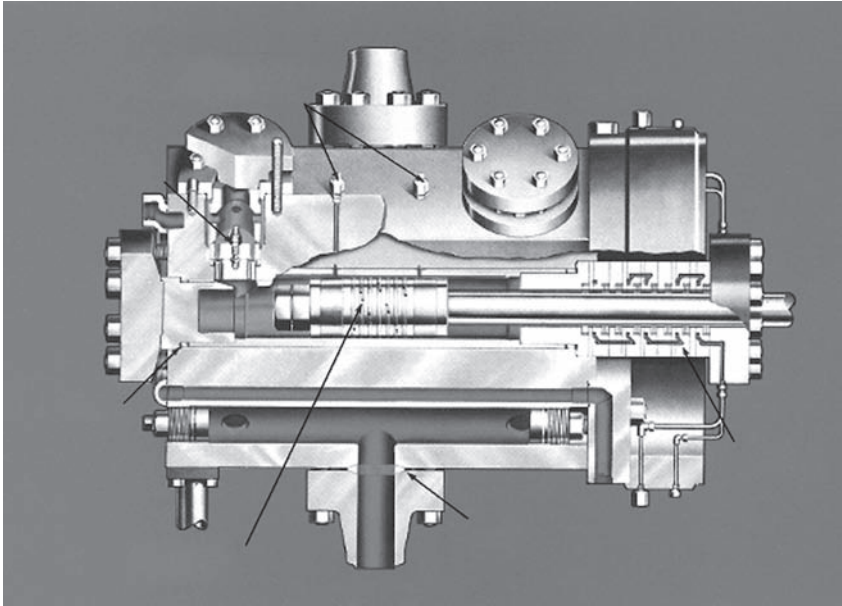


FIGURE 17.29 Compressor cylinder with clamped liner, cooling, and lubricating provisions. (Source: Dresser-Rand Company, Painted Post, N.Y.)

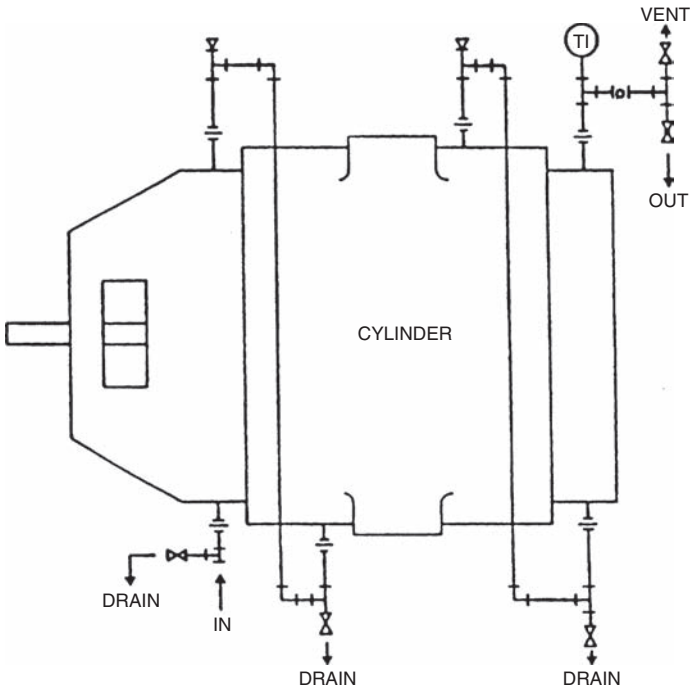


FIGURE 17.30 Forced cooling arrangement for large compressor cylinders. (Source: Dresser-Rand Company, Painted Post, N.Y.)

damage due to cylinder overheating. A closed-cooling system using tempered water-glycol mixture is recommended. This system has the following advantages:

- Minimization of deposits inside the system
- Prevention of liquid condensation from saturated gases in the cylinder

Figure 17.31 illustrates a typical cooling system. This system can be used for one or several compressors.

The cooling system has the following purposes:

- Equalizes the temperature of the cylinders
- Prevents heat buildup

The cooling system removes heat generated by friction. The inter- or aftercoolers remove the heat generated by the compression process. Figure 17.32 illustrates a thermosyphon or static-filled cooling system. This system can be used when the cylinder

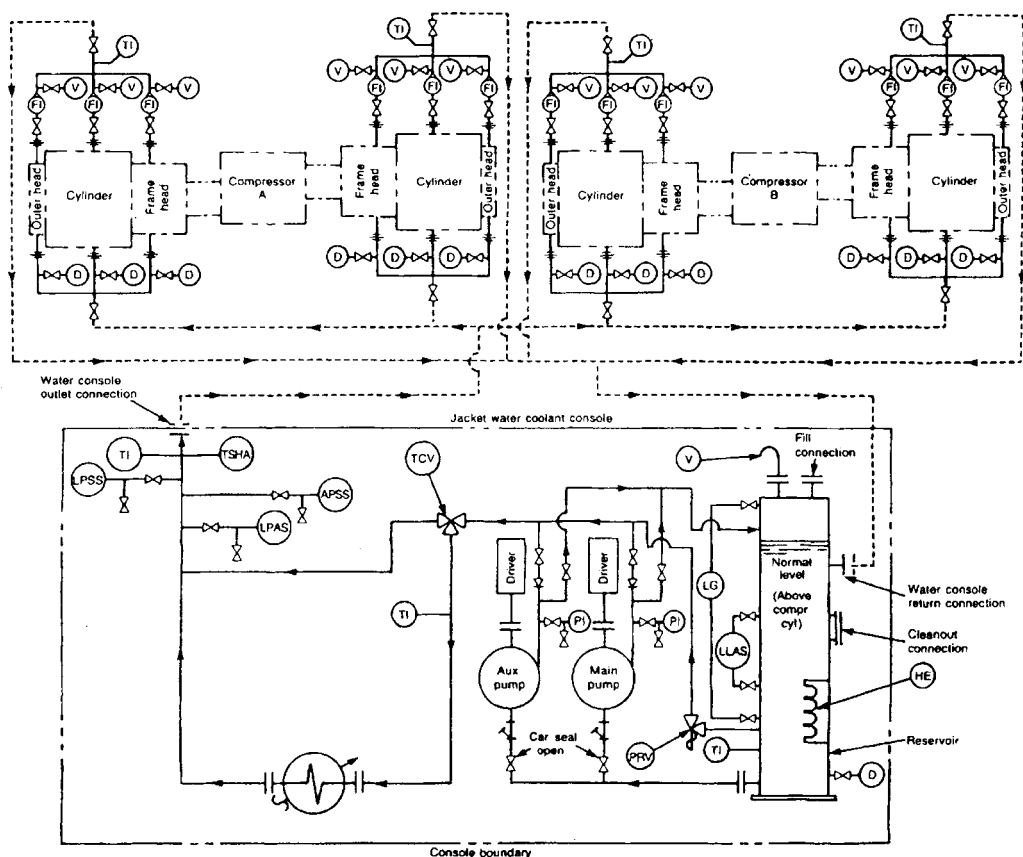
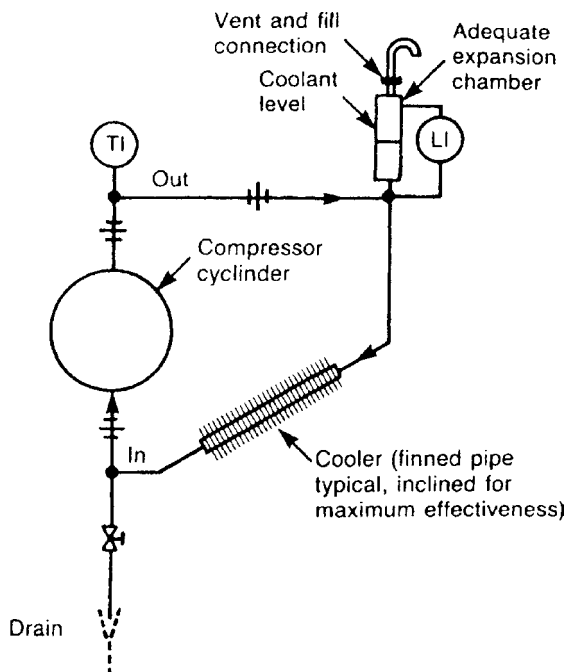


FIGURE 17.31 Closed cooling water system per API 618. (Source: American Petroleum Institute, Washington, D.C.)

FIGURE 17.32 Thermosyphon cooling arrangement per API 618.
(Source: American Petroleum Institute, Washington, D.C.)



discharge temperatures is below 88°C (190°F). The coolant inlet temperature should be at least 6°C (10°F) above the gas inlet temperature. This is done to prevent condensation of liquid droplets inside the cylinders. Serious valve and piston problems could occur due to the presence of liquid droplets inside the cylinder.

17.8 Pistons

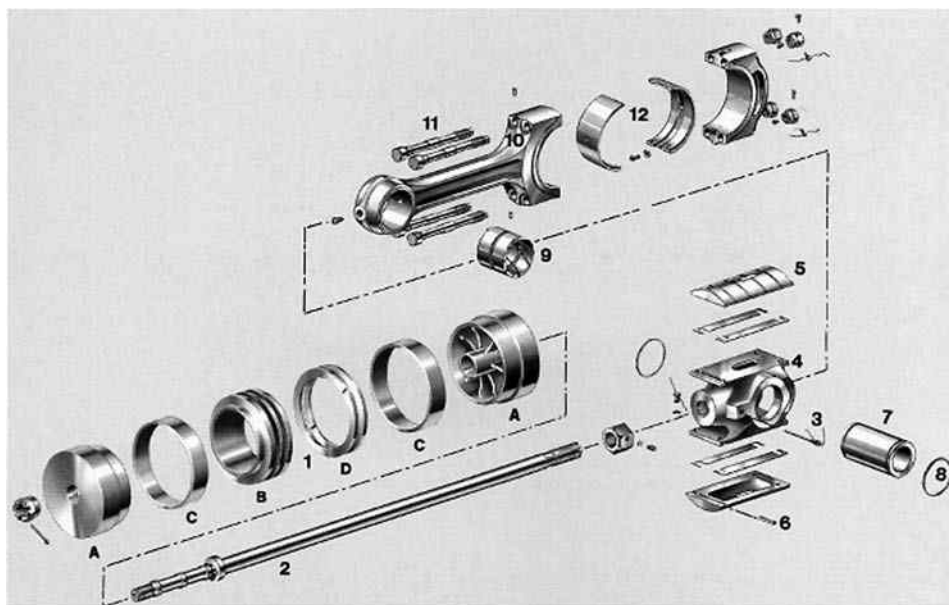
The pistons are manufactured from cast iron in most compressors. However, aluminum is used in the following applications:

- Large pistons
- Higher-speed units

This is done to reduce and balance the inertia forces. One-piece integral steel piston and rod construction is used for applications involving pressures over 15 MPa_(abs). This material is used to increase the piston strength.

17.9 Piston and Rider Rings

The piston and rider rings (Fig. 17.33) used in most units today are made from Teflon [polytetrafluoroethylene (PTFE)] or other high-performance polymers. However, three-piece bronze segmental rings are used normally for applications involving pressures exceeding 30 MPa_(abs). Special plastics or high-performance polymers have been used for some non-lubricated applications.



1. **3-piece piston** allows for two solid slip-on Rider Bands.

- A. Piston end bells
- B. Ring carrier
- C. Solid rider bands
- D. Piston rings

2. **Piston rod**

3. **Dowel pin** so piston rod can not turn

4. **Box-section crosshead**

5. **Babbitt-faced cast iron cross-head slippers**

6. **Crosshead slipper key**

7. **Full-floating crosshead pin**

8. **Crosshead-pin retainers**

9. **Renewable crosshead pin bearing**

10. **High-strength connecting rod**

11. **Connecting bolts**

12. **Split crankpin bearing**

FIGURE 17.33 Reciprocating components of a typical compressor. (Source: Dresser-Rand Company, Painted Post, N.Y.)

TFE rider rings have been used for many lubricated and all non-lubricated applications. These rings support the weight of the piston and piston rod. The following are the types of rider rings:

- *Split type*: These rings are normally located in the center of the piston as shown in Fig. 17.34.
- *Band type*: These rings are normally stretched onto the piston. The bearing pressure on the rider rings is normally below 68.7 kPa (10 psi). The cleanliness of the gas is critical for the longevity of the piston, rider, and packing rings. The entrainment of dirt, piping rust, and scale into the cylinders will rapidly deteriorate the condition of the following components:
 - Piston rings
 - Rider rings
 - Packing rings
 - Cylinder bore
 - Valves



FIGURE 17.34 PTFE rider bands used to support pistons of contact-type non-lubricated compressors. (Source: Dresser-Rand Company, Painted Post, N.Y.)

17.10 Valves

Most compressors use spring-loaded gas-actuated valves. Figures 17.35 through 17.39 illustrate two of the many valve configurations. The valves are normally symmetrically placed around the outer circumference of the cylinder. They can be removed from the outside for servicing and maintenance. Poor valve design may result in the fall of failed components into the cylinder. This may lead to catastrophic damage and safety incidents.

Valves that meet the specification API 618 have a center bolt (Fig. 17.35). This feature assists in preventing the failure of the guard or seat. The bolt design prevents it

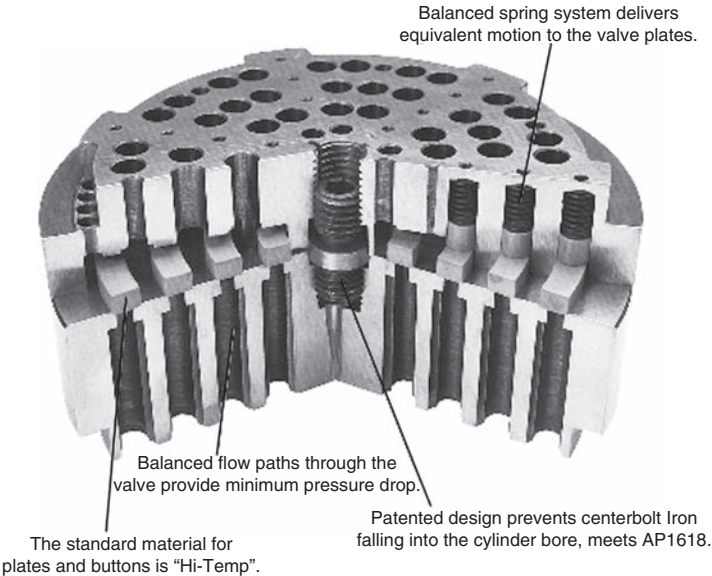


FIGURE 17.35 Plate valve. (Source: Dresser-Rand Company, Painted Post, N.Y.)

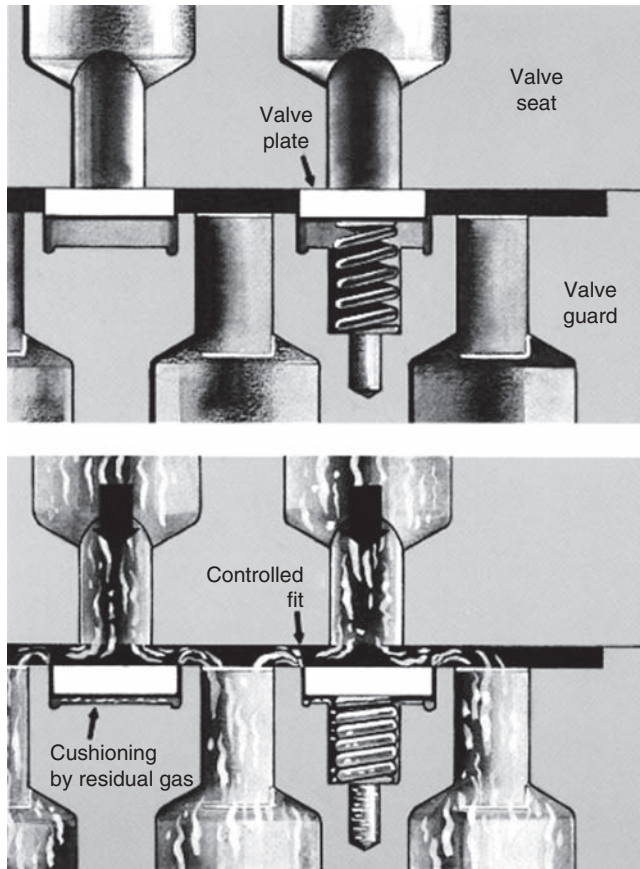


FIGURE 17.36 Cutaway of the closed and open positions of a damped plate valve, illustrating the pneumatic damping feature. (Source: Dresser-Rand Company, Painted Post, N.Y.)

from dropping into the compression chamber when it fails. This bolt plays an important role in determining the valve fixed-clearance and physical strength. All the differential pressure will be carried by the valve seat alone if the center bolt is not used. The center bolt allows the use of the guard's physical strength. This is because it ties the guard and seat together. This feature reduces the size of the clearance volumes. This advantage results from the thinner seats and guards in the valves employing the center bolt. Poorly designed valves can also result in a significant drop in the compressor polytropic efficiency. Thus, the highest-quality valves should be used to optimize the performance of these units.

Figure 17.35 illustrates a plate valve. The principle of pneumatic cushioning is implemented sometimes in the enhanced versions of these valves. A small amount of gas is trapped as shown in Fig. 17.36. This reduces the wear of the plates. Figures 17.37 and 17.38 illustrate a deck-and-a-half and a double-deck valve, respectively. These valves have larger flow areas than normal valves. This feature improves the compressor polytropic efficiency.

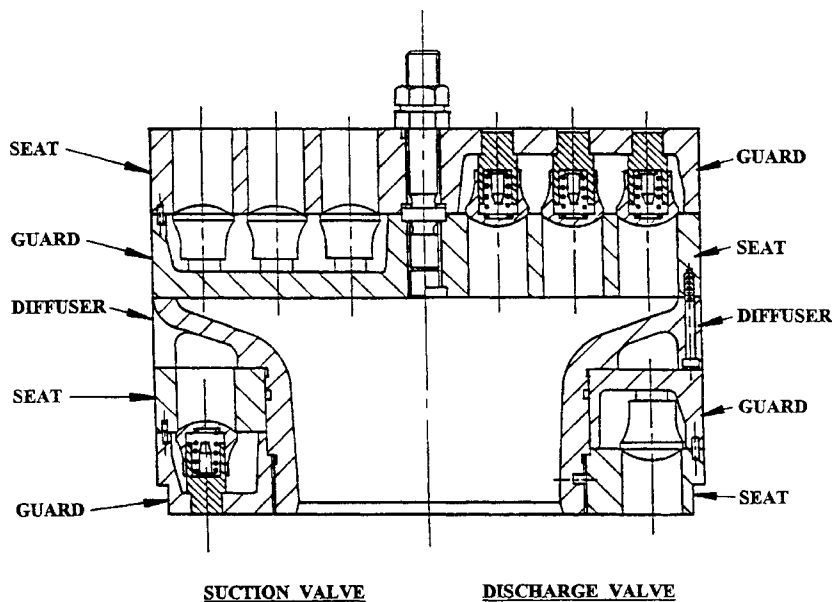


FIGURE 17.37 Deck-and-a-half compressor valve. (Source: Reprinted with permissions from ACI Services, Inc.)

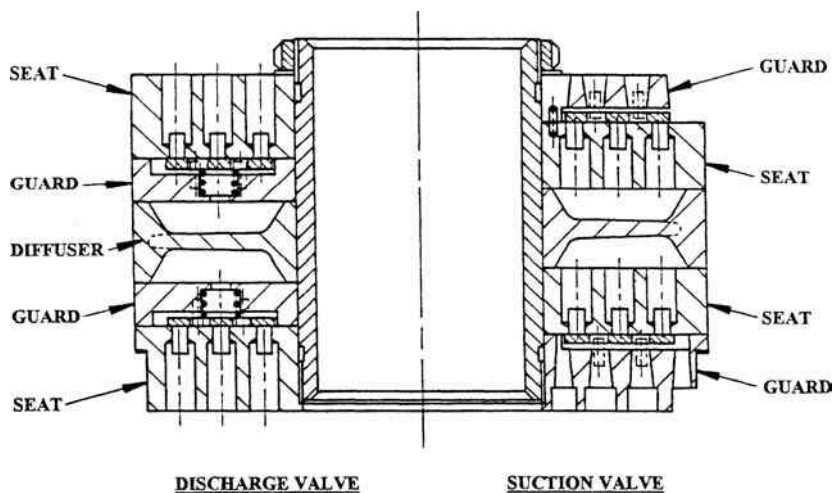


FIGURE 17.38 Double-deck valve, ported plate, opposed-flow type. (Source: Reprinted with permissions from ACI Services, Inc.)

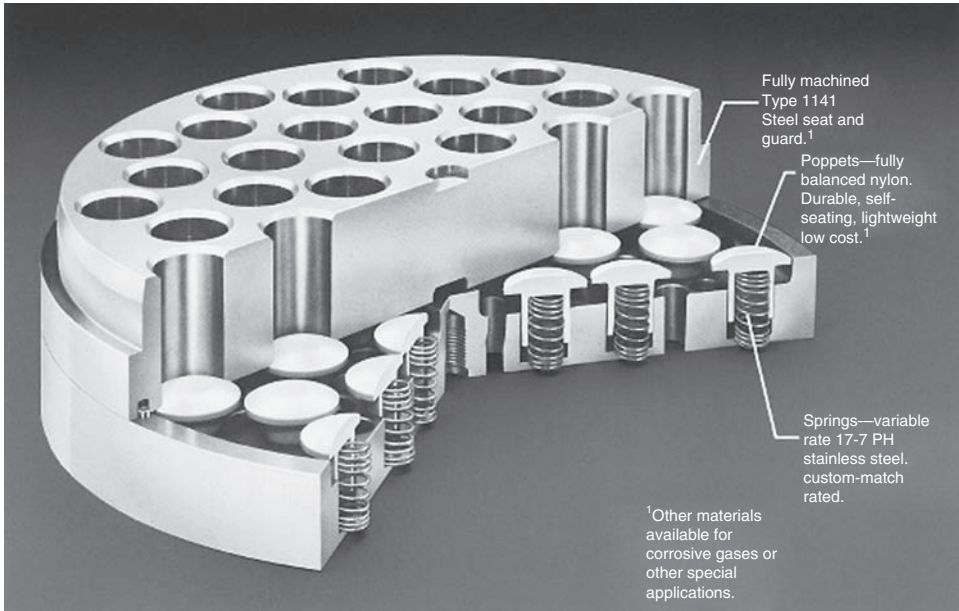


FIGURE 17.39 Poppet valve. (Source: Reprinted with permissions from Cameron Corporation, Houston, TX.)

The following types of valves are also used in industry:

- *Straight channel valves:* These valves are used mainly for high-pressure applications.
- *Circular channel ring valves:* These valves are used for medium-pressure applications.
- *Poppet valves:* These valves are used primarily in low-pressure applications (Fig. 17.39).

Many valves use components manufactured from high-performance polymers. A typical material used in these valves is known as PEEK (polyether ether ketone). The damped valve (Fig. 17.36) has replaced the channel-type design in most high-pressure applications. This is due to the following advantages in damped valves:

- The lower mass of the plates.
- The greater damping.
- The plates have fewer stress concentrations.

Poppet valves have been used primarily in low-pressure ratio, slow- to medium-speed gas transmission applications. However, maintenance problems have occasionally occurred in these valves. This is due to alignment problems between the valve seat and valve guard. Nevertheless, well-designed poppet valves are widely employed in the application range shown in Fig. 17.40.

Figure 17.41 illustrates the basis for a valve or compressor manufacturer's calculations. Figure 17.42 illustrates acceptable and unacceptable valve behavior.

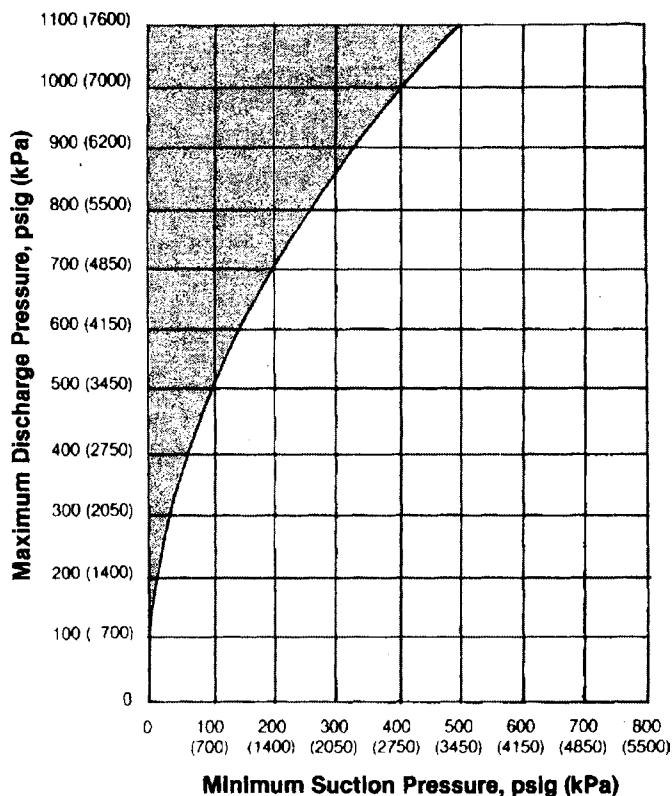


FIGURE 17.40 Poppet valve application range. (Source: Reprinted with permissions from Cameron Corporation, Houston, TX.)

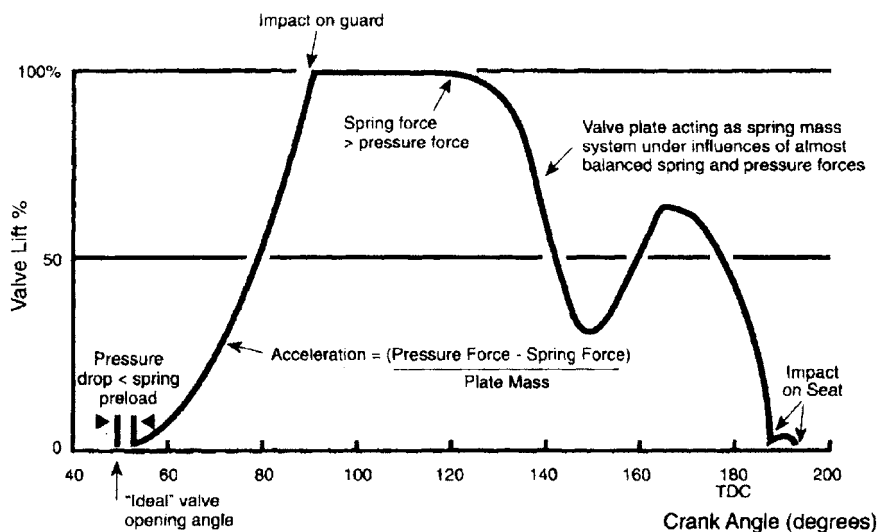


FIGURE 17.41 Basis of valve dynamics calculation. (Source: Dresser-Rand Company, Painted Post, N.Y.)

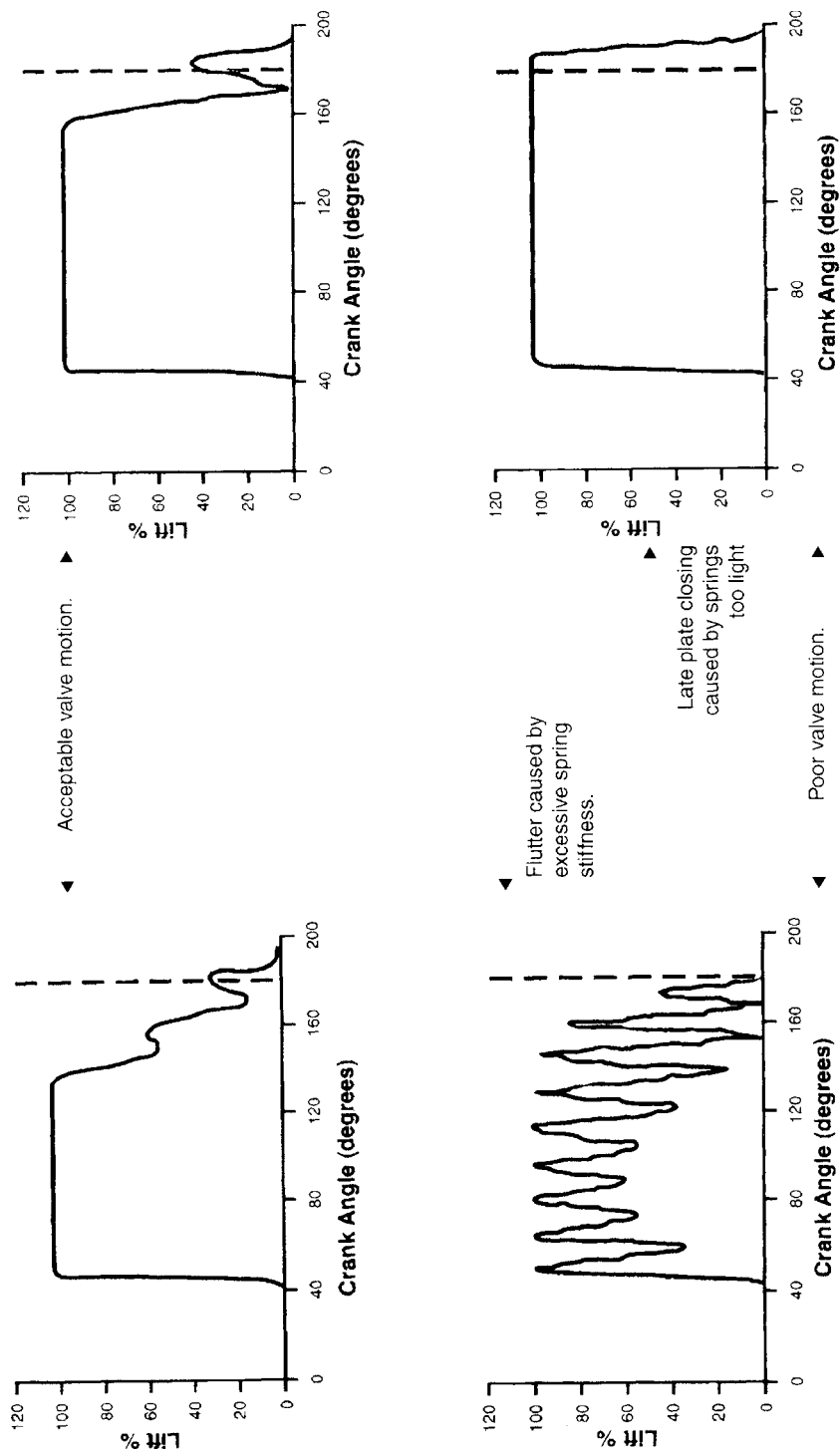


FIGURE 17.42 Acceptable and unacceptable valve motion illustrated on lift versus crank-angle diagrams. (Source: Dresser-Rand Company, Painted Post, N.Y.)

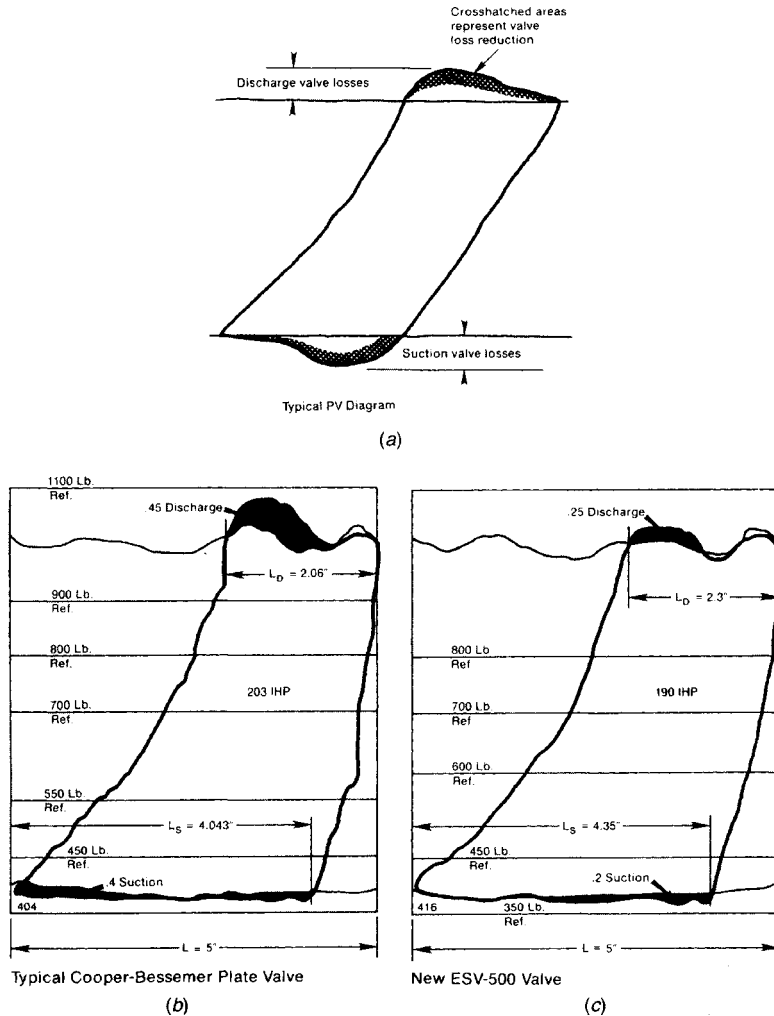


FIGURE 17.43 The P-V diagrams can reveal valve losses. Typical diagram (a) is compared with a traditional plate valve (b) and enhanced design (c). (Source: Reprinted with permissions from Cameron Corporation, Houston, TX.)

Figure 17.43 illustrates the pressure volume (P-V) diagram of three types of valves. The valve losses are shown on each diagram.

17.11 Piston Rods

Piston rods are normally made from AISI 4142 alloy steel. They are induction hardened in the packing travel segment. The minimum acceptable Rockwell hardness in this segment is 50 C. The piston rods are also made from stainless steel in some applications. This material has a high resistance to corrosion. However, it cannot be hardened above

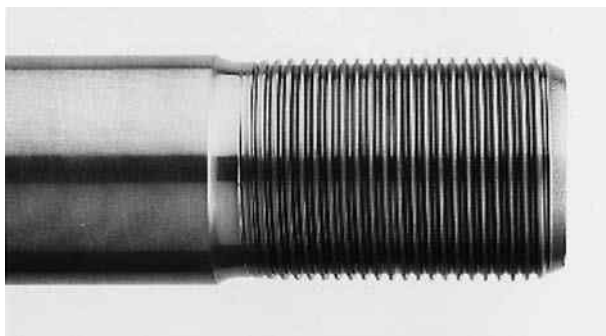


FIGURE 17.44 Rolled thread on a piston rod. (Source: Dresser-Rand Company, Painted Post, N.Y.)

Rockwell hardness of 40 C. Hard coatings are applied to these rods. They include tungsten carbide and ceramic materials. The piston rods are furnished with precision-controlled rolled threads (Fig. 17.44) in high-quality process compressors. This feature provides a much higher fatigue strength than cut threads. The specification API 618 also specifies the use of rolled threads on these rods.

17.12 Packings

Packings (Fig. 17.45) are used to seal the gas in the compressor. They are installed wherever piston rods protrude from the cylinders and distance pieces. Vented, full-floating, self-lubricating PTFE packing is used in most compressors. This is due to their longevity

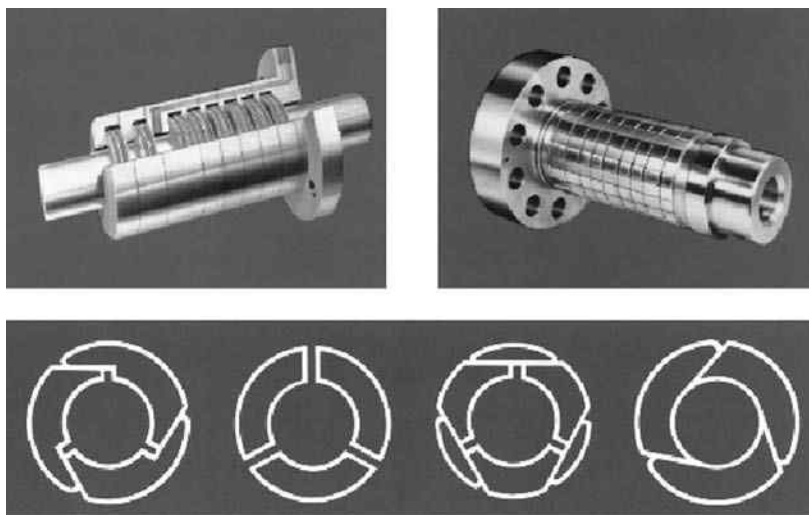


FIGURE 17.45 Packing cartridges and available arrangements. Single-, double-, radial-, or tangential-cut rings with passages for lubrication, coolant, and venting are provided, as required by the application. Surfaces are usually lapped. (Source: Dresser-Rand Company, Painted Post, N.Y.)

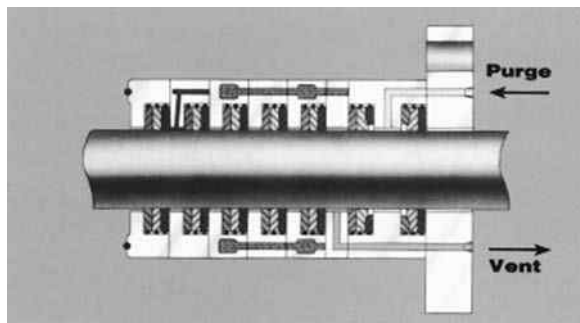


FIGURE 17.46 Lubrication and cooling passages on rod packing. (Source: Dresser-Rand Company, Painted Post, N.Y.)

and reliability. The gas leakage from this type of packing is minimal. They require only one lubrication feed normally. However, two feeds (Fig. 17.46) are required in applications involving pressures over 15 MPa_(abs). Most packing cases have internal cooling passages. Figure 17.47 illustrates a typical self-contained cooling system for piston rod pressure packing.

17.13 Cylinder Lubrication

The compressor maintenance requirements will be reduced significantly when the cylinders receive the proper lubrication. A forced-feed lubrication system is used typically for the cylinders. This system is different from the crankcase system. The lubricator is normally driven by the crankshaft. However, some of them are driven by a motor. There is at least one lubricator feed in the top of the cylinder bore. There is also one lubricator feed into the bottom of the bore and packing are also provided for some cylinders that have a large diameter and high pressure. The pumping unit of each lubricator delivers from 10 to 50 drops/min. A flow adjustment is provided on each unit. The flow rate varies depending on the application. The required flow rate can only be determined by experience and cylinder inspection. This inspection involves checking for adequate oil film in the cylinders. The following lubricants are used for the cylinders:

- Heavy well-refined oil. This oil is normally compounded with 5% to 10% animal fat if the gas is saturated.
- Diester synthetic lubricants.

17.14 Distance Pieces

The distance pieces are normally provided as steel or cast iron castings or steel weldments. Their shape varies to meet the application. Figure 17.48 illustrates a standard distance piece. It is a single compartment with vent and drains.

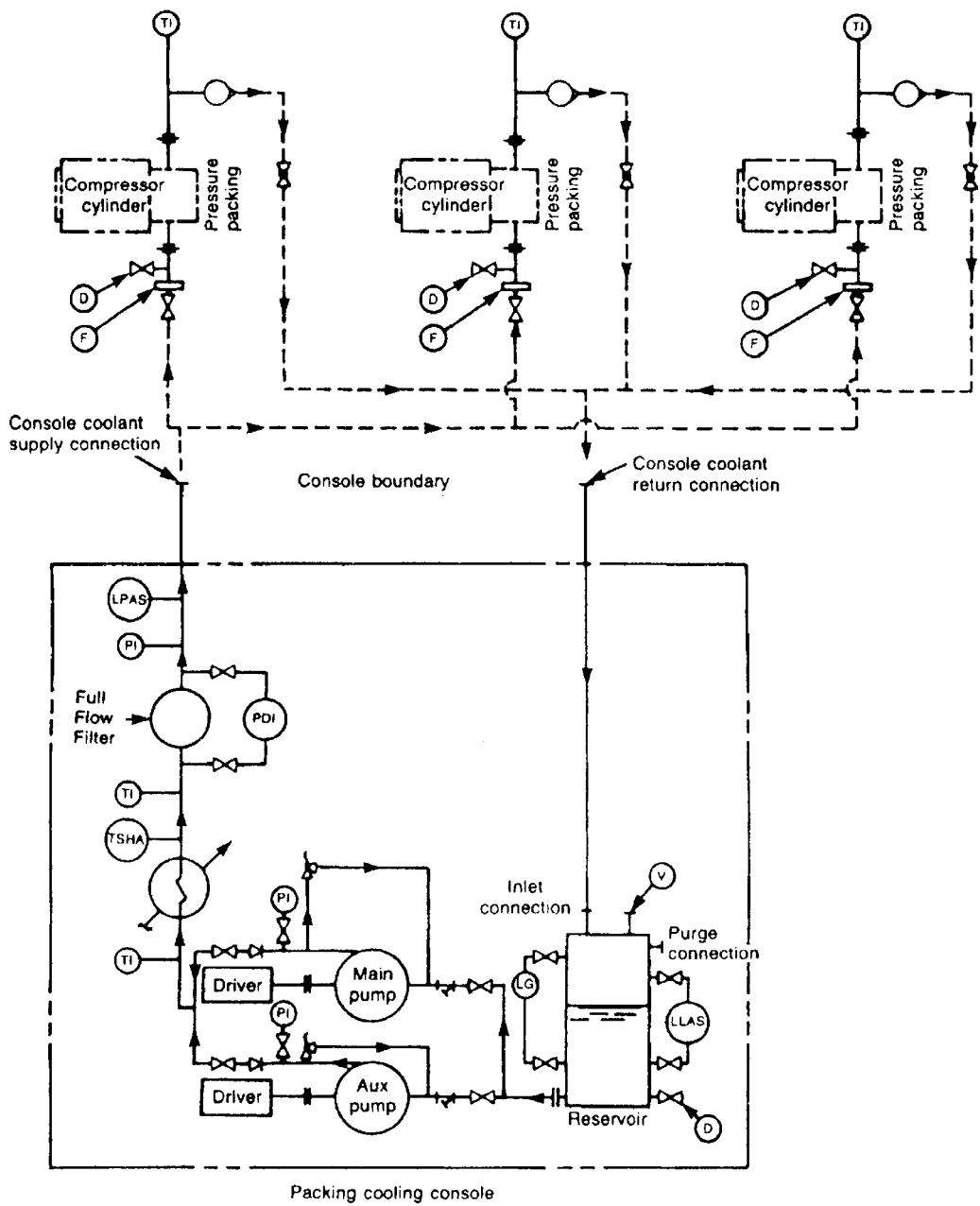


FIGURE 17.47 Typical self-contained cooling system for a piston rod pressure packing schematic, per API 618. (Source: American Petroleum Institute, Washington, D.C.)

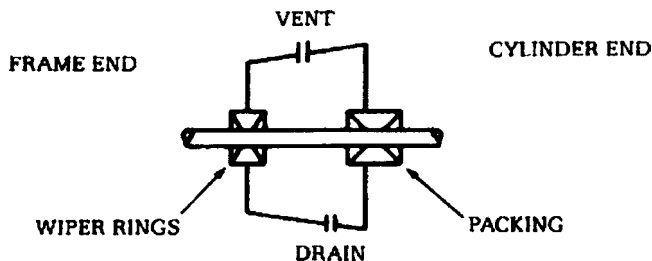


FIGURE 17.48 Single-compartment API standard distance piece for general compression applications. (Source: Dresser-Rand Company, Painted Post, N.Y.)

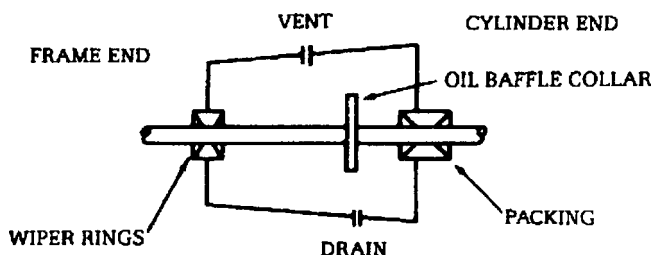


FIGURE 17.49 Extended-length single-compartment distance piece. No portion of the piston rod that enters the packing will enter the frame oil wiper rings. (Source: Dresser-Rand Company, Painted Post, N.Y.)

The purpose of the extended-length distance piece is to prevent the portion of the piston rod that enters the crankcase oil wipers from entering the cylinder packing (Fig. 17.49). The primary purpose of the oil wiper is to prevent oil leakage from the crankcase. However, it will not prevent the gas from entering the crankcase.

Figure 17.50 illustrates a two-compartment distance piece. It has an intermediate packing. This design is recommended for the following applications:

- Gases that could contaminate the crankcase oil
- Hazardous gases

The gas is vented from the cylinder-side compartment to a safe area. The crankcase-side compartment is vented by any of the following methods:

- To atmosphere
- Purged with nitrogen

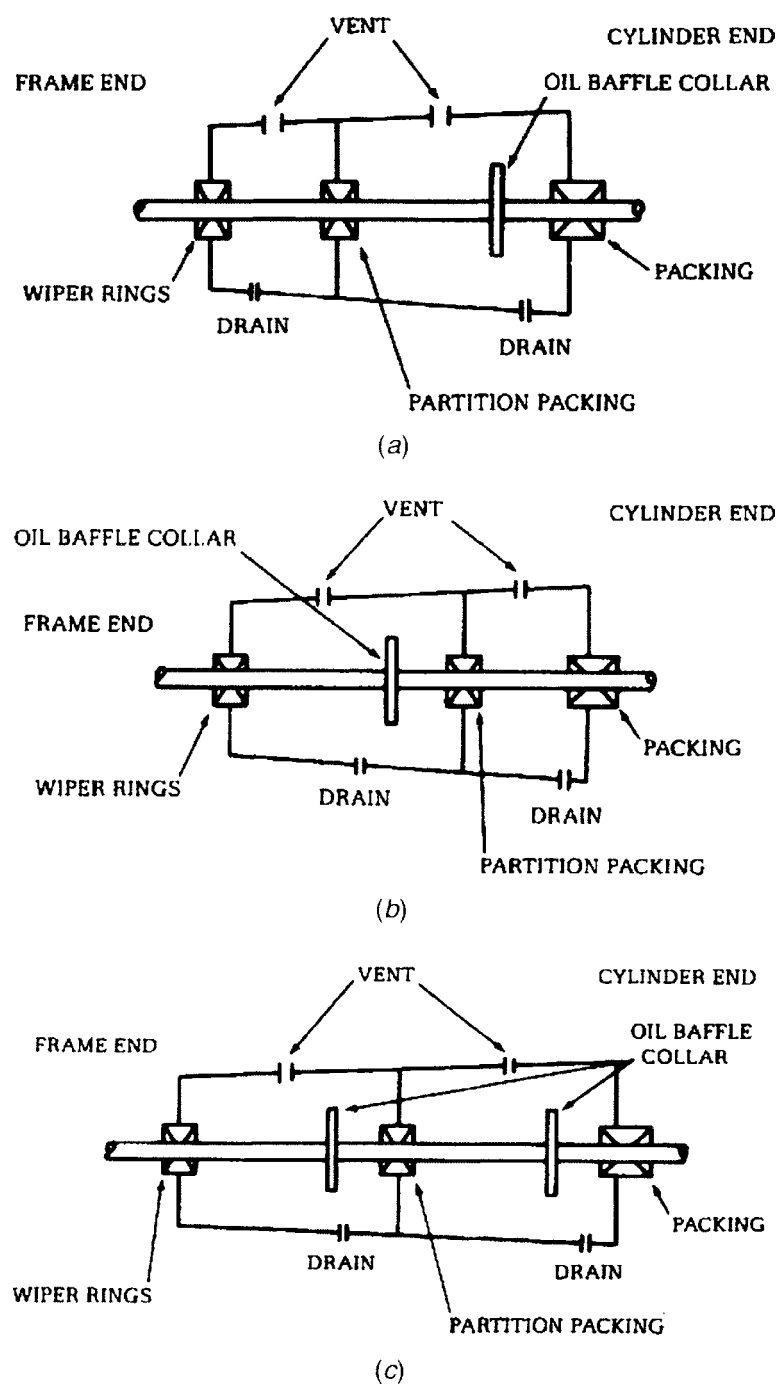


FIGURE 17.50 Two-compartment distance pieces with lubricated partition packing. Baffle collars at cylinder end (a), frame end (b), or both ends (c) prevent oil migration. Venting or purging of each compartment is possible. (Source: Dresser-Rand Company, Painted Post, N.Y.)

The vents from the cylinder-side compartment and the crankcase-side compartment should always be separated.

17.15 Bibliography

Bloch, H. P., *A Practical Guide to Compressor Technology*, 2nd ed. John Wiley & Sons Inc., Hoboken, New Jersey, 2006.

CHAPTER 18

Reciprocating Air Compressors Troubleshooting and Maintenance

18.1 Introduction

An adequate compressor-maintenance program will maximize the reliability and efficiency of the compressor system. The following sections provide guidance to establish a troubleshooting and maintenance program for reciprocating air compressors.

18.2 Location

The following recommendations will enhance the compressor maintenance:

- The compressor area should be kept clean and well-lighted.
- Adequate space should be provided around the compressor to allow the dismantling of any parts during maintenance.
- Adequate overhead lifting capability (cranes) should be provided to facilitate the compressor overhaul.
- The compressor should be positioned near access to clean, cool, outside air. This will reduce the cost of suction air piping.

18.3 Foundation

An adequate compressor foundation (Fig. 18.1) is essential for satisfactory operation and maintenance of the compressor. A foundation designed without sufficient mass will lead to compressor vibration. This vibration will have the following consequences:

- Cracks in the compressor discharge, suction, and water line
- Excessive wear of compressor components

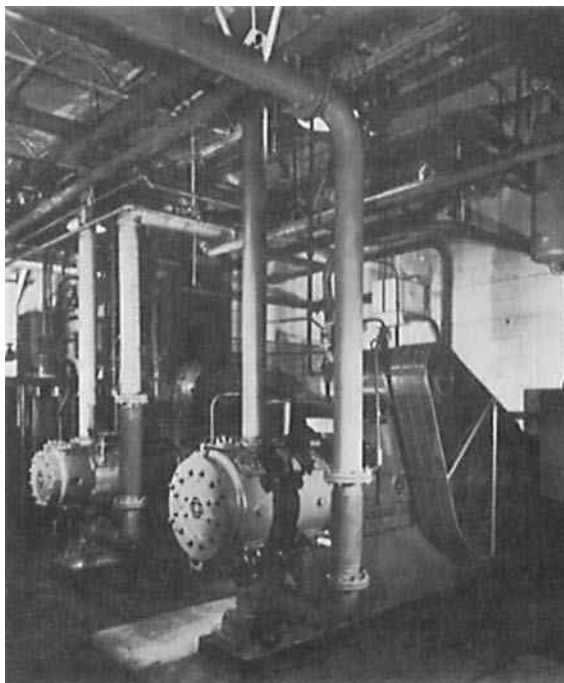


FIGURE 18.1 Compressors on proper foundation. (Source: Reprinted with permission from Higgins L., *Maintenance Engineering Handbook*, 5th ed., McGraw-Hill, New York, 1995, with permission from McGraw-Hill.)

For compressors requiring concrete foundations, the manufacturer of the compressor furnishes the following:

- Drawings showing the foundation above the floor line
- The weights of the components to be mounted on the foundation
- The out-of-balance forces that must be absorbed by the foundation

The type of soil under the foundation will determine its amount. A competent foundation engineer should be consulted to determine the depth and size of the foundation below the floor line. The foundation engineer will take test cores. The results of these tests will be used in calculating the soil carrying capacity. The foundation engineer will use this information along with the weights and out-of-balance forces to design an adequate foundation for the compressor.

Many small vertical compressors are installed on concrete floors. They normally operate well. This is because the large mass of the floor provides sufficient dampening for any out-of-balance forces of the compressor.

Some compressors cannot be installed on a foundation or a concrete floor that is poured on the ground. They are located on a floor that does not have a solid base under it. Isolation dampers should be used for these applications. They are installed under the base supporting the compressor and its driver. Flexible connection should be used in these applications to connect the suction, discharge, and water lines of the compressor.

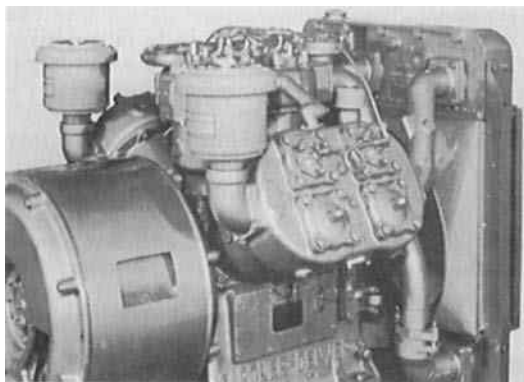


FIGURE 18.2 Air filter for a compressor. (Source: Reprinted with permission from Higgins L., *Maintenance Engineering Handbook*, 5th ed., McGraw-Hill, New York, 1995, with permission from McGraw-Hill.)

This is required to prevent vibration and noises from being transmitted through the building. The manufacturers of isolation dampers should be consulted in these applications.

18.4 Air Filters and Suction Lines

An air filter must be installed at the inlet to every compressor. It should be the most efficient type made for the application. Explicit instructions for servicing the air filter at regular intervals should be provided to maintenance staff. Figure 18.2 illustrates an air filter located at the inlet to the compressor.

The air suction line to the compressor must meet the following requirements:

- Airtight
- Free of dirt, chips, and scale
- Corrosion resistant

PVC piping meets most of these requirements. Some pipe lengths generate acoustical resonance. This phenomenon can result in excessive pulsations within the piping. Care should be taken to avoid pipe lengths that generate acoustical resonance. The compressor manufacturer should provide a list of pipe lengths that generate acoustical resonance. These pipe lengths should be avoided. The shortest possible suction line is normally preferable.

The air filter should be cleaned regularly. The time interval for cleaning the air filter depends on its type and location. This time interval is best determined by the differential pressure across the filter. Most air filter manufacturers provide differential pressure devices. These devices indicate when the filter should be cleaned.

18.5 Air Receiver Location and Capacity

The air receiver dampens the pulsations in the discharge line from the compressor. It also smoothen the air flow to the service lines. The air receiver serves as a reservoir of compressed air. It handles the sudden and unusual momentary demands that exceeds

the capacity of the compressor. The air receiver also precipitates moisture that may condense in it and prevents this moisture from being entrained into the air-distribution system. The air receiver should be located as close as possible to the compressor. This is done to minimize the length of the compressor discharge line. This arrangement reduces the pressure drop between the compressor and the air receiver. Some receivers are located outdoors. Problems occur in these receivers when the temperature drops below 0°C (32°F). The ordinary top-outlet safety valve can freeze shut when the temperature drops below the freezing point. This creates a hazard. The safety valve freezes shut at these temperatures because the water in it freezes. The safety valve should be placed with its opening down. This arrangement keeps the water out of the valve. The valve will operate when necessary. The receiver drain valve can also freeze when the compressor is shut down and the ambient temperature is below 0°C (32°F). This is because the air stops flowing through the receiver. In some cases, parts of the drain break when the drain valve freezes.

The manufacturer of the compressor has normally charts that list the required receiver sizes for various compressor sizes. Compressors that start-and-stop frequently require a larger receiver than the compressors that operate continuously. This feature is required to prevent the compressors that start-and-stop frequently from starting too often. The electrical inrush (starting) current to the motor is around 6 to 8 times higher than the normal operating current. Thus, the reduction of the frequency of starting the motor will reduce the cost of electricity for the industry.

The air leaving the compressor should enter the receiver at the bottom. Condensate can cause problems in the system. An efficient water-cooled aftercooler and separator should be installed between the compressor and receiver. The moisture will be condensed in the aftercooler. The separator will collect most of the condensed water. The separator can be drained manually or automatically. The aftercooler dries and cools the air. This enhances the efficiency and safety of the system. The air leaving most aftercoolers is within 8.3°C (15°F) of the incoming cooling water temperature. Air-cooled aftercoolers are also available. They are used in locations where water supply is short or expensive. However, they are not as effective as water-cooled aftercoolers. They can drop the temperature of the air leaving the aftercooler to within 11.10 to 16.67°C (20 – 30°F) of the ambient temperature. The compressor vendor should be consulted about receiver problems. The receiver (pressure vessel) must meet the safety codes. It must also pass the inspection by the insurance companies.

18.6 Starting a New Compressor

A careful check of the lubricating system must be made before starting a new or an overhauled compressor. All the places requiring lubrication should have been oiled per manufacturer's recommendations.

Some compressors require cooling water from a water main. The water supply to the compressor should be turned on. A thorough check for leaks should be done. The circulation of water through all parts requiring cooling water should also be confirmed. Some compressors have a self-contained water-cooling system. This system should be filled with water. A check should be done to confirm that all the air is out of it. The discharge line between the compressor and receiver should be checked. Any globe, gate, or check valves located between the compressor and receiver should be placed in the open state. A safety valve must be located between the compressor and the valves. The unavailability of the safety valve can result in an explosion. This would occur if

the valve located downstream of the compressor is left closed when the compressor starts or if the overload protection fails to operate.

Apply driving power momentarily to the compressor after all the points have been checked. Allow the machine to coast to rest. The coasting period should be observed closely to determine if there is any excessive tightness. The proper direction of rotation can also be confirmed during the coasting period. A good indication of the no-load friction can be determined by the time that the unloaded compressor continues to roll after interruption of the driving power to it. The unit can be operated unloaded if the direction of rotation is correct and no additional problems have been detected. Some compressors have pressurized lubrication system for the crankcase. The oil pressure should be checked immediately after start-up. The adequate oil pressure should be reached within 10 seconds following start-up. Otherwise, the unit must be shut down. The cause of the failure should be determined. Some compressors have mechanical forced feed lubricators. The proper feed rate in drops-per-minute from these lubricators to the cylinders and piston packings of the compressor while it is operating unloaded should be specified in the operator's manual. The adequate feed rate from these lubricators should be confirmed before continuing the start-up process.

Excessive amount of condensate will form when operating a water-cooled compressor with a high-cooling water flow rate through the system. This will lead to increase cylinder wear because cold cylinders will not be properly lubricated. The amount of horsepower required in this case will be higher than normal due to poor lubrication. This will lead to an increase in the maintenance and operating cost of the system. The outlet temperature of the water from the compressor should be maintained between 43.3 to 48.9°C (110–120°F). Operation in this range will allow for adequate cooling and lubrication. It will also minimize the condensation in the cylinder. The cold water supply should not be introduced into the cylinder water jackets. This is required to prevent condensation from occurring on the interior cylinder walls. The cold water supply should be circulated through aftercoolers or other water-cooled heat exchangers in the vicinity. The warmed water should then be directed to the compressor cylinder jackets. The cooling water is initially introduced into the interstage intercoolers in applications having multistage compressors. It is then directed to the cylinder jackets. The temperature of the cooling water entering the cylinder jackets should be maintained at least 5.6°C (10°F) above the temperature of the air entering the cylinder to prevent condensation.

The compressor should be operated for 1 to 2 hours unloaded. It should be stopped periodically during this period to check for any overheating in the bearings or other working parts. Partial load should be applied at this stage. The load should then be increased gradually until the maximum load and pressure are reached. The entire break-in period from start-up until the full load is reached should take 4 hours.

The break-in period is very important. Problems can be detected early before they result in a compressor failure. The compressor will operate adequately following the initial run by providing it with the following:

- A clean air supply
- Sufficient cooling water
- Adequate lubrication

A routine maintenance program should be established to maintain efficient operation and minimize maintenance costs.

18.7 Lubrication

Figure 18.3 illustrates a pressurized lubrication system. This system must be checked regularly to ensure adequate operation. The compressor should be kept well lubricated. The oil level should be checked at least once every 8 hours during operation. The oil and greases recommended by the compressor manufacturer should be used. The oil used should have the following characteristics:

- Low carbon-forming tendency
- Low sulfur content
- Contain an oxidation inhibitor
- Viscosity suitable for the existing temperatures

The frequency of oil replacement in the crankcase of a compressor varies depending on the atmospheric conditions. The amount of dirt, dust, and sand in the air varies with the location. The foreign material in the compressor contaminates the oil. The oil will also oxidize during operation. The frequency of oil changes should be determined by the discoloration and changes in the physical condition of the oil. Laboratory services should be used to analyze mail-in oil samples. The oil analysis will provide the following:

- Assist in determining the proper oil change periods.
- Provide a warning of excessive surface wear due to friction and other unusual foreign material contamination.

The inside of the crankcase or power end should be wiped clean with lint-free rags when the oil is being replaced. A good grade of flushing oil should be used to remove

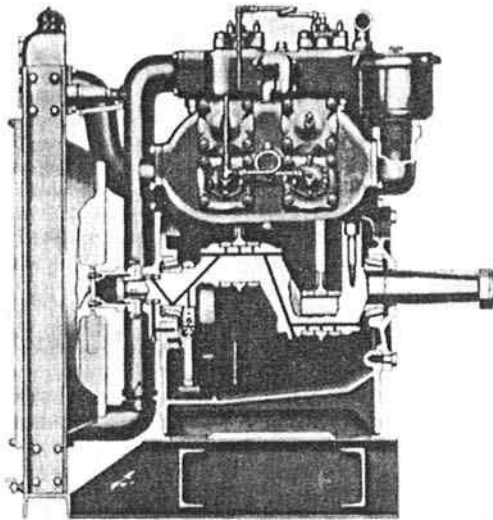


FIGURE 18.3 Pressurized lubrication system. (Source: Reprinted with permission from Higgins L., *Maintenance Engineering Handbook*, 5th ed., McGraw-Hill, New York, 1995, with permission from McGraw-Hill.)

any particles that may have settled on the crankcase floor if it is impossible to wipe it with rags. The oil pump suction strainer which is normally located in the lower part of the oil sump should be cleaned on units having pressurized crankcase lubrication. The replacement or cleaning of the oil filter elements (if applicable) should also be performed. Ensure that the oil-filling container is free from all dirt, grit, or dust while refilling the compressor oil pump.

18.8 Non-Lubricated Cylinders

A large portion of double-acting reciprocating compressors used today are equipped with non-lubricated cylinders. Piston rings and rider rings made from filled-PTFE (polytetrafluoroethylene) are normally mounted on the pistons of these cylinders. They are designed to avoid any contact between the pistons and cylinder boxes. Non-lubricated piston rod packings are also used in these compressors. They are normally made of carbon or filled-PTFE packing rings. Non-lubricated, double-acting cylinders normally have the following components:

- Extended length piston rods
- Distance pieces

These components prevent the portion of the piston rod that enters the lubricated crankcase from entering the non-lubricated piston rod packing. Baffle rings are also mounted on the piston rods. They prevent the creepage of lubricant from the portion of the rod that is wetted with lubricant to the portion that enters the rod packing. The expected service life and reliability of the unit will drop significantly due to any of the following:

- Increase in the intake of particulate and liquid contaminants into the cylinder
- Formation of condensate within the cylinder

The start-up procedure for these units is the same as the one that was described for lubricated cylinder units. The only exception is the comment related to the cylinder forced-feed lubrication.

18.9 Valves

The valves used in reciprocating compressors should be kept in an excellent operating condition. Leaking or broken valves will have the following consequences:

- Excessive operating temperatures
- Loss of air flow from the compressor

These valves should be checked periodically. The operator should ensure that they are in an excellent operating condition. Figure 18.4 illustrates different designs of compressor valves. The inspection times for the valves depend on the following:

- Efficiency of the air cleaner
- Carbon-forming tendency of the oil used
- Overall condition of the compressor

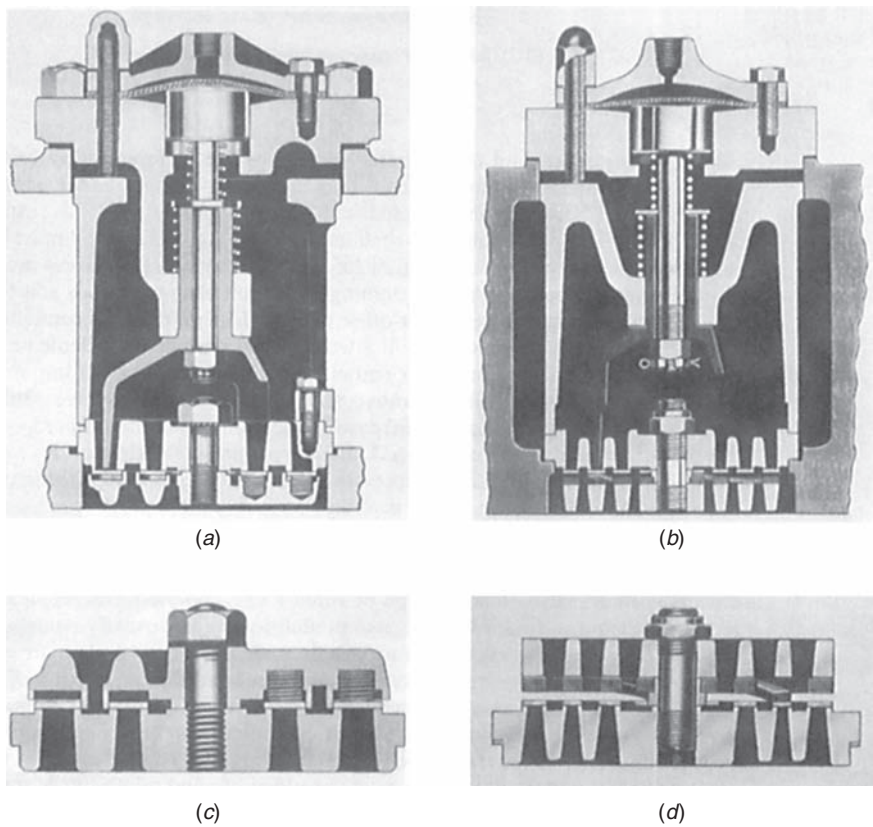


FIGURE 18.4 Compressor valves. (a) and (b) Different designs of suction-unloading valve assemblies. (c) Compressor valve with individual disks and coil springs. (d) Compressor valve with plate disk and finger springs. (Source: Reprinted with permission from Higgins L., *Maintenance Engineering Handbook*, 5th ed., McGraw-Hill, New York, 1995, with permission from McGraw-Hill.)

The impurities will be kept out of the airstream if the air cleaner is efficient and serviced regularly. This will prevent the impurities from lodging in the valves. The carbon buildup on the valves is minimized by using low-carbon-forming oil. The following synthetic lubricants are recommended for air compressor cylinder lubrication:

- Diesters
- Polyol esters
- Polyalphaolefins

These synthetic oils are more expensive than mineral oils. However, they provide the following benefits:

- Cleaner running valves
- Significant reduction in the deposits formed in all hot areas of cylinder air passages and air piping

The synthetic oils are compatible with compressors that have been manufactured in the last decade. However, the compatibility of these oils with older compressors should be confirmed with the manufacturer. Some synthetic oils have solvent like action on hydrocarbon deposits within the compressor and air distribution system. Thus, proper procedures from synthetic oil manufacturers should be followed during the conversion from mineral oils to synthetic oil lubrication. The pistons, rings, and cylinder walls should be kept in a good condition to prevent oil from passing through them. The valve life increases in compressors that have a low oil consumption. This is due to the reduction in carbon deposit buildup on the valves. The inspection interval of the valve varies depending on the application. This interval should be determined based on an investigation performed by the maintenance staff. The valve inspection interval of new compressors is 200 hours of operation.

Spare compressor valves should be kept readily available. This will allow the defective valves to be replaced immediately. The defective valves can be refurbished when time allows. The following are the symptoms of valve troubles:

- Low-discharge air from the compressor
- Heating around the valve compartments

The usual method used to detect defective valves on a single-stage compressor is by checking the temperature of the valve cover plates. The hottest valve under the cover plate should be examined. A blow-back noise will be heard in the air cleaner during operation if the suction valves are leaking.

The intercooler pressure gauge is used to identify defective valves on two-stage compressors. The valves on the low-pressure cylinders should be examined upon detection of low intercooler pressure. The valves on the high-pressure cylinders should be examined when high intercooler pressure is found. The temperature of the valve cover plates should be checked. The defective valve can be located under the hottest cover plate. The indicator of the intercooler gauge will fluctuate above normal intercooler pressure if the high-pressure suction valves are leaking. The intercooler safety valve will also open in this case. The intercooler pressure will increase steadily if the high-pressure discharge valves are leaking. The pressure in the intercooler will continue to increase until the intercooler safety valve releases it.

The air will blow back through the suction line and air cleaner when the low-pressure suction valves leak while the compressor operates under load. The intercooler pressure will fluctuate below its normal value when the low-pressure discharge valves leak. Indentation in the valve disks or plate are indicative of wear between the valve disks or plate. The valve disks or plate should be replaced under the following conditions:

- There are signs of wear on the valves.
- The valves have nicks, chips, or cracks.

The valve seats should be resurfaced if they show any signs of wear. The valve lift should be checked after resurfacing the seat. The valve bumper should be cut down to obtain the correct lift if the lift is found to be more than the one recommended by the manufacturer. The valve will wear out and fail quickly if the lift is excessive.

The valve disks or plates and springs should be replaced if the valve has experienced some overheating. The valve parts will have a much shorter life if they had

experienced overheating. The gasket of the compressor valves must be in an excellent condition. This gasket is normally located under the seat. It should be replaced if it shows any signs of imperfection. A leaking valve-seat gasket will eventually lead to an impairment of the compressor.

18.10 Piston Rings

The compressor efficiency will drop due to the following reasons:

- Wear or impairment of the valves
- Wear of the piston rings

Lubricated piston rings wear very slowly. However, operating time will eventually wear them. The gap between the piston ring and the cylinder will increase. This allows leakage through the gap. Worn piston rings should be replaced at the earliest opportunity to restore the efficiency of the compressor. Scored cylinders will always allow leakage. This will also lead to deterioration in the performance of the compressor. Cylinders should be refurbished if they have been scored. Some cylinder can be rebored. Oversized pistons and rings should be used in this case. Replaceable cylinder liners are used in some applications. These liners should be replaced if they have been damaged.

The piston rider rings should be examined for wear in applications involving non-lubricated cylinders. Piston-to-bore contact will occur if these rings wear out. The piston rider rings that are installed in horizontal and angular-mounted cylinders experience a higher rate of wear than the ones installed in vertically mounted cylinder. A feeler gauge should be used to confirm the minimum piston-to-bore clearance in these applications when the valves are inspected or overhauled. These rings should be replaced if the minimum clearance is less than 0.038 cm (0.015 in). The compression rings should also be examined for wear and end gap when the rider rings are being replaced.

18.11 Intercoolers and Aftercoolers

The moisture traps or compartments of intercoolers and aftercoolers should be drained properly. The coolers act as a condenser. The condensate will buildup if its not drained regularly. Water will be carried over to the high-pressure cylinders (downstream of the intercooler) if it has not been drained. Automatic or timed drain traps will confirm that the intercooler, aftercooler, and air receiver have been drained properly. The mineral content of water will also buildup in water-cooled intercoolers and aftercoolers. This will reduce the effectiveness of this equipment. Thus, these coolers should be inspected regularly. All the deposits should be removed to restore their effectiveness.

Dirt will deposit on the outside of the core section of air-coolers, intercoolers, and radiators. This will reduce the effectiveness of this equipment. The core section of this equipment should be cleaned regularly. The cleaning of this equipment from dust involves blowing air in the opposite direction to the normal flow. A solvent should be used if the dirt is contaminated with oil. The solvent should be allowed to soak for a while. The contamination should then be blown clean.

18.12 Cleaning

The compressor outside casing should be kept clean. Buildup of dirt and oil on the outside of the casing will form a thermal insulation. This insulation prevents the dissipation of heat to the atmosphere. The dirt which accumulates on the surface of the compressor will eventually enter into the internal components of the compressor. The outside surfaces of the compressor should be kept clean to reduce the operating and maintenance cost.

18.13 Packing

Double-acting compressors employing piston rods have oil-stop-head packing. These packings require periodic inspection. The oil-stop-head packing (Fig. 18.5) consists of a set of metallic scraper rings. This packing requires little maintenance. This is because they are designed to scrape oil off the rod only. This packing will fail if the piston rod becomes damaged. New scraper rings should never be placed on a piston rod that is nicked, scratched, or worn.

Figure 18.6 illustrates a cylinder-head packing. The rings of this packing are self-adjusting. The design and quantity of these packing rings depend on the type of gas being compressed and the discharge pressure. Metallic packing requires little maintenance. However, the part of the piston rod that passes through the packing should be inspected. The packing must be removed and inspected for embedded material if any scratches are found on the piston rod. The packing should not be disturbed if it does not leak or score the rod.

This is because the metal rings are self-adjusting for the slight wear that occurs during operation. The service check chart shown in Table 18.1 lists the common causes of compressor malfunction.

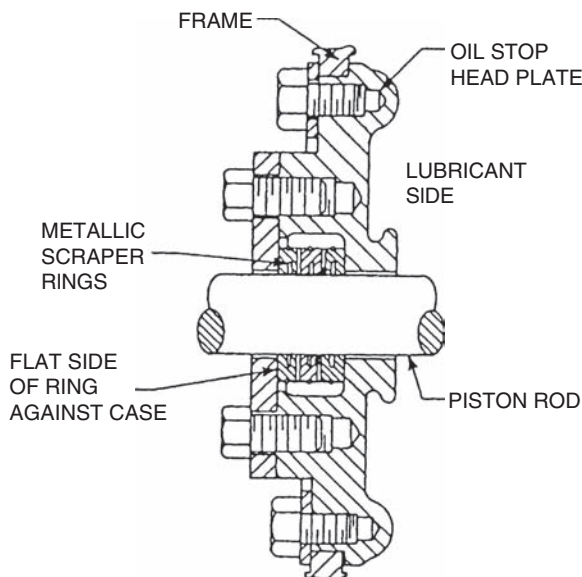


FIGURE 18.5 Oil-stop-head packing. (Source: Reprinted with permission from Higgins L., *Maintenance Engineering Handbook*, 5th ed., McGraw-Hill, New York, 1995, with permission from McGraw-Hill.

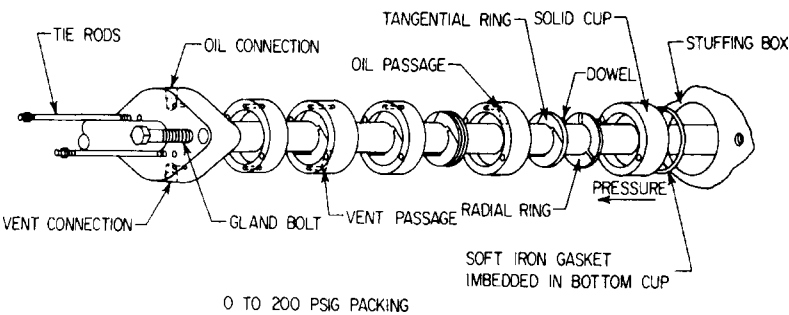


FIGURE 18.6 Cylinder-head packing. (Source: Reprinted with permission from Higgins L., *Maintenance Engineering Handbook*, 5th ed., McGraw-Hill, New York, 1995, with permission from McGraw-Hill.)

1. Low oil pressure a. Low oil level b. Plugged oil-pump suction strainer c. Leaks in suction or pressure lines d. Worn-out bearings e. Defective oil pump f. Dirt in oil-filter check valve g. Broken oil-filter-check-valve spring h. Oil-pressure-bypass leaks
2. High oil pressure a. Plugged oil-pressure lines b. Defective oil-filter mechanism c. Excessive spring tension on filter check valves d. Excessive spring tension on oil-pressure adjusting mechanism
3. Incorrect delivery of mechanical lubricator a. Dirty or gummed pumps b. Broken spring in check valve at cylinder c. Leak in lines or sight feed d. Low oil level e. Plugged vent in lubricator reservoir

Note: Remember to read the instruction book carefully and to keep it and the parts list in an accessible place so that when information to make adjustments and repair is needed, shutdown time can be held to a minimum.

TABLE 18.1 Service Check Chart, Mechanical Parts

4. Overheated cylinder a. Insufficient cooling water b. Scored piston or cylinder c. Broken valves or valve springs d. Excessive carbon deposits e. Packing too tight f. Insufficient lubrication g. Corroded or clogged cylinder water passages
5. Water in cylinders a. Leaking head gaskets b. Cracked cylinder or head c. Condensate caused by too much cooling water or inoperative trap
6. High intercooler pressure a. Broken or leaking high-pressure valves b. Defective gage c. Defective or leaking valve-seat gaskets
7. Low intercooler pressure a. Broken or leaking low-pressure valves b. Leak in intercooler c. Piston-rod-packing leaking
8. Knocks a. Excessive carbon deposits b. Scored piston or cylinder c. Defective lubricator d. Foreign material in cylinder e. Piston hitting cylinder head f. Loose piston or piston pin g. Burned-out or worn rod bearings h. Loose main bearings i. Scored crosshead or crosshead guides j. Loose valve set screws
9. Scored cylinder, liner, or piston a. Foreign material b. Dirty or inefficient air filter c. Lack of lubrication d. Too much and too cold cooling water causing excess condensate and washing out lubrication e. Excessive heat f. Plugged water jackets

TABLE 18.1 Service Check Chart, Mechanical Parts (*Continued*)

10. Broken valves and springs a. Too much condensation, causing rust b. Carbon deposits c. Foreign materials not removed by air filter d. Incorrect assembly e. Acid condition prevailing at location of suction air inlet
11. Control trouble a. Suction-valve unloader stuck open or closed b. Pressure switch defective c. Solenoid burned out d. Foreign material in three-way valves e. Excessive vibration of control f. Voltage drop or loss of power g. Plugged air line or strainer h. Incorrect voltage or cycle
12. Incorrect operation of suction-valve unloaders a. Leaks in unloader line b. Foreign material in guides or seats c. Worn plungers d. Leaking or ruptured diaphragms and O-rings e. Broken springs f. Manual shutoff partly closed g. Wrong pressure-switch settings

TABLE 18.1 Service Check Chart, Mechanical Parts *(Continued)*

18.14

Bibliography

Higgins, L. R., *Maintenance Engineering Handbook*, 5th ed., McGraw-Hill, New York, 1995.

Diaphragm Compressors

19.1 Introduction

Diaphragm compressors are used in applications where leakage is not tolerated. These units have a piston with piston rings. The piston moves a volume of hydraulic oil on its upward stroke. The movement of the oil bends a set of diaphragms upward. This compresses the gas. The seals of this compressor are static. They are located at the ends of the bellows. The compression is achieved without any gas leakage.

19.2 Theory of Operation

Figure 19.1 illustrates the following components for a diaphragm compressor:

- Typical drive arrangement
- Crankcase
- Hydraulic system
- Compression head

The prime mover for a diaphragm compressor is usually an electric motor. This motor drives a flywheel using flexible belts. The flywheel rotates the crankshaft. This crankshaft moves the crosshead in a reciprocating motion through a connecting rod on an eccentric journal. The connecting rod is attached to a crosshead through a wrist pin. This crosshead moves in a cylinder.

The crosshead is connected to a hydraulic piston. This piston moves in a hydraulic cylinder. Piston rings seal the oil above the hydraulic piston. The piston moves a fixed volume of hydraulic oil up and down. This oil moves a set of diaphragms. The upward movement of the diaphragms against the gas head compresses the gas.

The piston rings constitute the only dynamic seal in a diaphragm compressor. The following are the advantages of this dynamic seal over the one used in a reciprocating piston compressor:

- The dynamic seal in a diaphragm compressor acts on the oil side. Thus, there is no gas leakage associated with the dynamic seal.
- The dynamic seal in a diaphragm compressor is lubricated with oil.

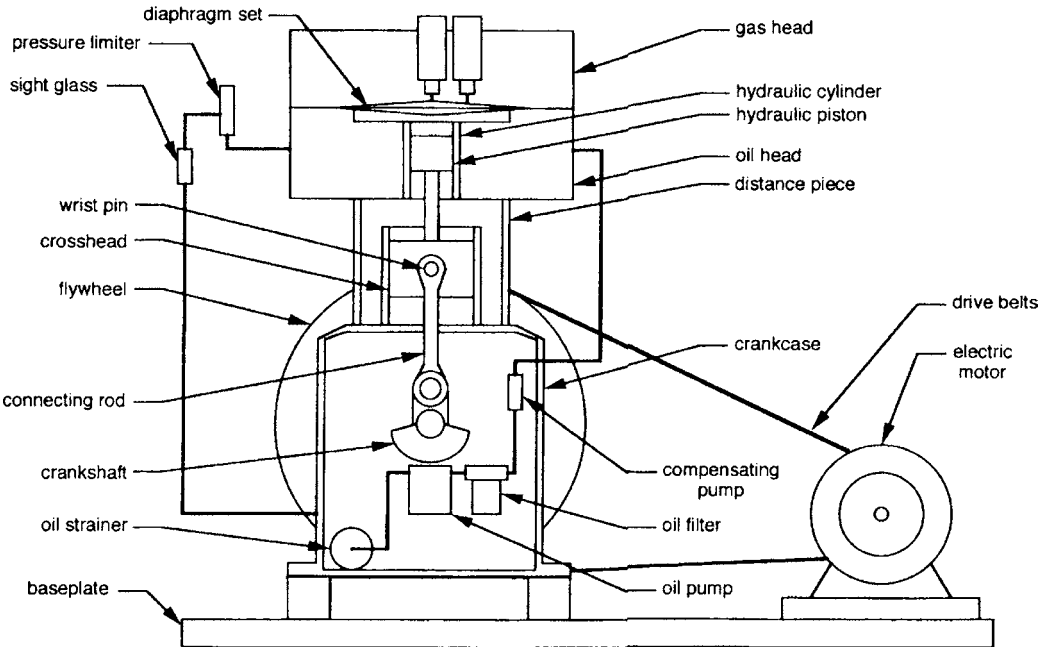


FIGURE 19.1 Main components of a diaphragm compressor.

The hydraulic oil used in a diaphragm compressor provides the following functions:

- Bends the diaphragms up and down
- Enhances the cooling of the area between the diaphragms and the hydraulic piston

The bottom of the crankcase acts as a reservoir for the lubricating oil. The oil enters the circuit through a strainer. This oil is cooled through a water-cooled heat exchanger in some applications. The flow leaving the oil strainer or the heat exchanger enters the main oil pump. This flow is then discharged through a filter. The oil leaving the filter splits into two streams. Most of the oil flows through the first stream. This oil lubricates the following components:

- Crankshaft bearings
- Connecting rod journals
- Wrist pins
- Sliding surfaces of the crosshead

The remaining oil flows through the compensating circuit. The flow in this circuit is a small portion of the main oil flow. The compensating circuit provides makeup for any oil that leaked past the piston rings on the hydraulic piston. There are check valves

upstream and downstream of the compensating pump. The oil flowing through the low-volume reciprocating compensating pump enters into the oil head. The hydraulic oil pushes the diaphragm set into full contact with the gas head during every upward stroke of the hydraulic piston. The volume of the oil located between the hydraulic piston and the diaphragm set would decrease if the leakage past the hydraulic piston rings were not made up. Thus, the diaphragm set will not be able to establish full contact with the gas head on the upward stroke of the hydraulic piston. This will result in a decrease in the compressor performance. Therefore, the compensating circuit plays a major role in maintaining the performance of the compressor.

The gas discharge pressure is controlled by a valve mounted on the oil head. This valve regulates the flow of makeup oil. It maintains the proper oil pressure inside the oil head. This valve also allows any excess oil to return to the crankcase. It is known as the hydraulic pressure limiter. This valve is adjusted to establish the desired gas discharge pressure in the compression head. The hydraulic pressure limiter opens when the peak oil pressure (the limiter pressure) is reached. This valve is closed by a spring.

The compression head consists of the following:

- An oil head bolted to a gas head
- A diaphragm set
- Seals
- Suction valve(s)
- Discharge valve
- Valve retainer and bolting

The hydraulic oil is in the oil head. The gas being compressed is in the gas head. The diaphragm separates the oil from the gas. The suction and discharge valves open and close due to differential pressure. They are self-actuating. These valves allow the gas to enter and leave the head at the proper instances.

Figure 19.2 illustrates the variations of pressure inside the oil head during a typical revolution. The hydraulic piston is at the top dead center (TDC) at point A. This point represents the maximum oil and gas pressure. The oil pressure drops rapidly when the piston moves downward. The gas pressure on the other side of the diaphragm set varies proportionally to the oil pressure. The discharge valve closes when the discharge pressure becomes higher than the internal gas pressure in the compression head. The pressure of the remaining gas in the compression head drops from the discharge pressure to suction pressure as the piston continues to move downward. The suction valve opens when the gas pressure in the compression head becomes slightly lower than the suction pressure (point B). Fresh gas starts to enter the compression head through the suction valve at this stage.

The compensating pump injects a small amount of oil into the oil head when the piston approaches the bottom dead center (BDC). This is done to make up for the piston ring leakages. The fresh gas flow into the compression head stops when the piston reaches the BDC. This is shown as point C in Fig. 19.2. The diaphragms are at their most extended position into the oil head at this point.

The suction valve closes as the piston starts its upward stroke toward the gas head. This occurs when the internal gas pressure exceeds the external gas pressure. The gas is now trapped inside the compression head. The oil pressure and the gas pressure increase

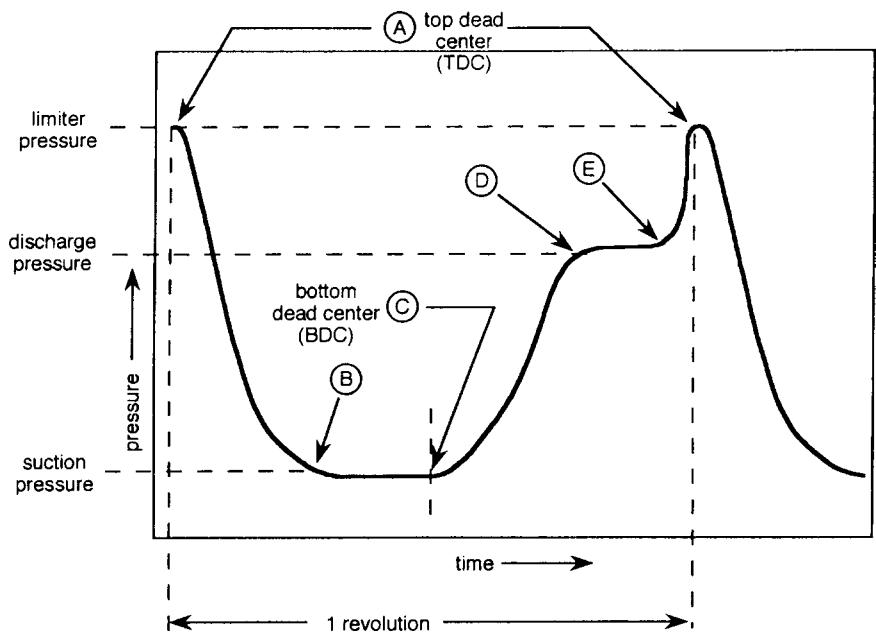


FIGURE 19.2 Internal pressures during compression.

simultaneously as the piston continues its upward stroke toward the gas head. The discharge valve opens when the pressure inside the compression head exceeds the discharge pressure. This occurs at point D. The gas flows from the compression head into the discharge piping. The discharge valve closes at point E. This occurs when the diaphragm set contacts the gas head. However, the piston continues to move upward until it reaches the TDC. This occurs at point A. Thus, the pressure in the oil head continues to increase until the TDC is reached. The hydraulic pressure limiter opens at point A. This allows a small amount of oil to flow from the oil head back to the crankcase. The next cycle begins at point A. The piston starts to travel downward. The hydraulic pressure limiter closes. This maintains the pressure in the oil head. The same activities are repeated again during the next cycle.

19.3 Compressor Design

The model number of diaphragm compressors normally denotes the following:

- Basic crankcase (with a given stroke and rod load)
- Configuration
- Maximum pressure rating of the oil head and gas head
- Stroke from 3.18 cm (1.25 in) to 22.9 cm (9 in)

However, most strokes are between 6.35 cm (2.5 in) and 12.7 cm (5 in). The configuration of diaphragm compressors varies significantly. Single-stage units are used

commonly in many industries. The compressor head can be mounted either vertically or horizontally. Two-stage designs are also used in a wide variety of applications. They can be constructed in "L," "V," or horizontally opposed styles. Three- and four-stage units are not used commonly in industry. The pressure rating of these compressors varies from vacuum conditions to more than 306 MPa (45,000 psig). However, most units operate between 1 and 100 MPa (150 psig and 15,000 psig).

Electric motors are used commonly to drive these compressors. The rating of these motors is normally between 1 and 200 hp. The power is transmitted to the motors using belt drives. A flywheel is mounted on the shaft of these units. This flywheel provides rotational inertia for these compressors. This feature reduces the motor current during transient conditions.

Proper oil filtration is essential for successful operation of these units. Bearing damage will occur if contaminated oil is used for lubrication. The diaphragm life will decrease significantly as well if the oil contains impurities or metal particles. The oil is kept clean by passing it through a strainer located at the suction of the oil pump. The oil is discharged through a filter located downstream of the oil pump. This filter is normally rated at 25 μm .

Diaphragm compressors are normally lubricated using a shaft-driven gear pump. This pump is driven by any of the following two methods:

1. Direct coupling onto the crankshaft
2. A gear attached to the crankshaft

An auxiliary lubricating pump is provided in some units. This pump is normally driven by an electric motor. Relief valves are installed at the discharge of these lubricating pumps. The overflow from these relief valves is returned to the pump in the crankcase. An oilheater is normally required for these compressors in low-temperature applications. An immersion-type heater is generally used for this service. This heater is installed directly into the crankcase. Oil coolers are also used in these compressors as well. The oil cooler is installed normally downstream of the oil pump. Some compressors operate in a wide range of operating temperatures. These units use both an oil heater and an oil cooler.

The compensating pump is driven by an eccentric. This eccentric is mounted on the crankshaft. There is a small plunger inside the compensating pump. This plunger is sealed to the pump body by either of the following:

- Piston rings
- Tightly toleranced fit

Check valves are installed at the suction and discharge ports of the pump. They prevent backflow. The hydraulic pressure limiter (also known as limiter) determines the peak hydraulic pressure. This limiter opens when the peak oil pressure is reached. It has the following characteristics:

- Closes by a spring action
- Its opening pressure can be externally adjusted
- Rugged design
- Ability to open and close 500 times per minute

The selection of suitable oil is critical for proper operation of diaphragm compressors. Standard hydraulic or general purpose oil is used for most applications. The ISO viscosity range of this oil varies between 46 and 100 cSt. This oil contains normally anti-wear and antifoam additives. Synthetic lubricants are used in some applications. These lubricants include halocarbon oils. Applications involving oxidizers such as oxygen and fluoride normally use halocarbon oils. Care must be taken when synthetic oils are used to ensure that all the components in the hydraulic circuit are compatible with them.

Figure 19.3 illustrates a typical cross-sectional view of a diaphragm compressor head assembly. The piston rod is shown at the bottom end of this assembly. This piston is attached to the crosshead. The piston moves in a hydraulic cylinder. Rider rings guide the piston. Piston rings seal it. The piston rings were made exclusively of cast iron. However, they are now made commonly of specialized filled plastic materials. A series of holes and grooves distributes the oil evenly underneath the diaphragms during the upward movement of the piston. A shallow cavity limits the deflection and stresses in the diaphragm set. This cavity is on the side of the oil head facing the diaphragms. There is also a cavity on the side of the gas head facing the diaphragms. This cavity matches the one in the oil head. This cavity limits the deflection of the diaphragms.

Many diaphragm compressors are located in corrosive environments. These environments include the following:

- Salt spray from the ocean
- A caustic atmosphere in a chemical plant

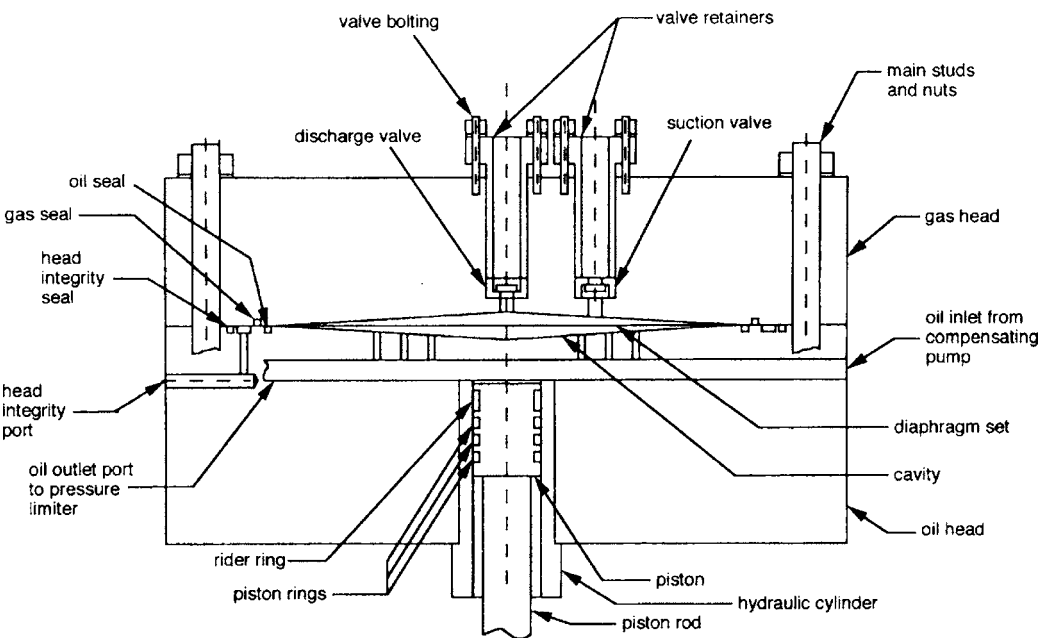


FIGURE 19.3 Diaphragm compressor head assembly.

Thus, plated or coated bolts are used in the compressor head. This feature prevents rust. Cadmium or zinc plating was used previously on these bolts. However, modern designs use PTFE coatings. These coatings provide the following two features for the bolts:

- Lubricity
- Corrosion resistance

The diaphragms are flat, circular pieces of sheet metal. They have a smooth finish. Diaphragm compressors normally employ a stack of three diaphragms. Each diaphragm has a different purpose. The gas diaphragm must have a high resistance to corrosion. This is because it is in contact with the process gas. The middle diaphragm has holes. These holes conduct any leakage from a failed gas or oil diaphragm to the periphery. These leakages are monitored to identify any problem with the compressor. The oil diaphragm transmits the hydraulic pressure to the other diaphragms. This diaphragm rarely has corrosion problems. The interfaces between the diaphragms should be lubricated. This is due to the tiny differential displacement between the three diaphragms. The middle diaphragm is coated in some applications with a dry lubricating film. Some units employ diaphragms made from dissimilar metals. This is done to reduce the wear between the diaphragm surfaces.

The seals used in most diaphragm compressors are elastomer o-rings. The function of these seals is critical for proper operation of these compressors. These o-rings are available in a wide variety of materials. The o-ring material should be selected to suit the process gas.

Figure 19.4 illustrates a head integrity system used in a diaphragm compressor. This system is also known as a leak detection system. It can detect any of the following leakages:

- Failed gas diaphragm
- Failed oil diaphragm

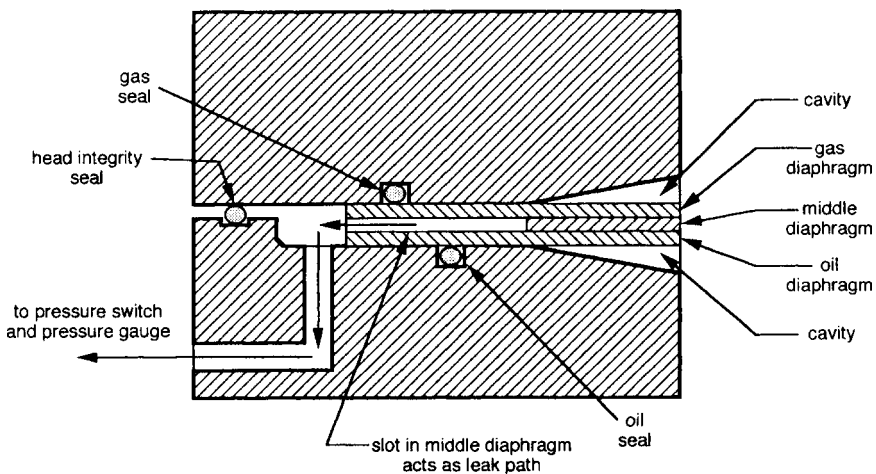


FIGURE 19.4 Head integrity system.

- Leaking gas seal
- Leaking oil seal

The gas or oil will leak into the slot located in the middle diaphragm upon a diaphragm failure. The head integrity seal prevents the leakage of the gas or oil to the atmosphere. The leaking fluid is conducted to an external port. The head integrity system monitors the leak from this external port. The following are the components of the head integrity monitoring system and their function:

- *Relief valve.* This valve prevents over-pressurization of the head integrity monitoring system.
- *Pressure gauge.* This component allows the verification of the leak.
- *Pressure switch.* This component shuts down the compressor when a predetermined pressure is reached in the head integrity monitoring system.
- *Manual vent valve.* This valve vents the head integrity monitoring system. It also allows the resetting of the pressure switch.

Most valves used in diaphragm compressors are self-actuated. These valves open by the flow of suction or discharge gas. They close and seal by differential pressure. The springs used in these valves perform the following functions:

- Control the valve seal
- Damp out unwanted seal movement

The following sealing elements are used in these valves:

- Flat discs
- Guided poppets
- Balls

The valve seals are usually elastomer o-rings, metal gaskets, or metal seal rings.

19.4 Materials of Construction

Most modern diaphragm heads are made from a carbon steel plate or forgings. The oil heads are typically made from SA516 grade 70. The gas heads are made from a carbon steel plate in applications involving inert gases and from a stainless steel plate such as SA240 type 304 or 316 for applications involving corrosive gases. The gas heads are also made from a variety of materials for severe environments. These materials include monel or alloy 20.

The diaphragm material is made from a carbon steel sheet for applications involving inert gases. However, most diaphragms are made from a high tensile-strength stainless steel. This material includes stainless steel type 301, 306, or 17-4PH. Monel or alloy 20 is used in applications requiring higher corrosion resistance.

Diaphragm compressor valves are normally made from a corrosion-resistant material such as alloy steels or stainless steel series 300 or 400. The gas-contacting

parts of the valves are usually made from monel or alloy 20. The sealing elements of the valves are made in most applications from specialized plastics such as PEEK (polyether ether ketone) or Vespel. The valve springs are commonly made from stainless steel or inconel.

19.5 Accessories

Diaphragm compressors normally have a suction filter. This filter removes dirt and debris entrained in the incoming gas to the compressor. The compressor performance will deteriorate quickly if the filter malfunctions. Modern filters are made of pleated stainless steel mesh. They normally have a 10- μ m rating. Applications requiring online change of the filter use double suction filters with changeover valves. This feature reduces the downtime of the compressor.

Accumulators are recommended for applications that do not have a significant volume of gas in the suction and discharge piping. These accumulators smoothen the compressor operation. A separator and trap are required for applications where the gas contains moisture. The separator and trap should be installed at the following locations:

- Suction to each compression stage
- Any point where condensation may occur (especially downstream of a cooler)

Liquid droplets entrained with the gas can damage the diaphragms and the compression heads if they are allowed to enter the compressor. The gas should be cooled after each compression stage. A variety of heat exchange designs can be used to perform this function. Intercooling of the gas provides the following advantages:

- Protection of the compressor seals
- Protection of the equipment located downstream of the compressor
- Increase in the compressor efficiency

Screwed joints are used for most compressor piping. However, welded joints are used in applications where the gas has any of the following characteristics:

- Flammable
- Toxic
- Pyrophoric (gas that combusts spontaneously upon exposure to air)

Socket welds are used for most compressor piping. However, butt welds are used to meet the most stringent requirements in applications where radiography can confirm the soundness of a joint.

The compressor performance should be monitored regularly using the following instrumentation:

- Pressure gauge at the suction of each compression head.
- Pressure gauge at the discharge of each compression head.

- Pressure switches. These switches can be installed at the suction or discharge of the compressor. They are not normally installed at the interstage sections of the compressor.
- Discharge temperature switch or transmitter.

Diaphragm compressors require overpressure protection. A relief valve or rupture disc must be installed at the discharge of each compression stage. Serious damage to personnel and equipment could occur if this equipment malfunctions.

The capacity of diaphragm compressors is controlled using a gas bypass line. This line returns the excess flow from the compressor discharge to its suction. A control system is normally used with these compressors. This control system maintains a constant pressure at the compressor discharge. An air-actuated control valve receives a signal from this control system. This valve throttles the flow from the compressor discharge to maintain a constant compressor discharge pressure. On-off control is also used in some applications. However, the compressor should not be restarted more than a few times per hour. This is done for the following reasons:

- Frequent compressor restarts can damage the compressor.
- Frequent motor restarts increase the power cost significantly. This is because the starting current (also known as inrush) of the motor is 6 to 8 times larger than the operating current.

Variable speed drives are not commonly used with diaphragm compressors. This is because the shaft-driven oil pumps will not be able to maintain the required oil pressure and flow. Mechanical valve unloaders are not normally used with diaphragm compressors. This is because these unloaders cannot be installed directly over the suction and discharge valves in these compressors. The following instrumentation is used to protect the hydraulic system of the compressor:

- Oil pressure gauge
- Oil pressure switch
- Oil level switch (used in some applications)

These switches can shut down the compressor when the variable drifts beyond a predetermined value. Water flow switches are also used with these units. They protect the compressor from an accidental loss of coolant.

19.6 Cleaning and Testing

The components of a diaphragm compressor must be carefully cleaned to maintain a reliable operation. Diaphragm failure will occur due to buildup of dirt and debris on either side. The manufacturers of these compressors have established high standards of cleanliness to prevent diaphragm failure. Applications involving oxygen or oxidizer service have even higher standards of cleanliness. The gas side components of these compressors must be free of both debris and oil. The presence of a slight amount of these contaminants can result in a dangerous reaction with the process gas. The crankcase and all hydraulic equipment of the compressors used in these applications must

also be free of hydrocarbon oils. This is because ignition can occur if there is any contact between the oxidizer and the hydrocarbons.

The manufacturer of a high-quality diaphragm compressor will complete a performance test before shipping the unit to the user. The following activities are performed during the test:

- An inert gas (typically nitrogen) is used to simulate the process gas.
- The actual flow rate and discharge pressure are measured.
- The entire compressor system is checked thoroughly for leaks. This includes the gas side, the hydraulic side, and the cooling circuits.

All the required nondestructive examination is completed before shipment. This includes the following:

- Liquid penetrant tests
- Radiographic tests for the compressor piping

The purchaser of the compressor should specify these tests in writing. The manufacturer should provide the results of these tests to the compressor purchaser.

19.7 Applications

Diaphragm compressors are used in applications where ordinary piston compressors could experience the following problems:

- Gas leakage
- Gas contamination
- Discharge pressure limitations

Diaphragm compressors are ideally suited for the following applications:

Type of Gas	Examples
Toxic	Boron trifluoride
Corrosive	Fluorine
Flammable	Hydrogen
Pyrophoric	Silane
Oxidizer	Chlorine
Radioactive	Uranium hexafluoride

Diaphragm compressors provide also cost-effective service in applications that require the following three:

1. Low contamination
2. High discharge pressure
3. High discharge temperature

Diaphragm compressors are used commonly also in the industrial gas production and distribution. These compressors are used to fill cylinders with the following three gases:

1. Helium
2. Nitrogen
3. Argon

Diaphragm compressors provide a reliable, non-contaminating method to compress these gases in this industry. Diaphragm compressors are also used in the following applications:

19.7.1 Automotive Air Bag Filling

Diaphragm compressors are used to compress a mixture of gases to a high pressure in gas canisters. A leak test is performed on the gas canisters. The compressed gas remains in these gas canisters. This gas is used to inflate the air bag during an accident.

19.7.2 Petrochemical Industries

Diaphragm compressors are used in the petrochemical industry. They pressurize a wide variety of gases and distribute them throughout the plant to various processes.

19.8 Limitations

Diaphragm compressors operate normally between 300 and 500 rpm. Some of these compressors operate at a speed as low as 100 rpm. However, provisions must be made in these applications to maintain adequate oil pressure. Some modified diaphragm compressors operate at a speed of 750 rpm.

The following are the limitations of diaphragm compressors:

- Suction pressure: From 2 cm (0.79 in) Hg absolute to 3 MPag (5000 psig)
- Discharge pressure: Up to 306 MPag (45,000 psig)
- Flow rate: From 1.7 standard m³/h (1 scfm) to 1189 standard m³/h (600 scfm)

Diaphragm compressors may not be suitable for some applications. This is due to the large size and weight of the gas and oil heads. The diameter of these heads can reach 101.6 cm (40 in) in some low-pressure applications. This can limit their installation in tight quarters. The weight of these heads can exceed 909 kg (2000 lb). This can increase the difficulty in removing the heads for maintenance.

19.9 Installation and Maintenance

The foundation design of a diaphragm compressor should account for the vibration caused by unbalanced forces and moments. These values should be provided by the manufacturer of the compressor. Pulsation dampeners should be considered to avoid acoustic resonance in applications involving large flow rates. Guards should also be

considered to prevent contact with the discharge lines in applications where personnel have routine access to the compressor. These discharge lines are normally very hot. They can cause burns.

The primary goal of a diaphragm compressor is to provide contamination-free compressed gas. Filters are required at the suction of the compressor. They prevent particulates from entering the compressor. Separators are also required in applications where liquids could be entrained with the gas entering the compressor. These separators should also be installed at the compressor suction. All the compressor components should be cleaned thoroughly during any maintenance work. These components must be very clean before the compressor is reassembled following maintenance work. This is essential to ensure successful operation of the compressor.

The maintenance plans for a diaphragm compressor should include the following:

- Lifting equipment to remove the gas head and oil head
- Ability to clean all the external surfaces of the components on the gas and oil side
- Adequate lighting
- Accessibility to the compressor

Figures 19.5 through 19.7 illustrate diaphragm compressors operating at different pressures.

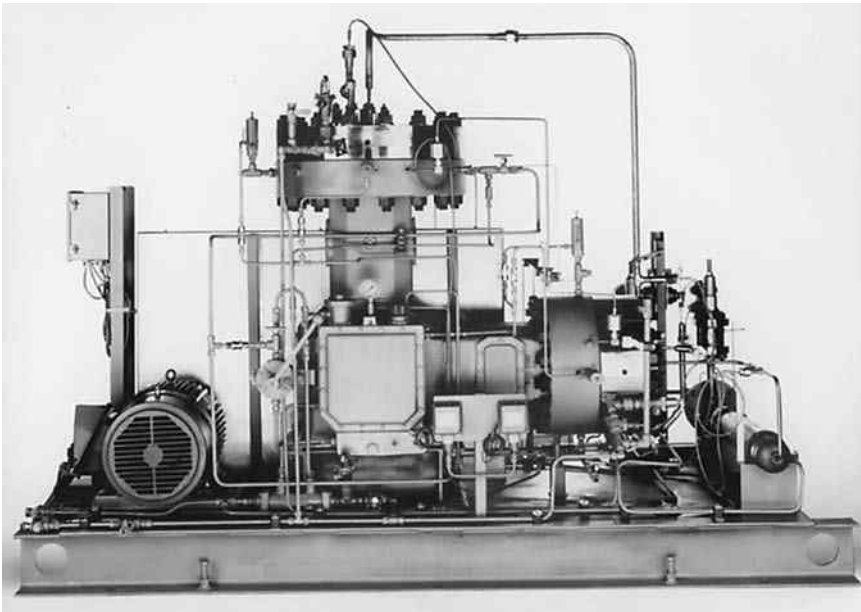


FIGURE 19.5 Burton Corblin two-stage 13.6-MPa (2000-psi) diaphragm compressor.



FIGURE 19.6 Burton Corblin two-stage 40.4-MPa (6000-psi) diaphragm compressor.

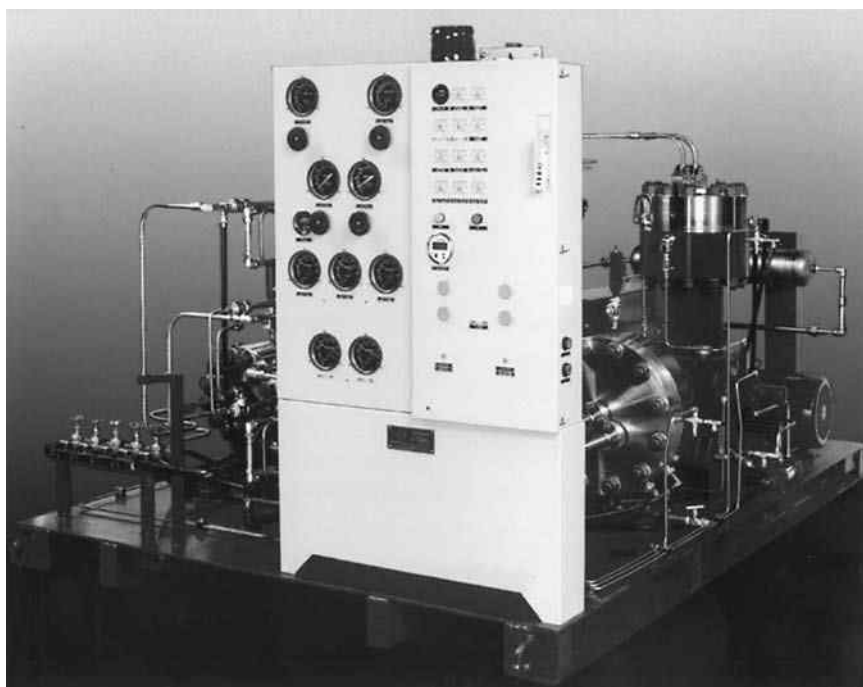


FIGURE 19.7 Burton Corblin two-stage 54.4-MPa (8000-psi) diaphragm compressor.

19.10 Diaphragm Compressor Specification

A specification is required to procure a diaphragm compressor. This specification should include the following information:

- Range of flow rate flowing through the compressor
- Physical and chemical properties of the gas being compressed
- Characteristics of the gas such as corrosiveness, etc.
- The amount of liquid and contaminants entrained with the gas
- Range of suction pressure and temperature
- Range of required discharge pressure and temperature
- Classification of the electrical area
- Availability of power supply, e.g., voltage level, etc.
- Availability of power required for the control system
- Site conditions, including ambient pressure, temperature, and humidity
- Preferred compressor materials
- Limitations on the size of the compressor
- Applicable national and international standards

A compressor designed for one site may be inadequate for the same application in a different country. The following are the standards invoked in the United States:

- American Society of Mechanical Engineers (ASME) code for pressure vessels
- Piping code ASME B31.3
- National Electrical Code for wiring, protection, etc
- American Petroleum Institute (API) 618 for compressor construction

The purchaser of the compressor should also include in the specification any in-house standards that apply. This includes the in-house standards for motors, pressure vessels, and piping.

19.11 Bibliography

Hanlon, P. C., *Compressor Handbook*, McGraw-Hill, New York, 2001.

This page intentionally left blank

Rotary Screw Compressors and Filter Separators

20.1 Twin-Screw Machines

Figures 20.1 to 20.4 illustrate rotary screw compressors. The gas inlet and discharge nozzles are at the opposite ends. Figure 20.5 illustrates these machines having three-, four-, and five-lobe rotors.

20.1.1 Compressor Operation

The rotary screw compressor is a positive displacement machine. It produces a uniform, non-pulsating, continuous flow. Figure 20.6*a* illustrates the suction phase of the compressor. A vacuum is created when the lobe of the male rotor begins to un-mesh from the female rotor. The gas is drawn from the inlet port into this vacuum. The size of the interlobe space between the male rotor and the female rotor continues to increase as the rotors turn. The gas continues to flow into the interlobe space. The entire length of the interlobe space from end to end becomes open just before the inlet port closes. The male and female rotors are completely un-meshed at this stage. The interlobe space is thus completely filled with gas.

The transfer phase follows the suction phase. It occurs before the compression phase. It is a transitional phase. The trapped gas in the interlobe space is isolated from the inlet and outlet ports in this phase. The gas is transported radially through a specific number of degrees of angular rotation. The gas remains at suction pressure in this phase.

Figure 20.6*b* illustrates the compression phase. Further rotation meshes a male lobe with the gas-filled interlobe space. This male lobe is different from the one that was disengaged previously. This is because there are four lobes and six interlobe spaces (4:6 relationship). The gas is compressed from the suction end toward the discharge port. The meshing point of the lobes moves axially from the suction to the discharge end. Thus, the volume of the gas trapped in the interlobe space decreases gradually. This leads to a gradual increase in pressure.

Figure 20.6*c* illustrates the discharge phase. The outlet port opens at the discharge pressure. The compressed gas is discharged from the compressor through the outlet port. Gas is drawn into the interlobe space at the suction as the gas that is trapped in the previous interlobe space moves axially toward the discharge end. Thus, the operating phases of the compressor cycle are repeated.

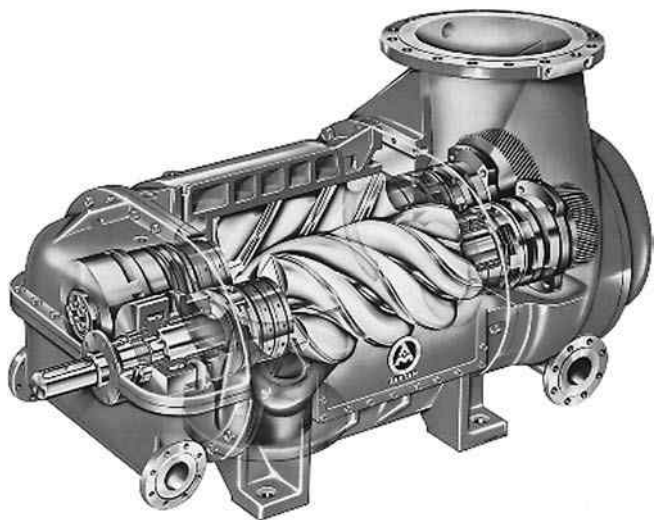


FIGURE 20.1 Rotary screw compressor (double-helical-screw machine). (Source: Aerzen USA Company, Coatesville, Pa.)



FIGURE 20.2 Small packaged rotary screw compressor. (Source: Aerzen USA Company, Coatesville, Pa.)

20.1.2 Applications of Rotary Screw Compressors

Rotary screw compressors are the most suitable choice for air compression in industry. This is due to their simplicity and high reliability. The following are the types of these compressors:

- Oil-free
- Oil-wetted

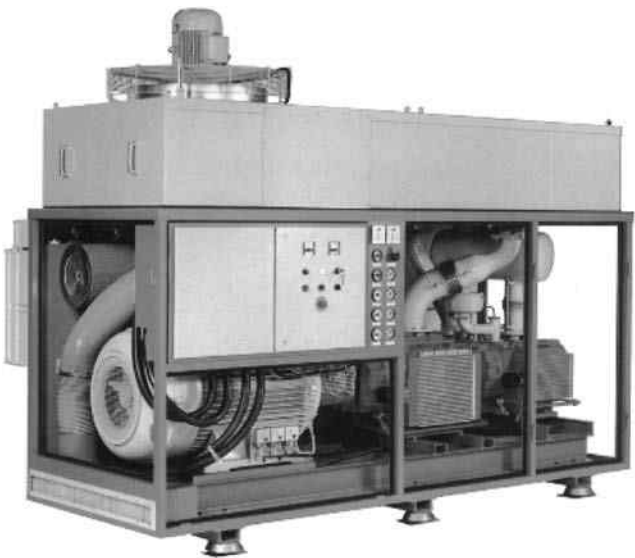


FIGURE 20.3 Medium-sized rotary screw compressor package. (Source: Aerzen USA Company, Coatesville, Pa.)

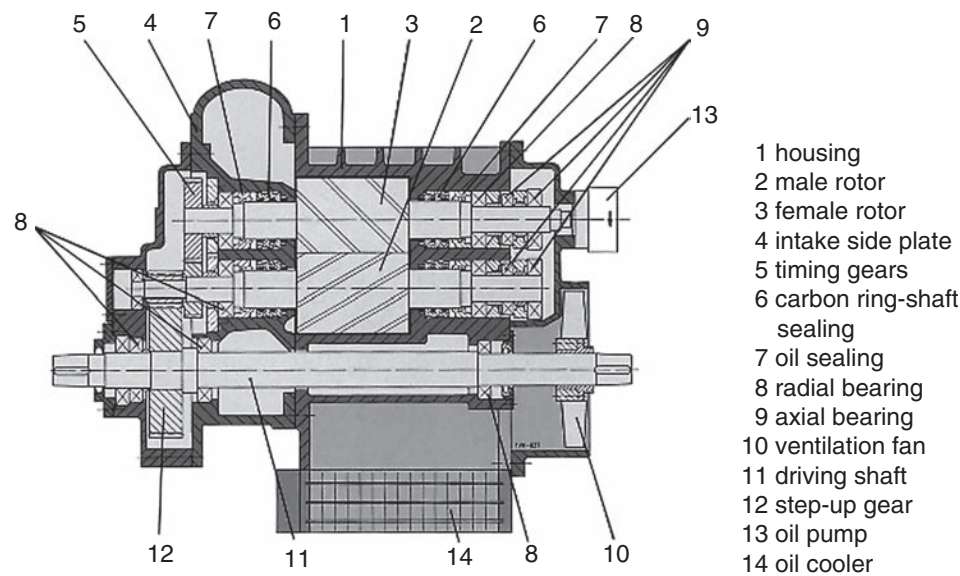


FIGURE 20.4 Oil-free rotary screw compressor with integral step-up gears. (Source: Aerzen USA Company, Coatesville, Pa.)

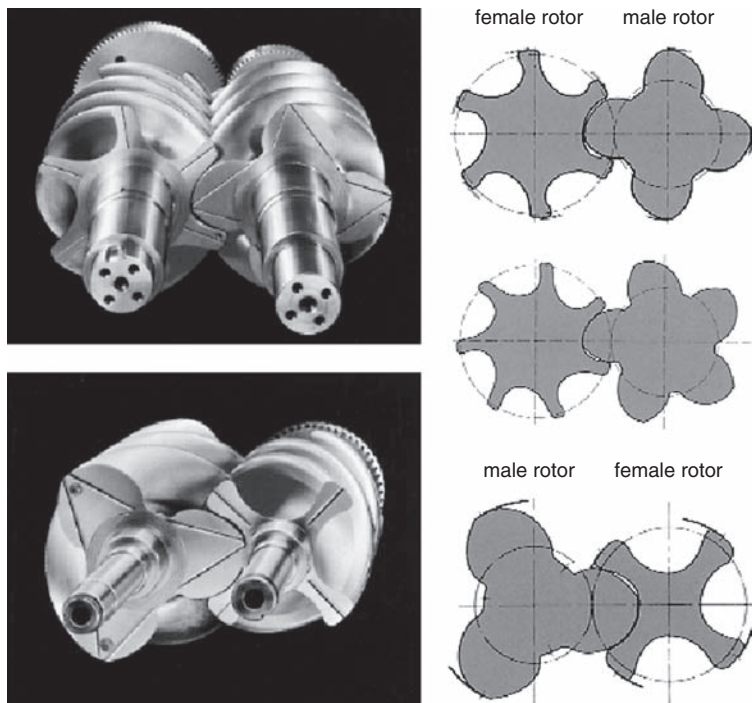


FIGURE 20.5 Typical screw compressor rotor combinations. (Source: Aerzen USA Company, Coatesville, Pa.)

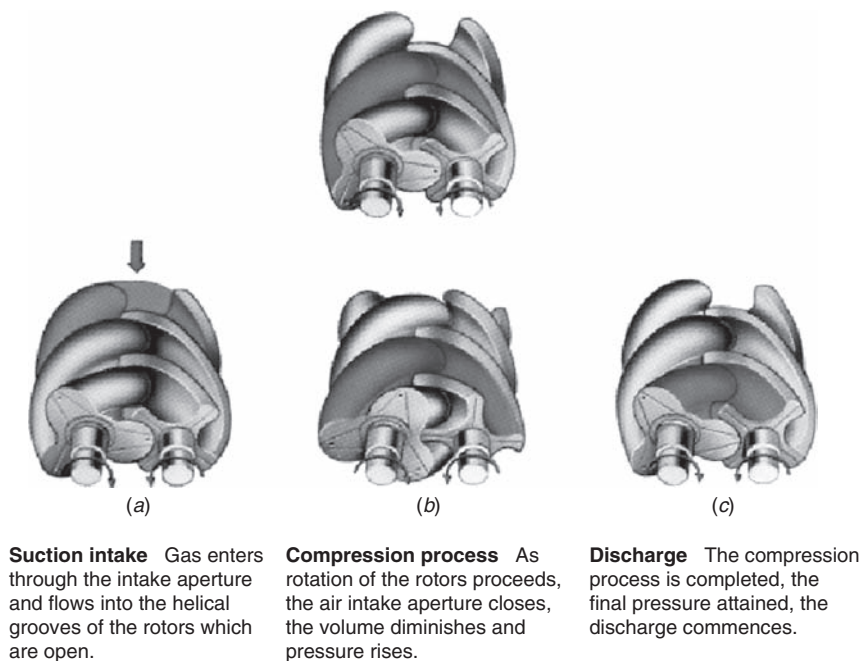


FIGURE 20.6 Working phases of rotary screw compressors. (Source: Aerzen USA Company, Coatesville, Pa.)

Rotary screw compressors are equally suited for applications involving a wide variety of gases. These gases include the following:

- Ammonia
- Natural and synthetic pipeline gases
- Flare gas mixtures
- Swamp and biomass gases
- Coke oven or coal gas
- Flue gas
- Crude or raw gas
- Sulfur dioxide
- Nitrous oxide
- Hydrogen

Rotary screw compressors are being selected today to substitute other compressor types in many applications. This is due partially to the modern sealing and liquid injection technology used in these machines. The sophisticated contour machining and improved metallurgy employed in these compressors has allowed single- or multistage units to be used in the following applications:

- Suction volumes from 300 to 600,000 std m³/h (176–35,310 scfm). The standard conditions are 101.3 kpa_(abs) and 15.56°C (14.7 psia and 60°F).
- Discharge pressure up to 4 MPa (580 psi).
- Vacuum as low as 0.09 bar_(abs) (1.3 psia).

20.1.3 Dry and Liquid Injected Compressors

The following are the two types of rotary screw compressors used in industry:

1. Dry machines
2. Wet liquid-injected units

The following are the types of liquid-injected rotary screw compressors:

- Oil-injected units
- Machines using other liquids

Shaft-mounted gears are employed to maintain the two rotors in the proper positions in dry compressors. These machines are commonly used in the following applications:

- Pharmaceutical industry
- High-purity chemical industries
- Aeration services in the brewing industry
- Gas systems that must be free of entrained air and other contaminants

Timing gears are not normally used with oil-injected rotary screw compressors. However, other liquid-injected compressors employ gears. This is done to keep the two rotors in proper mesh. The following are the three liquids injected in these machines:

1. Water
2. A heat-removing fluid
3. Other liquids

The two screws are separated by an oil-layer in oil-injected compressors. This oil-layer maintains adequate separation even as one rotor drives the second. The following are the advantages of all liquid-injected compressors:

- The liquid injected cools the gas. Thus, this prevents the following from occurring:
 - Polymerization of the gas
 - Operating the gas in the explosion-prone temperature range
- Liquid-injected machines provide considerably higher compression ratios than their dry counterparts. This is partially because the liquid-injected compressors do not require seals at the end of the rotor chambers. This feature allows these units to have a shorter bearing span. Thus, the rotor deflection is minimized. For example, a single-stage of a liquid-injected compressor can produce the same pressure ratio as two or more stages of a dry compressor.

20.1.4 Operating Principles

The gas entering a screw compressor is entrapped in the spaces between the convolutions of the two helical rotors. The volume of the gas diminishes gradually as it moves toward the discharge port. This increases the pressure of the gas.

The following are the definitions of the built-in volume ratio, v_i , and compression ratio, π_i , across the compressor:

$$v_i = \frac{\text{Volume of a given mass of gas at the discharge pressure}}{\text{Volume of the same mass of gas at the suction pressure}} = \frac{V_{\text{discharge}}}{V_{\text{suction}}} = \frac{V_2}{V_1}$$

$$\pi_i = \frac{\text{Pressure of the gas at the discharge of the compressor}}{\text{Pressure of the gas at the inlet of the compressor}} = \frac{P_{\text{discharge}}}{P_{\text{suction}}} = \frac{P_2}{P_1}$$

The value of π_i is calculated using the following equation:

$$\pi_i = v_i^k$$

where k is the ratio of the specific heat of the gas at constant pressure and, constant volume, respectively:

$$k = c_p / c_v$$

Figure 20.7 illustrates the theoretical pressure-volume diagram of the compression process. The design compression ratio is shown as π_i on Fig. 20.7. The gas is discharged

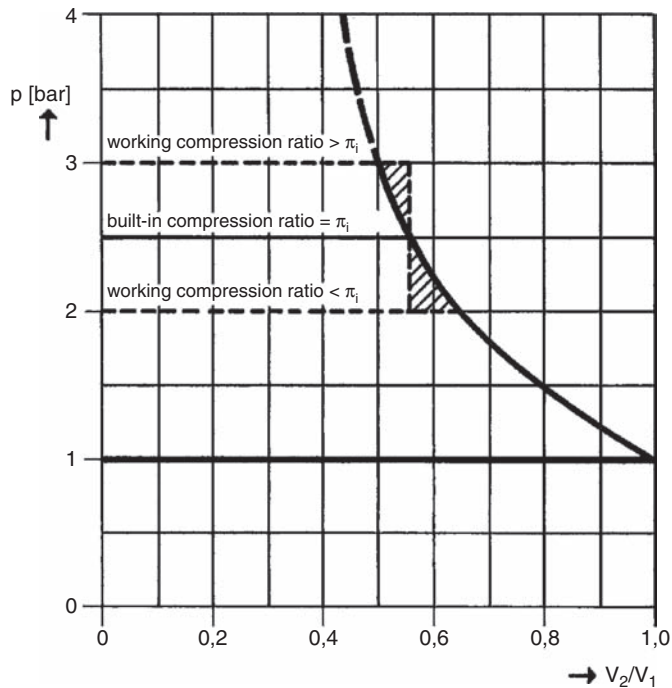


FIGURE 20.7 The P-V diagram of a modern helical screw (rotary screw) compressor. (Source: Aerzen USA Company, Coatesville, Pa.)

from the compressor into a receiver having a compression ratio higher than π_i in some applications. The compressor end wall will be exposed to the higher pressure in this case.

Rotary screw compressors can easily produce compression ratios (discharge pressure) higher than π_i . They can also accommodate lower compression ratios than π_i . However, the compressor polytropic efficiency will drop when its compression ratio deviates from π_i . The magnitude of this drop in compressor polytropic efficiency increases when the compression ratio deviates further from π_i . These losses in the compressor polytropic efficiency are shown as shaded areas in Fig. 20.7 for higher and lower compression ratios than π_i .

Dynamic compressors (centrifugal and axial compressors) will likely undergo *surge* when operated at higher compression ratios than their design compression ratio (π_i). These compressors experience the following variations during the surge process:

- Flow reversal
- Higher vibrations
- Higher discharge temperature
- Significant drop in compressor polytropic efficiency

The surge phenomenon can cause serious damage to these compressors. However, rotary screw compressors can easily accommodate this increase in compression ratio. They are only limited by the following two constraints:

1. Strength of the machine components
2. Input power

Thus, rotary screw compressors have a significant advantage over dynamic compressors in accommodating higher compression ratios than π_r .

20.1.5 Flow Calculation

The theoretical volumetric flow rate of a rotary screw compressor can be calculated using the following formula:

$$Q_0 = \frac{nq_0}{1000}$$

where Q_0 = Theoretical volumetric flow rate through the compressor in std m^3/min

n = Compressor operating rpm

q_0 = The unit volume conveyed by 1 revolution of the main helical rotor in L/r

The actual volumetric flow rate through the compressor, Q_a , is less than the theoretical one (Q_0). The difference between these two values is Q_v . This term represents the amount of gas flowing back through the very small clearances between the various components of the compressor. Thus,

$$Q_a = Q_0 - Q_v$$

The Q_v is also known as the volume flow lost via component slippages. It depends on the following terms:

- Total cross-sectional area of the clearances
- Density of the gas being compressed
- Compression ratio
- Peripheral speed of the rotor
- Built-in volume ratio

20.1.6 Power Calculation

The volumetric efficiency of the compressor, η_v , is given by the following formula:

$$\eta_v = \frac{Q_a}{Q_0} = 1 - \frac{Q_v}{Q_0}$$

The following equation provides the theoretical power required, W_0 (in kW) to compress the induced volumetric flow rate, Q_a :

$$W_0 = \frac{10^{-3}}{60} \rho_a Q_a H_a$$

where ρ_a = the gas density at inlet conditions. It is expressed in kg/std m³

H_a = the amount of energy required to compress 1 kg of gas adiabatically from pressure P_1 to P_2 .

The theoretical power required can also be determined from the following equation:

$$W_0 = \left(\frac{10^4 Q_a P_1}{6000} \right) \left(\frac{k-1}{k} \right) \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right]$$

where Q_a = expressed in std m³/min

P_1 = in bar

However, the actual power required by the compressor, W_a , is given by the following equation:

$$W_a = W_0 + W_d + W_v$$

where W_d = the dynamic flow loss

W_v = the mechanical losses

W_v is normally around 8 to 12% of the actual power. It includes viscous or frictional losses in the following three components:

1. Bearings
2. Timing gears
3. Step-up gears

The dynamic flow losses, W_d are typically around 10 to 15% of the actual power. These losses depend heavily on a crucial factor know as N_{id} . This factor depends on the following terms:

- Compression ratio
- Mach number (ratio of gas velocity and the velocity of sound in the gas) at compressor inlet conditions

The following formula can be used to determine the dynamic flow power loss:

$$W_d = C_f \frac{L}{D} \frac{k}{1.4} \frac{P_1}{1.013} \left(\frac{Q_0}{60}, N_{id} \right)$$

where C_f = empirical factor determined from Fig. 20.8

L = rotor length

D = rotor diameter

N_{id} = empirical factor determined from Fig. 20.9

The following are the reference conditions assumed in the previous formula for W_d and the associated charts:

$$Q_0 = 60 \text{ std m}^3/\text{min}$$

$$L/D = 1.0$$

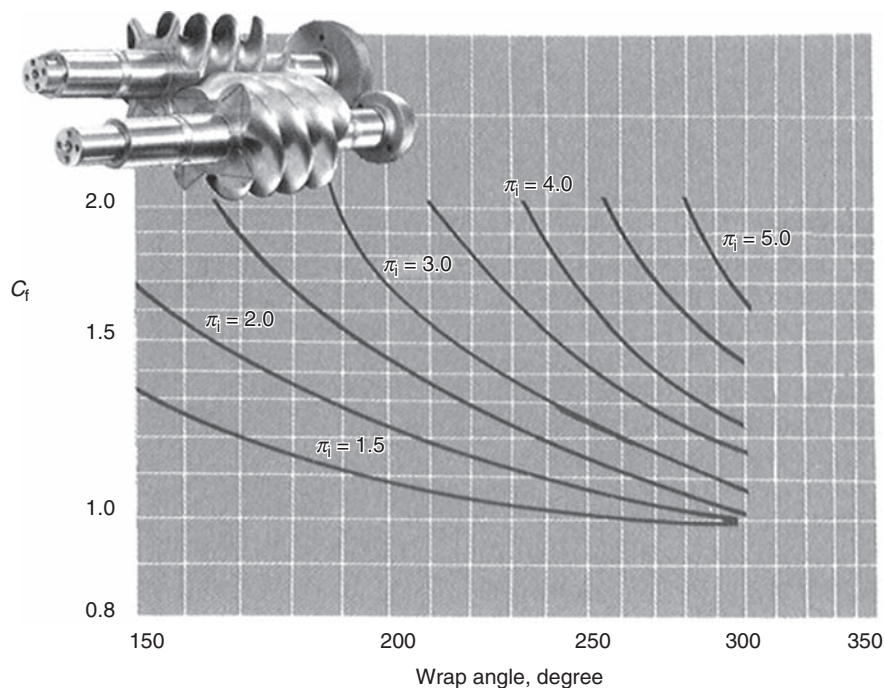


FIGURE 20.8 Empirical loss factor C_f as a function of the compression ratio π_1 and the wrap angle of screw compressor rotors. (Source: Aerzen USA Company, Coatesville, Pa.)

The wrap angle is defined as the rotation of a point on the thread of a screw as it travels from the bottom to the top of the rotor. It is equal to 300° in this case.

$$P_1 = 1.013 \text{ bar}$$
$$k = 1.4 \text{ (gas constant for air)}$$

Thus, the following formula is used to determine the actual power requirement for a rotary screw compressor:

$$W_a = W_0 + W_d + W_v$$

Screw compressors used for important process applications in North America are built in compliance with the American Petroleum Institute (API) Standard 619. This standard includes the following two requirements:

1. The compressor capacity is not allowed to have negative tolerance.
2. The compressor power requirement may not exceed the stated horsepower by more than 4%.

Some screw compressors manufactured in Europe are in compliance with API Standard 619. However, the others are normally built in compliance with Verein Deutscher Ingenieure (VDI; Society of German Engineers) Specification 2045. This specification has different tolerances than API Standard 619.

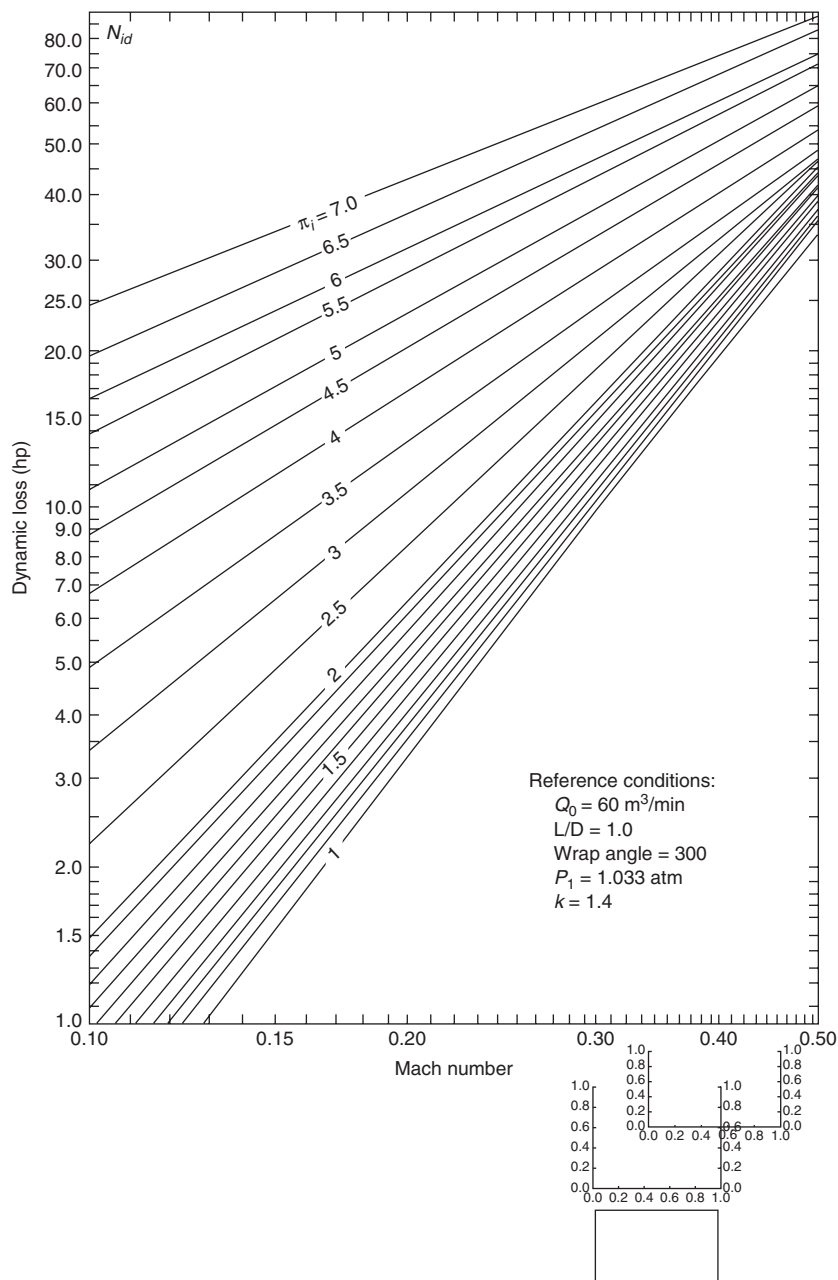


FIGURE 20.9 Empirical loss factor N_{id} versus Mach number at different compression ratios π_i . The effect of compressor inlet conditions on the dynamic flow power losses is shown as a function of the Mach number of the gas. (Source: Aerzen USA Company, Coatesville, Pa.)

20.1.7 Temperature Rise

The following equations are used to determine the discharge temperature (in °C) of a dry screw compressor:

$$\Delta T_0 = \frac{T_1}{\eta_v} \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right]$$

$$T_2 = T_1 + \Delta T_0$$

The maximum final discharge temperature allowed for a screw compressor when operating under oil-free, dry running conditions is 250°C. This temperature corresponds to a compression ratio (P_2/P_1) of 4.5 when air is being compressed. The adiabatic exponent k is equal to 1.4 in this case. However, gases having a k value of 1.2 are permitted to have a maximum compression ratio of 7.0 to meet the same temperature limit.

Figures 20.10 and 20.11 illustrate an oil-injected screw compressor. The oil flowing through this compressor removes most of the heat generated by the compression process. The oil flow injected into the compressor is adjusted to prevent having a final discharge temperature exceeding 90°C (194°F).

The value of the compression ratio in these applications is allowed to reach 21 if the air entering the compressor is under atmospheric conditions.

20.1.8 Capacity Control

The flow of a screw compressor is best controlled using a variable-speed method. This is because these compressors are positive displacement machines. The discharge pressure of these machines does not drop significantly when their rotational speed decreases.

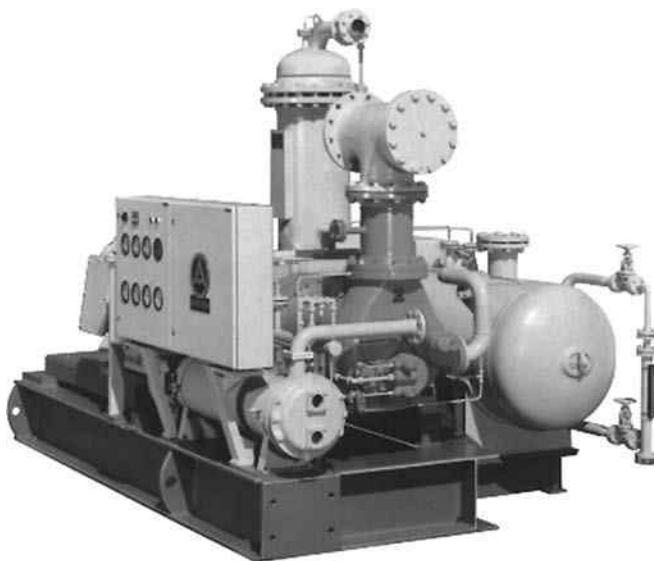


FIGURE 20.10 Oil-injected rotary screw compressor. (Source: Aerzen USA Company, Coatesville, Pa.)

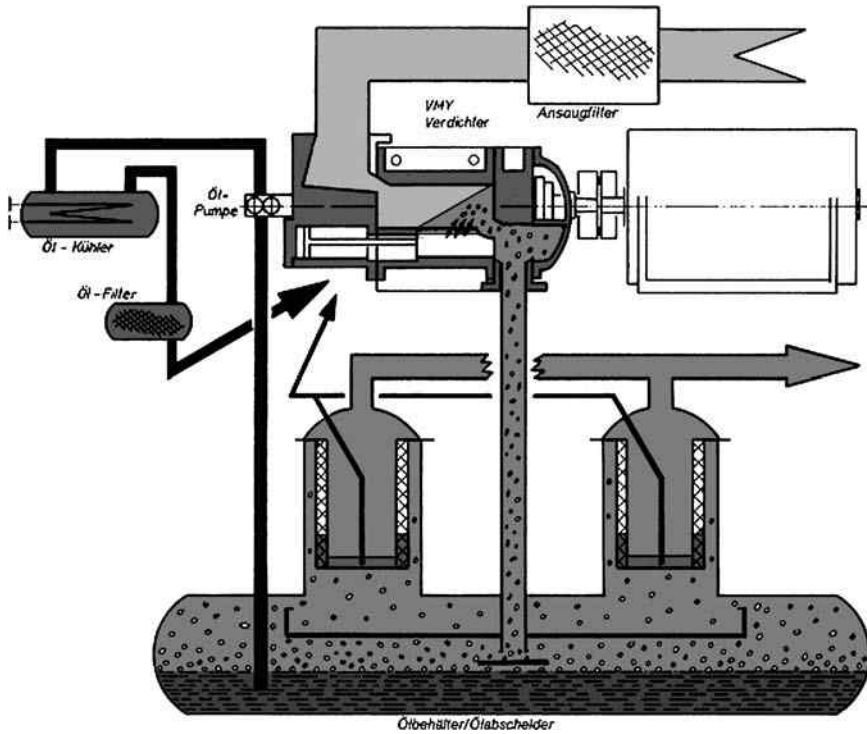


FIGURE 20.11 Operating principle of oil- or liquid-injected rotary screw compressors.
(Source: Aerzen USA Company, Coatesville, Pa.)

However, the discharge pressure of a dynamic compressor will drop significantly if its rotational speed is decreased. Thus, variable speed methods cannot be used to control the flow of a dynamic compressor. These methods include the following:

- Electric motor using a variable-speed drive
- Torque converter
- Steam turbine

The rotational speed of a screw compressor can be reduced to about 50% of the maximum permissible value. The capacity (volumetric flow rate) and power requirement of the compressor will drop by the same proportion.

The capacity of a screw compressor can also be controlled using a bypass. This method involves returning a portion of the discharge flow (surplus gas) back to the intake of the compressor. A control system is used to throttle the portion of the flow returning back to the compressor intake. This system throttles the flow returning back to the compressor intake by maintaining the desired compressor discharge pressure. An intermediate cooler is used to reduce the temperature of the surplus gas returning to the compressor intake. This temperature is reduced to the same level as the inlet temperature to the compressor. The compressor capacity is also controlled using a full-load and idling-speed governor. This method involves the use of a transducer to actuate a

diaphragm valve (Fig. 20.12). This transducer opens the diaphragm valve when a predetermined final pressure is reached. This opens a bypass between the discharge side and the suction side of the compressor. The compressor operates in an idling mode in this situation. The compressor discharge flow returns to the intake of the compressor. The compressor continues to idle until the system pressure drops to a predetermined

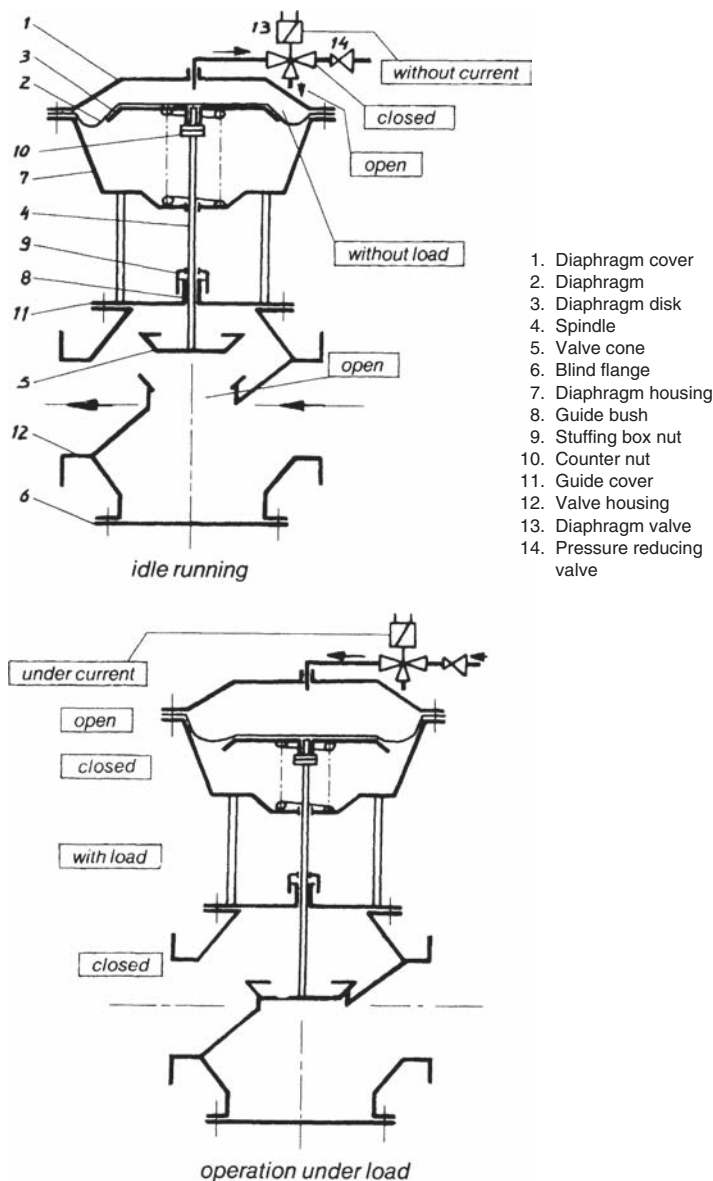


FIGURE 20.12 Diaphragm valve associated with constant-speed unloading devices in rotary screw compressors. (Source: Aerzen USA Company, Coatesville, Pa.)

minimum value. The transducer will close the diaphragm valve at this stage. The compressor will again become fully loaded.

The compressor capacity can also be controlled by the following two methods:

1. Throttling the suction of the compressor
2. Unloading the compressor discharge

These methods are suitable for air compression in applications involving the manufacturing of industrial gases. However, the suction throttling method has a disadvantage. It generates a drop in pressure at the inlet to the compressor. This increases the compression ratio of the compressor. However, some machines may not be designed to handle this increase in compression ratio. Thus, it is necessary to ensure that the increase in compressor discharge pressure does not exceed its mechanical design limitations.

Oil-flooded screw compressors operate over a wider range of compression ratio than dry compressors. This is because the temperature of the final compression stage is controlled by the amount of oil injected. These units can easily achieve a smooth adjustment of the volumetric flow rate.

Figure 20.13 illustrates an internal volume-regulating device. This equipment can readily be used in larger compressors. It consists of a slide. This slide is shaped to match the contours of the housing. The slide can move in a direction parallel to the rotors. This

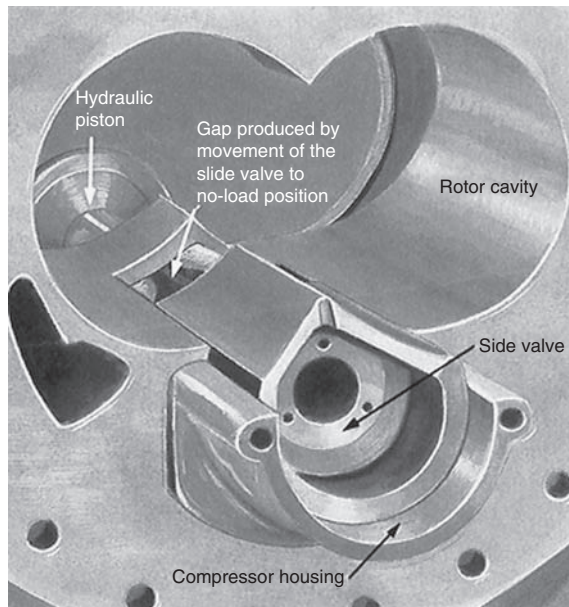


FIGURE 20.13 Internal volume regulating device for oil-injected rotary screw compressors. The position of a slide valve can be shifted in a direction parallel to the axes of the rotors. This provides control of the volume flow of the compressed gas. (Source: Aerzen USA Company, Coatesville, Pa.)

permits the reduction of the effective length of the rotors. This method provides a smooth, indefinitely variable control of the compressor capacity. The compressor capacity can be varied from 100 to 10% of the full compressor capacity using this method. The compressor power requirement will drop when the compressor capacity is reduced. These methods of capacity control are used mainly on liquid-injected screw compressors where the liquid has lubricating properties.

20.1.9 Mechanical Construction

Rotary screw compressors used in high speed and pressure applications have the following types of bearings:

- Sleeve bearings
- Self-adjusting multi-segment thrust bearings

These bearings are illustrated in Fig. 20.14.

Figure 20.15 illustrates various types of compressor seals. They include carbon ring seals. These seals are used in conjunction with the following:

- Buffer gas injection
- Leak-off ports that connect the seal back to the compressor suction

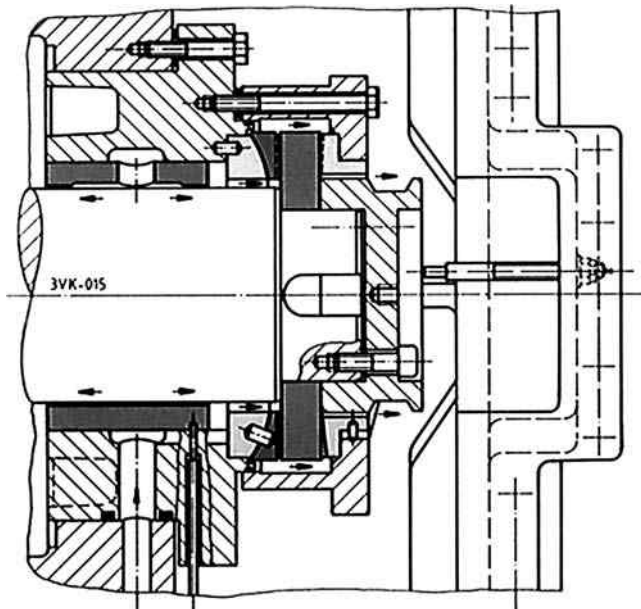


FIGURE 20.14 Radial and thrust bearings furnished with large rotary screw compressors.
(Source: Aerzen USA Company, Coatesville, Pa.)

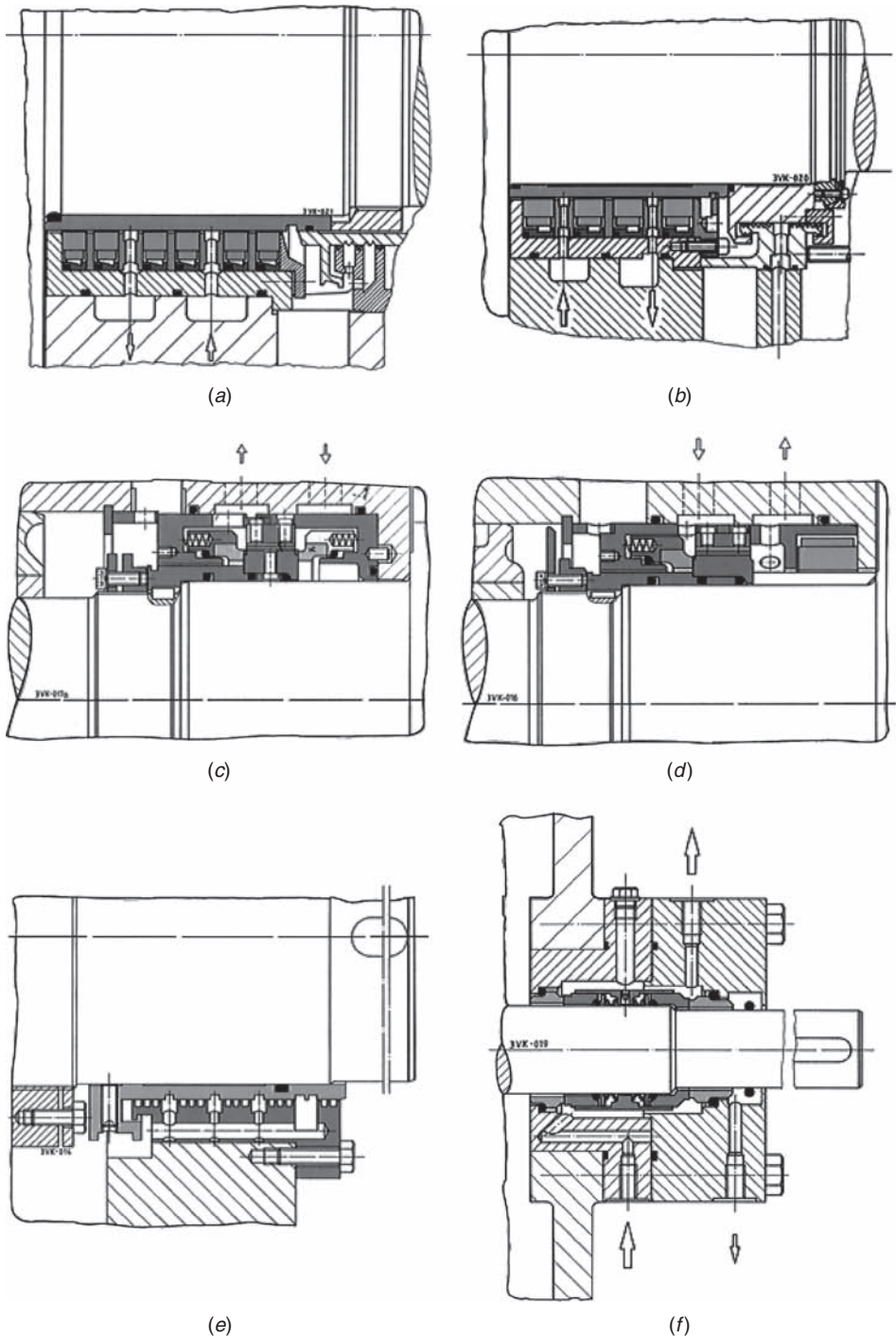


FIGURE 20.15 Sealing arrangements for rotary screw compressors. At the conveying chamber: (a) carbon labyrinth seal, (b) water-sealed floating rings, (c) double-acting slide ring seal, (d) combined floating ring and slide ring seal. At the drive shaft: (e) labyrinth seal, (f) double-acting slide ring seal. (Source: Aerzen USA Company, Coatesville, Pa.)

Figure 20.15*b* illustrates floating ring seals. They contain barrier water. This water provides the following functions:

- Sealing
- Cooling
- Flushing
- Gas scrubbing medium

Most of the barrier water returns to its supply. It is reused in the seal. Figure 20.15*c* illustrates stationary double-mechanical seals. These seals are lubricated with either pressurized water or oil. They are normally used in applications where the seal emissions must be minimized. Figure 20.15*d* illustrates a stationary single seal combined with a floating ring. These seals are typically used in applications that have a high-differential pressure across the seal. Labyrinth and mechanical seals are illustrated in Fig. 20.15*e* and 20.15*f*. These traditional seals are normally used on the transmission side of the casings of geared rotary screw compressors.

20.1.10 Industry Experience

Modern water-injected two-stage screw compressors are used commonly in coke gas* producing plants. These compressors have proven to have superior capabilities to conventional centrifugal compressors in these applications. This is because centrifugal compressors experience significant performance degradation when handling coke gas.

This degradation is caused by the rapid polymerization[†] of this dirty and hydrogen-rich gas. In fact, centrifugal compressors require cleaning every 6 to 8 weeks in this applications. The associated cost with this downtime prohibits the use of these compressors in this application.

Figure 20.16 illustrates two-stage, water-injected screw compressors. They are installed in a coal gasification plant. They replaced centrifugal compressors that were used in this application. These screw compressors are driven by a 5.5 MW (7400 hp) electric motor. Their availability and reliability are superior to their predecessors. Each one of them compresses around 33,000 std m³/h (19,420 scfm) of coke-oven gas. The discharge pressure of these compressors varies from 1 to 12 bar (14.5–174 psia). Their operational cost is significantly lower than their predecessors. This is due to a reduction in power and overall maintenance requirements.

The discharge temperature from a screw compressor that does not use water injection (dry compression) in coke gas plants can easily exceed 100°C (212°F). This will evaporate the higher hydrocarbons. The asphalt-like residue will remain in the gas. It forms a coating on the rotors and the housing. These deposits will reduce the efficiency and flow through the compressor. This sticky residue may also attach the two rotors together when the compressor is shutdown.

Water injection into the screw compressor limits its discharge temperature to 100°C. This is because the evaporation of this water requires a considerable amount of the heat generated by the compression process. This reduction in the compressor discharge temperature prevents the polymerization of the gas.

*See Appendix for details.

[†]Polymerization: chemical covalent bond formation between smaller molecules to form a larger molecule. This larger molecule will be in the form of aerosol (liquid in the gas).

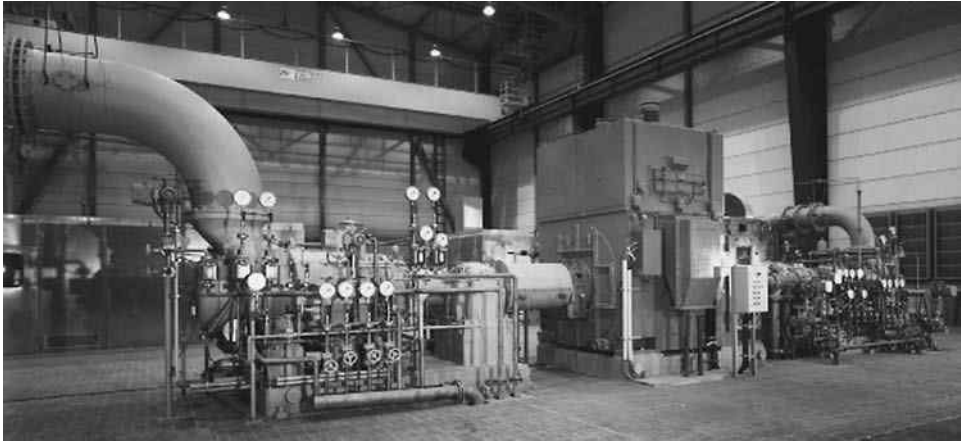


FIGURE 20.16 Large (approximately 7000 hp) rotary screw compressor installation at a German coal gasification plant. (Source: Aerzener Maschinenfabrik, Aerzen, Germany.)

The water is supplied by the following methods:

- Through the four-shaft seals of each compression stage
- Spraying the water into the inlet nozzle of each compression stage

A control system is used to adjust the flow of water into the compressor. It relies on a temperature transducer installed at the compressor discharge. Figure 20.17 illustrates a flow diagram of a two-stage unit including its water injection system.

The control system maintains the gas temperature below the dew point. This allows the water to flush the deposits. This water is then drained off at the following locations:

- Discharge silencer
- Water separators of the intercoolers and aftercoolers of each compression stage

The compressor is made from chromium-nickel alloy steel in this application. This material has a high resistance to corrosion. It is selected because the gas contains the following corrosive components:

- Hydrogen sulfide
- Ammonia
- Hydrogen cyanide

This material is also known for its erosion resistance. However, corrosion-related wear can occur in the compressor over a period of time. It is mainly caused by the water injection. Thus, a yearly inspection should be performed to determine the extent of corrosion-related wear in the compressor.

This application employs a control system known as *intermediate pressure regulation*. This system matches the volumetric flow through the compressor quickly with the

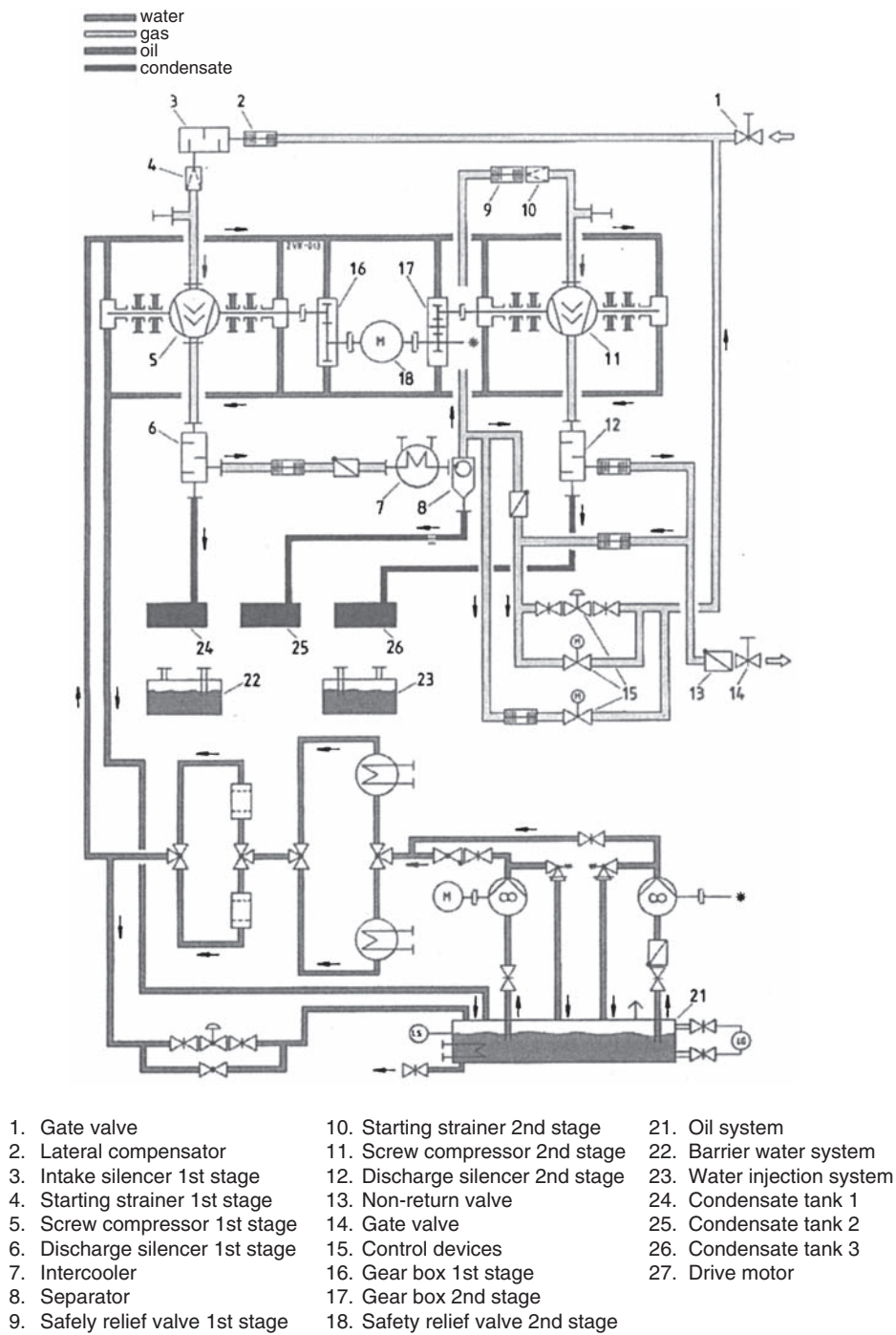


FIGURE 20.17 Flow diagram of a two-stage rotary screw compressor unit with a barrier water seal. (Source: Aerzener Maschinenfabrik, Aerzen, Germany.)

process gas requirement. The gas that is not required by the process is returned after the first stage through a bypass to the compressor-intake side. This feature has the following advantages:

- Reduces the power consumption significantly.
- Provides continuous regulation of volumetric flow rate from 20,000 to 33,000 std m³/h (11,770 to 19,420 scfm).

The following example shows the amount of reduction in power consumption:

- The power consumption is about 3.2 MW (4.29 hp) when the flow is 20,000 std m³/h.
- The power consumption is about 3.95 MW (5.295 hp) when the flow is 30,000 std m³/h.

Separate superchargers are installed at the compressor inlet. They provide the capability of boosting the inlet pressure to about 1.6 bar (23 psia). This provides an intake volumetric flow rate of 46,000 std m³/h (20,070 scfm). Thus, the combination of intermediate pressure regulation and supercharging provides continuous regulation of the volumetric flow rate from 20,000 to 46,000 std m³/h. This capability allows the coking plant to adapt to any change in the process requirement.

20.1.11 Maintenance History

The three two-stage 5.5 MW rotary screw compressors were partially overhauled every 5 years. The carbon seal rings were found damaged during the overhaul. They were replaced. All the remaining parts of the compressors including the bearings were installed without any repair or modification. The maintenance costs for the three compressors were \$140,000 in 2006. This is a typical cost for a 5-year overhaul. It includes the cost of labor and materials. This cost is a fraction of the cost that was incurred by the centrifugal compressors that were used in the same application.

An inspection was also performed on a second application. This application operates two three-stage rotary screw compressors. This was the first inspection after 35,000 hours of uninterrupted service. The carbon seal rings showed signs of wear. However, all the remaining parts were found to be in an excellent condition. The inspection interval of this application was increased from 45,000 to 50,000 operating hours. This was mainly because the carbon seal rings could have still been used for an extended period of time beyond the initial 35,000 hours of operation.

The screw compressors were also inspected in a third application. The carbon seal rings were found to have experienced accelerated erosive wear. This was mainly caused by unacceptable water quality. These compressors require demineralized water. Poor quality water will shorten the life of the carbon seal rings. It will also lead to unnecessary forced outages.

Screw compressors are also used for the delivery and compression of contaminated gases as well. They are also ideally suited for compression of gases that tend to polymerize at relatively low temperatures.

20.1.12 Performance Summary

The performance of a screw compressor is affected by the following factors:

- Gas properties
- Internal clearances
- Length/diameter ratio of rotors
- Compression ratio
- Operating speed

The typical isentropic efficiency of a screw compressor is between 70 and 80%. Its maximum compression ratio is the value that corresponds to the final discharge temperature of 250°C (482°F). This value depends on k (ratio of c_p/c_v) of the gas being compressed.

The rotational speed of a screw compressor varies from 2000 to 20,000 rpm. The peripheral speed of the rotor is normally between 40 to 120 m/s (131 to 394 ft/s). The maximum peripheral speed of the rotor is 150 m/s (492 ft/s). This value is used for gases having low molecular weight. The value of the peripheral speed determines the magnitude of the rotational speed.

Rotary screw compressors have the following advantages:

- “Wet screw” compressors are available with an oil loop. This loop serves either of the following:
 - Both bearing lubrication and compressor
 - Bearings and compressor separately (dual loop)
- Lower sensitivity to changes in the molecular weight of the gas than centrifugal compressors.
- Higher tolerance for handling polymerizing gases than all other compressors except liquid ring compressors.
- Higher capability to accept more liquids and fine solids entrained with the gas than all other compressors except liquid ring compressors.
- Higher isentropic efficiency and lower maintenance requirements than liquid ring compressors.
- Availability exceeding 99.5%. The availability of these compressors may exceed that of centrifugal and axial compressors in some applications.
- Smaller size and lower cost than the equivalent reciprocating compressors used in the same application.
- Lower cost than the equivalent centrifugal compressors in applications requiring small or moderate-sized units [compressor power requirement below 3 MW (4021 hp)].
- Capability to produce higher discharge pressure than other types of rotary positive displacement machines.

The following are the disadvantages of rotary screw compressors:

- They generate high noise level during operation. However, most modern screw compressors have noise-reducing equipment (silencers). This equipment allows these compressors to meet the most stringent requirements (Fig. 20.18).

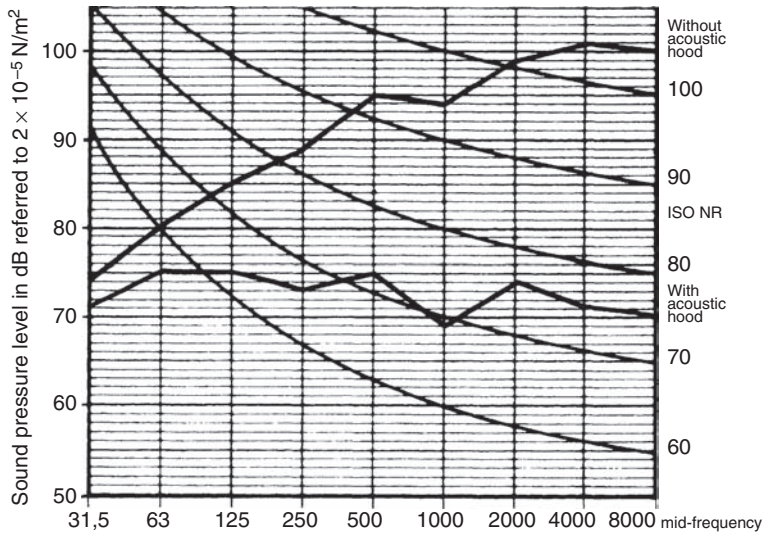


FIGURE 20.18 Typical sound levels obtained from rotary screw compressors with and without acoustic enclosures. (Source: Aerzen USA Company, Coatesville, Pa.)

- They require pulsation suppression equipment. However, these pulsations are not as severe as the piping pulsations encountered with the equivalent reciprocating compressors. Most modern screw compressors incorporate pulsation bottles to reduce the effects of these pulsations. Pipe expansion loops are also used in applications having high discharge temperatures.

20.2 Oil-Flooded Single-Screw Compressors

Oil-flooded single-screw compressors are available for applications having a flow near 1700 m³/h (1000 cfm) and discharge pressures around 5.5 MPa (800 psi). Figure 21.19 illustrates an oil-injected single-screw compressor. The cooling oil that circulates through the compressor performs the following functions:

- Absorb the heat of compression.
- Seals the gas in the compression spaces.
- Lubricates the rolling element bearings.

Synthetic lubricants are used in the following applications:

- Corrosive gases are flowing through the compressor.
- High condensation rates are found in the compressor.

Figure 20.20 illustrates the gas flow through the compressor. The intermeshing of the three rotating components compresses the gas. The gas enters the compressor through the inlet passage. It fills a groove in the single-screw rotor. A gate rotor tooth meshes with this groove. This traps the gas in the groove and seals the groove. The

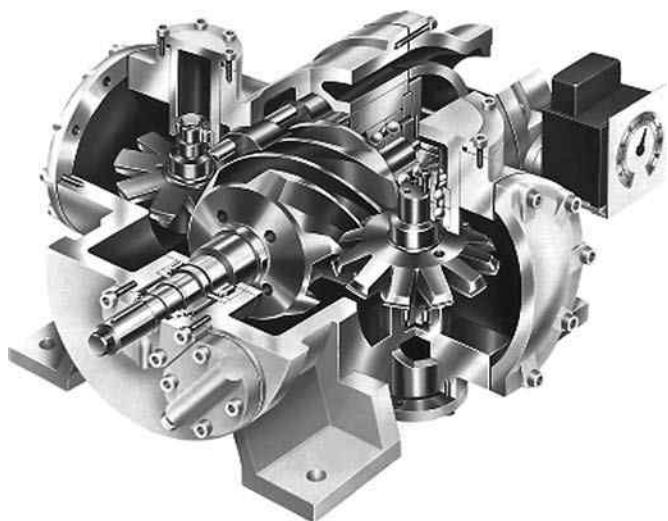


FIGURE 20.19 Oil-injected single-screw compressor. (Source: Dresser-Rand Company, Broken Arrow, Okla.)

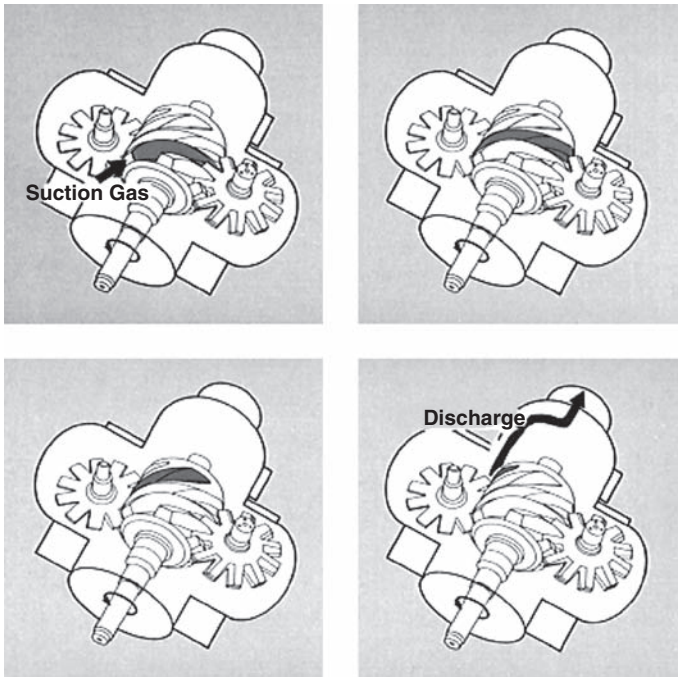


FIGURE 20.20 Gas flow in an oil-injected single-screw compressor. (Source: Dresser-Rand Company, Broken Arrow, Okla.)

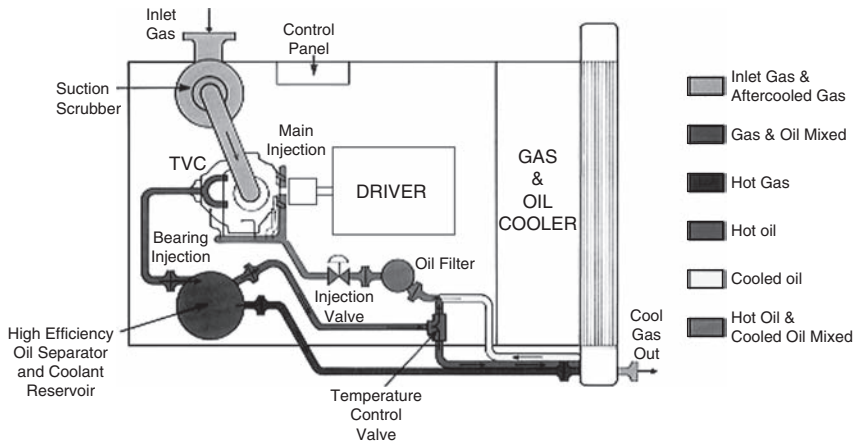


FIGURE 20.21 Process flow schematic showing an oil-injected single-screw compressor. (Source: Dresser-Rand Company, Broken Arrow, Okla.)

trapped gas is compressed with further rotation of the screw rotor. This is due to the reduction of its volume in the groove. The injected oil seals the clearances in the compressor. This prevents the leakage of the gas. The gas is then delivered from the groove to the discharge manifold through the discharge port.

The gas is compressed on both sides of the screw rotor simultaneously. This is due to the intermeshing of the two gate rotors with the screw rotor. Thus, the compressive forces are balanced radially on the screw rotor.

Figure 20.21 illustrates the process flow schematic of an oil-injected single-screw compressor. The mixture of oil and gas leaving the compressor enters a high-efficiency gas-oil separator. The oil passing through the separator is cooled and filtered. It is then re-injected into the compressor. The compressor discharge temperature is maintained at a constant value during operation. However, this discharge temperature varies depending on the application.

Slide pistons mounted in the compressor casing are used to control the capacity of the compressor. Rack-and-pinion gears are used to move the two slides axially. These slides are controlled by a stub shaft. This stub shaft protrudes from the side of the casing near the discharge flange. Figure 20.22 illustrates one capacity slide at 100% capacity and 0% capacity. The screw rotor has been taken out of Fig. 20.22 to clarify the details. The slide is placed in the 100% position so that the gas cannot leak during compression. Thus, all the gas that enters the groove is delivered to the triangular discharge port. When the slide valve is placed at a position less than 100%, some of the gas that had entered the groove returns to the suction prior to compression.

Figure 20.23 illustrates a single-shaft, two-stage, oil-flooded single-screw compressor. This compressor is designed for high-discharge pressure application. The two compression stages are connected through a gastight transition piece. The following are the main features of this compressor:

- Side-stream capability
- Infinitely variable capacity control between 40% and 100%.

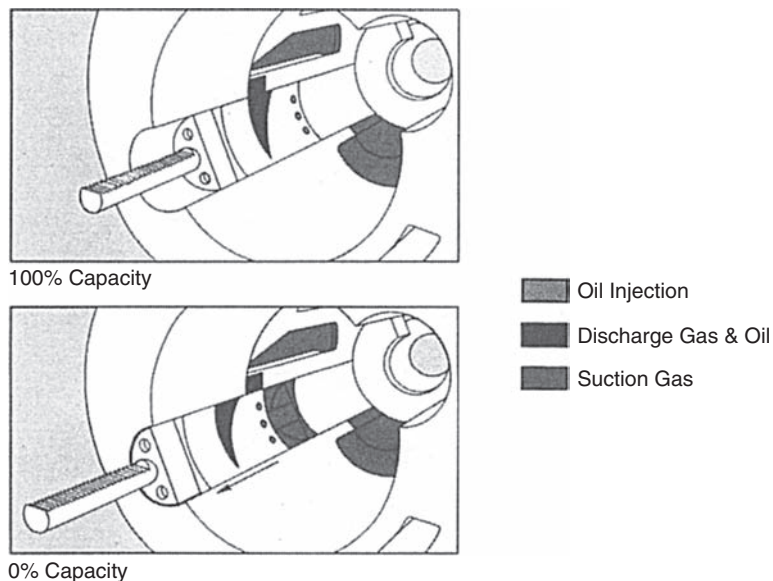


FIGURE 20.22 Capacity control slides in an oil-injected single-screw compressor. (Source: Dresser-Rand Company, Broken Arrow, Okla.)

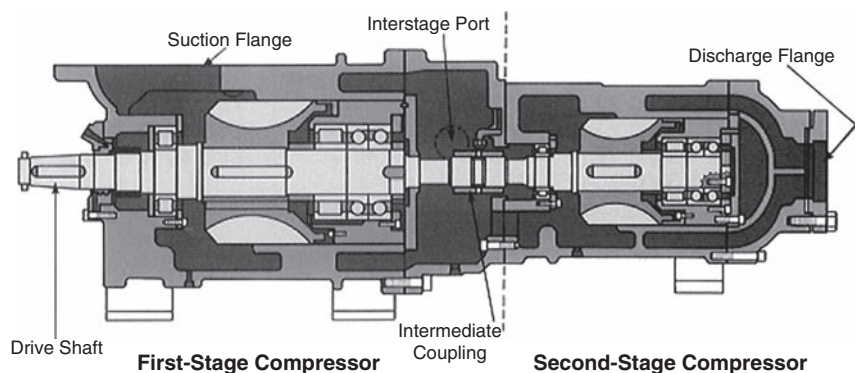


FIGURE 20.23 Two-stage version of an oil-flooded single-screw compressor. (Source: Dresser-Rand Company, Broken Arrow, Okla.)

Oil-flooded compressors having a single-oil circuit for the compressor and the bearings can have the following problems in *dirty gas** applications:

- Extended forced outages
- Frequent failures

*The term *dirty gas* indicates that the gas has trace quantities of H_2S .

These problems prohibit the use of single-oil circuit compressors in these applications. Thus, separate oil circuits are used commonly for these applications. Solids are entrained with the gas in some applications. They must be removed from the oil or other medium that is used in liquid-filled compressors. Filter separators are required for these applications.

20.3 Selection of Modern Reverse-Flow Filter Separators

The self-cleaning coalescers (SCCs) provide the lowest life-cycle-cost for many compressor applications. This equipment is discussed in detail in the following sections.

20.3.1 Conventional Filter Separators and Self-Cleaning Coalescers

Figure 20.24 illustrates a *conventional filter separator* (CFS). The gas velocity is reduced as it passes through the filter elements. The various and sundry contaminants (e.g., iron sulfides) are initially caught by the filter. However, the gas forces converts the contaminants to a particle size. This will allow it to pass through the filter elements.

The solid particles and liquids entrained in the gas coalesce on the inside of the filter element. These solid particles and liquids are then re-entrained in the gas flowing in the collector tube before entering the next separator section. This section has wire mesh or vanes. They allow fine mist droplets and particles (known as *globules* of liquid) to pass through. The size of these globules is about $3.8\text{ }\mu\text{m}$. Thus, a high percentage of liquid and small solids (particulates) pass through the filter. They remain entrained in the gas stream leaving the CFS.

Figure 20.25 illustrates a self-cleaning coalescer (SCC). This design reduces significantly the entrainment of liquid and particulates in the gas stream leaving the SCC. The gas entering the SCC passes first through the plenum. The gas then flows through collection tubes to the filter elements. A slug-free liquid knockout is installed at the front of an SCC. The gas velocity drops in the de-entrainment section. This will allow any particulates that have passed through the filter to drop out or to be entrained with the coalesced liquid droplets falling out from the SCC at this stage. The effectiveness of this

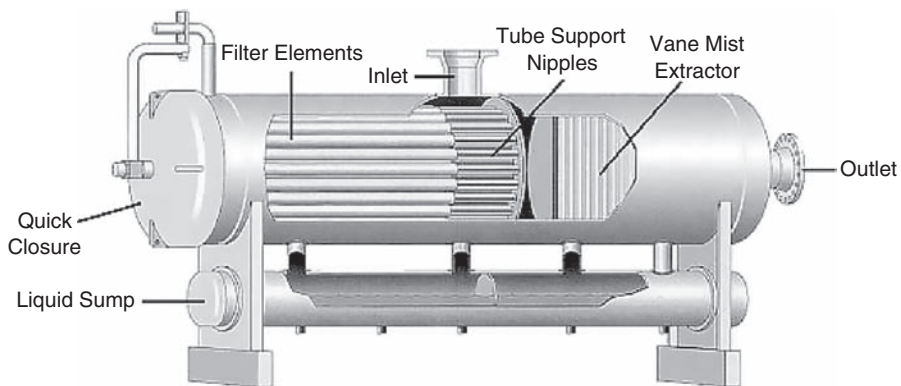


FIGURE 20.24 Conventional filter separator. (Source: King Tool Company, Longview, Tex.)

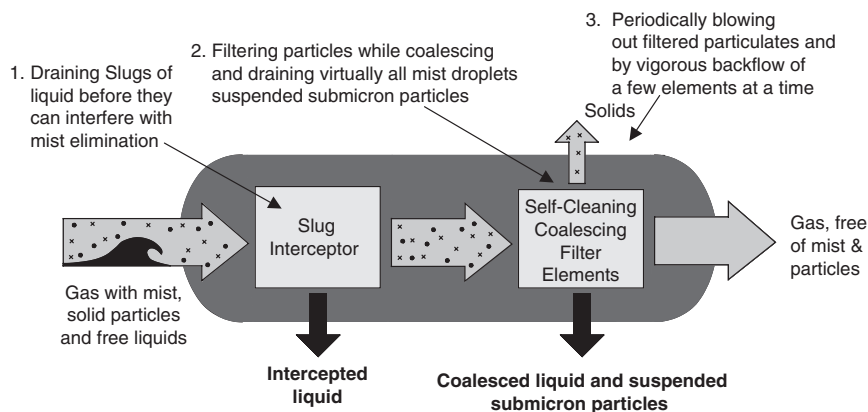


FIGURE 20.25 Self-cleaning coalescer. (Source: King Tool Company, Longview, Tex.)

design has been proven for more than three decades of field experience. The SCC is capable of removing all of the following items:

- Entrained particulates and mist globules
- Liquids and large agglomerated materials

20.3.2 Removal Efficiencies

Some manufacturers of conventional filter separators (CFSs) claim removal efficiencies for their CFSs much higher than those actually achieved. Some CFS designs have high pressure drop across them in the following applications:

- Gases containing moisture
- High-velocity gases
- Short-filter elements

Other CFS units do not have any slug-handling capability. Moreover, CFS units cannot achieve a filtration effectiveness down to $0.3\ \mu\text{m}$ (significantly less than one hundredth of the width of a human hair) unless high-efficiency particulate air (HEPA) filters are used. These filters are mandate for use in nuclear facilities and hospital operating rooms.

20.3.3 Filter Quality

The filter elements used in a conventional forward-flow filter separator operate on the coalescing principle. This separator is considered to be a coalescer. The filter elements coalesce the liquid droplets into globules. The size of these globules is around $10\ \mu\text{m}$. The Vane Mist Extractor removes these globules from the flow. In fact, the Vane Mist Extractor guarantees the removal of all 8 to $10\ \mu\text{m}$ particles. Multistage filter configurations are required to remove droplets in the $0.3\ \mu\text{m}$ size range. HEPA-like filter designs provide the highest quality filtration service.

Long fiberglass filter elements are used in good quality conventional filter separators (CFS). These fiberglass filter elements use certain microfiber enhancements. Thick filter elements provide a longer path for the gas. This results in better coalescing of the liquids.

20.3.4 Selection of the Most Suitable Gas Filtration Equipment

Self-cleaning coalescers (SCCs) can remove the following from the gas:

- Iron sulfides
- Viscous fluids
- Slugs

The pressure drop across an SCC is low. It is around 10 to 15 cm (4–6 in) of water. The filtration equipment depends on the application. The following are the options available:

Filtration Equipment	Application
1. Dry filter or dry filter self-cleaning	Gases containing entrained solids
2. Line separator	Gases containing entrained liquid mist
3. Vertical or horizontal separator	Gases containing entrained liquid globules (mist, aerosol), or gases containing entrained liquid particles (mist) and free liquid (slug)
4. Vertical or horizontal filter separator	Gases containing entrained liquid globules (mist, aerosol) and stable solids
5. Reverse-flow mist coalescer with slug chamber	Gases containing entrained liquid globules (mist, aerosol), slugs, and (stable or unstable) solids. High-efficiency removal to sub-micrometer level or better (can be provided in self-cleaning configuration during full operation).
6. Oil bath separator-scrubber	Gases containing entrained liquid globules (mist) and solids (stable or unstable), removed to 3 μm at 97% efficiency by weight
7. Tricon 3 stage separator	Gases containing entrained liquid globules (mist), slugs, and solids (stable or unstable), removal to 3 μm at 97% efficiency

20.3.5 Evaluation of the Proposed Filtration Configurations

The offers for separation equipment received from various bidders should be evaluated. This evaluation should include the following considerations:

- *Flow velocity.* The internal configuration of the separation equipment vessel should not accelerate the flow (e.g., the internal components of the separation equipment should not have the shape of a nozzle). This is because the increase in velocity will cause the liquid to split into smaller globules.
- *Pressure drop.* The pressure drop from flange to flange across any separation equipment should be limited. This pressure drop should not exceed 14 kPa (2 psi) when the vessel operating pressure exceeds 3.4 MPa (500 psi). The flange-to-flange pressure drop across the separation equipment should be limited to 7 kPa (1 psi) when the vessel operating pressure is less than 3.4 MPa (500 psi). This is because the operating cost of the separation equipment increases with the pressure drop across it. The pressure drop across a filter element arrangement should not exceed 3.5 kPa (0.5 psi). The pressure drop across the filter element increases fourfold when it becomes wetted and 50% plugged. For example, if the initial pressure drop across the filter element is 3.5 kPa (0.5 psi) and the elements become half-plugged, the pressure will increase to 14 kPa (2 psi). The pressure drop across the filter will increase to 56 kPa (8 psi) when the filter element becomes three-quarters plugged. The pressure drop across the filter at this stage is 16 times larger than the initial pressure drop. The filter element should be replaced due to the significant increase in pressure drop. Thus, the initial pressure drop across the filter element should be kept as far below 3.5 kPa (0.5 psi) as possible. This is required to avoid frequent replacement of the filter element. The disposal of the filter element is expensive. The reduction in the frequency of filter replacement reduces the operating cost of the separation equipment.
- *Filter element cost.* The increase in the cost of the filter element and spare parts should be limited within a period of time (e.g., 5 years) following the procurement of the separation equipment. This is because many vendors give away the separation equipment at a low cost initially. However, these vendors start to increase the cost of filter elements and spare parts at an alarming rate.
- *Vessel life.* The useful life of separation equipment is around 20 to 25 years under normal operation. However, corrosion problems, vibration or pulsation, overloads, lack of routine maintenance, etc. can reduce the operating life of the separation equipment.

20.3.6 Life-Cycle-Cost Calculations

The life-cycle-cost calculations are used to determine the most suitable equipment for the application. The following equation should be used to determine the life-cycle-cost of the equipment:

$$LCC = C_{ic} + C_{in} + C_e + C_o + C_m + C_{dt} + C_{de} + C_{env} + C_d$$

where LCC = life-cycle-cost

C_{ic} = initial cost, purchase price (system, pipe, auxiliary services)

C_m = installation and commissioning cost

C_e = energy costs ("incremental Δp "-related)

C_o = operation costs, if applicable

C_m = maintenance and repair costs

C_{dt} = downtime costs

C_{de} = incremental repair cost, downstream equipment

C_{env} = enviromental costs

C_d = decommissioning and/or disposal costs

20.4 Conclusions

The total ownership cost of the filtration equipment is normally about seven times higher than the initial purchase price. This is because the total ownership cost includes the following:

- Engineering
- Bid process
- Purchase order
- Administration
- Testing
- Inspection
- Spare parts inventory
- Training and auxiliary equipment

Suppose that a team of engineers has selected separation equipment that matches exactly all the requirements of the application. The performance of this equipment would deteriorate quickly if cheap incompatible spare parts were used in it. The cost of the downtime associated with this poor performance can be significant. It may exceed the savings that were made on the cost of spare parts by hundreds of times. Thus, it is always recommended to purchase the spare parts suggested by the manufacturer of the separation equipment. The risk-averse operator will always focus on minimizing the life-cycle-cost of the separation equipment. This cost includes the downtime costs, maintenance and repair costs, and incremental repair cost of the downstream equipment.

20.5 Bibliography

Bloch, H. P., *A Practical Guide to Compressor Technology*, 2nd ed., John Wiley & Sons Inc., Hoboken, New Jersey, 2006.

20.6 Appendix: Coke Fuel

20.6.1 Introduction

Coke is a solid carbonaceous material derived from destructive distillation of low-ash, low sulfur bituminous coal. The bituminous coal used to make coke fuel must meet a set of criteria. These criteria include the following:

1. Moisture content
2. Ash content
3. Sulfur content
4. Volatile content
5. Tar
6. Plasticity

Coke is produced by baking the bituminous coal in an airless oven at temperatures as high as 2000°C (3632°F). The volatile constituents include the following:

1. Water
2. Coal-gas
3. Coal-tar

The baking process fuses together the fixed carbon and residual ash. Coke is used as the main fuel in iron-making blast furnaces.

20.6.2 Properties and Usage

Coke can be burned with little or no smoke. This is because the smoke-producing constituents have been driven-off during the coking of the coal. This feature makes coke a desirable fuel for stoves and furnaces. Bituminous coal cannot be used in these applications. This is because it generates a large amount of smoke during the burning process.

20.6.3 Other Coking Processes

Coke is also formed from the solid residue remaining from the refinement of petroleum. Petroleum coke is used in many applications other than being a fuel. These applications include the manufacture of dry cells, electrodes, etc. Plants that manufacture syngas also produce coke as an end product.

20.6.4 Bibliography

Wikipedia encyclopedia.

Straight Lobe Compressors

21.1 Applications

Straight lobe compressors (also known as blowers) are used in the following applications:

- Pneumatic conveying of materials
- Aeration of liquids
- Extraction of gases and vapors
- Supercharging of engines
- Drying materials
- Delivery of low-pressure gas, etc.

21.1.1 Operating Characteristics

The following are the operating characteristics of straight lobe compressors:

- *Capacity range:* 8.5 m³/h (5 ft³/min) to 102,000 m³/h (60,000 ft³/min)
- *Pressure range:* up to 102 kPa (15 psi)
- *Vacuum range:*
 - 38.1 cm (15 in) hg vacuum for conventional compressor
 - 68.6 cm (27 in) hg vacuum for externally aspirated compressor or liquid sealed

Compressor staging is used to achieve higher pressure and vacuum levels.

21.2 Operating Principle

Most modern straight lobe compressors usually have rotors with two or three lobes. The following is the operating principle of a two-lobe (Fig. 21.1) compressor.

Two lobes (figure eight) are mounted on each rotor in a two-lobe compressor. The two rotors operate in parallel within an elongated cylinder. These rotors are synchronized through a set of timing gear. The lower rotor is the drive rotor. Gas enters the rotor on the right-hand side when the drive rotor rotates clockwise (position 1). The gas is discharged on the left-hand side. The upper rotor is the driven rotor. This rotor rotates counterclockwise. The drive rotor rotates the driven rotor by the timing gears (Fig. 21.2, item 6). The drive rotor delivers volume A in position 1

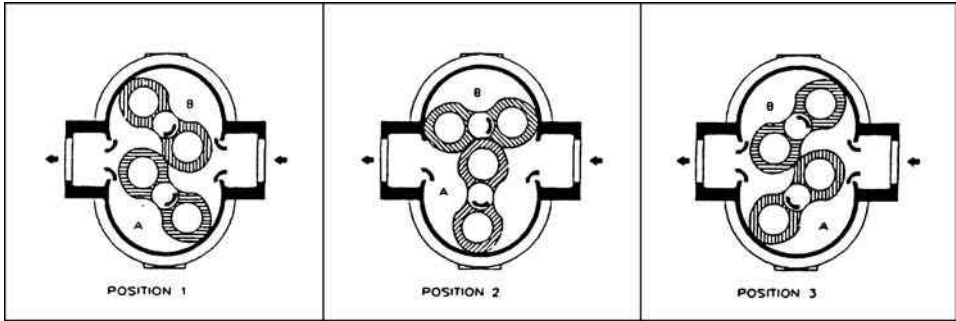


FIGURE 21.1 Two-lobed compressor pumping schematic.

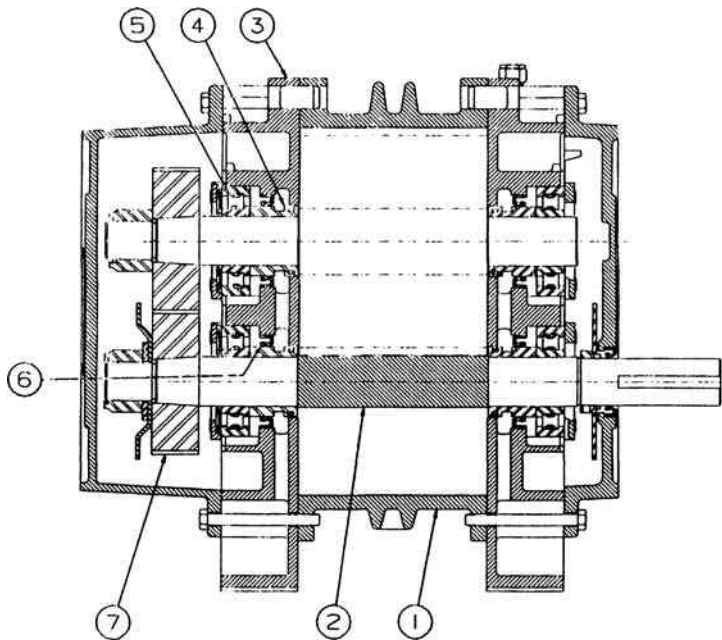


FIGURE 21.2 Cross section through straight lobe compressor.

to the discharge. This occurs while volume B is trapped between the driven rotor and the casing. Volume B becomes sealed off from the inlet and discharge ports when the driven rotor rotates counterclockwise to position 2. Volume B is still at the inlet conditions at this stage. The driven rotor discharges volume B in position 3. This occurs as the drive rotor is trapping volume A. Every rotation of the drive shaft discharges four equal volumes of gas from the compressor. The gas is not compressed inside the compressor. The system resistance downstream of the compressor determines the head.

21.3 Pulsation Characteristics

The trapped volume A in position 3 of Fig. 21.1 is suddenly exposed to the discharge pressure. This volume of gas was at the inlet pressure condition. This higher-pressure gas from the discharge end quickly rushes into volume A. This sudden rush of gas into the compressor generates a pressure pulse. The frequency of the pressure pulse is given by the following:

$$f = 2 \times N \times K$$

where f = pressure pulse frequency, Hz

N = compressor revolution per second

K = number of lobes

The pulse frequency in a two-lobe compressor is equal to four times the rotational speed of the compressor. Some compressor designs control the rate of the pressure increase in the trapped gas in volume A. This is done to reduce the magnitude of the discharge pulse.

Whispair design, Roots Operations is a division of Dresser Industries. This company has developed a modified version of a two-lobe compressor. This design uses discharge gas to increase the pressure in volume A gradually. This is done before exposing the gas in volume A to the discharge pressure.

Pulsation dampeners are required for most applications involving lobe compressors. These units are normally installed at the compressor discharge. They reduce the magnitude of the pressure pulsations. This prevents the equipment located downstream of the compressor from being damaged by these pulsations.

21.4 Noise Characteristics

Lobe compressors generate noise. This noise is caused by the following:

- Pulsation in the compressor discharge
- Gears
- Bearings
- Flowing gases

The discharge pressure pulsations are the main contributor to the noise generated by lobe compressors. The loads imposed on the bearings vary significantly during operation. This is due to the variation of stresses acting on the lobes. The bearing loads vary in magnitude and direction. This generates shock loading. The bearings transfer this shock loading to the mounting structures.

The vibrations of these structures generate noise. Lobe compressors are enclosed in an acoustic housing in some applications. This is done to reduce the noise generated by these compressors.

21.5 Torque Characteristics

The operation of straight lobe compressors generates a pulsating shaft torque. The torque pulsations vary within $\pm 10\%$ of the mean value. Stiff couplings are not recommended for lobe compressors. This includes gear-type couplings. These stiff couplings transmit the torque pulsations to the driver. These pulsations can damage the driver in some applications.

21.6 Construction

Figure 21.2 illustrates a cross section of a straight lobe compressor. The following are the main components of this compressor:

- Casing
- Rotors
- End plates (also known as head plates)
- Seals
- Bearings
- Timing gears

21.6.1 Rotors

The rotors consist of a set of two-toothed gears. The profile of the rotor lobes is normally involute. However, the profile of the lobes is cycloidal in some applications. The following two clearances are minimized:

1. Between the rotors
2. Between the casing and the rotor

This is done to reduce the leakages. The compressor efficiency drops significantly due to the increase in these leakages. Most rotors are hollow. However, they are plugged in dusty environments. This is done to prevent rotor imbalance.

21.6.2 Casing

The casing consists of the cylinder (item 1) and the end plates (item 3). The design pressure rating of the casing is 170 kPa (25 psi).

21.6.3 Timing Gears

The following are the purposes of the timing gears:

- Maintain the phasing of the rotors. The rotor phase is defined as the angle between the rotor and an axis used as a reference.
- Prevent contact between the rotors.

The timing gears are normally shrunk fit (interference fit) on the rotor.

21.6.4 Bearings

Lobe compressors normally use antifriction bearings. These bearings are also known as rolling contact bearings.

21.7 Staging

The following are the reasons for staging straight lobe compressors:

- Increase the final discharge pressure.
- Reduce the power consumption.

21.7.1 Higher Compression Ratios

The compression ratio of a single-stage straight lobe compressor is limited to 2.0. This corresponds to a pressure increase of 103.06 kPa (15 psi) if ambient air enters the compressor at 101 kPa_(abs) (14.7 psia). The increase in temperature across the compressor becomes higher than the design value if the compression ratio exceeds 2.0. Thus, intercooling is required between the stages to prevent the excessive increase in temperature generated by having a compression ratio above 2.0.

21.7.2 Power Reduction

The gas is not compressed inside a straight lobe compressor. Rectangle 1-2-4-3 shown on the P-V diagram of Fig. 21.3 represents the power required by a single-stage compressor. Line 1-2 of this rectangle represents the process of drawing the gas into the compressor at constant inlet pressure. Line 2-4 of this diagram represents the instantaneous increase in gas pressure at the compressor discharge. Line 4-3 represents the instantaneous discharge of the gas at constant pressure. Line 3-1 completes the cycle.

Area 2'-4'-4-5 of Fig. 21.4 represents the power reduction obtained by using intercooling between two compressors to achieve the same pressure increase shown in Fig. 21.3. Point 2' represents the discharge pressure of the first compressor.

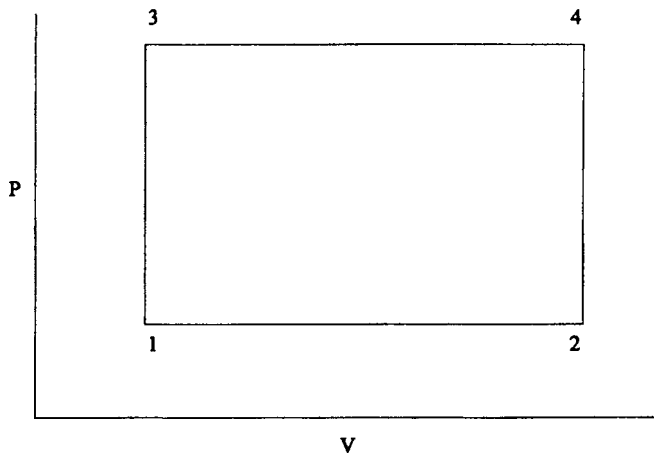


FIGURE 21.3 P-V diagram for single-stage compression.

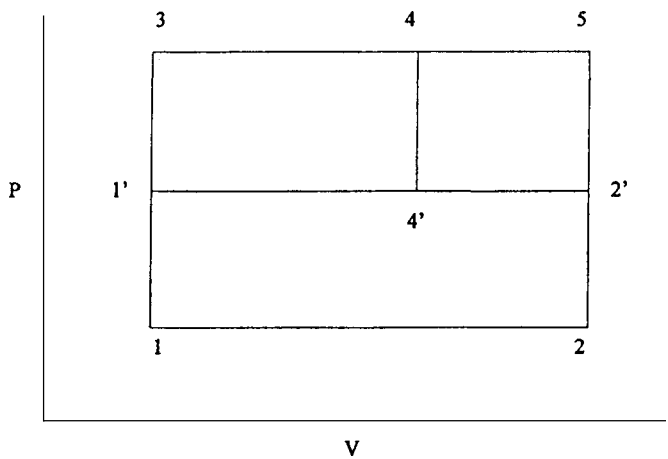


FIGURE 21.4 P-V diagram for two-stage compression.

Area 1-2-2'-1' represents the power required by the first compressor. The power required by the second compressor is shown by 1'-4'-4-3. The power saving (area 2'-4'-4-5) was obtained due to the reduction in gas volume from 2' to 4' in the intercooler.

21.8 Installation

Figure 21.5 illustrates the recommended arrangement of equipment at the suction and discharge of a straight lobe compressor. Resonance will occur if the discharge silencer is

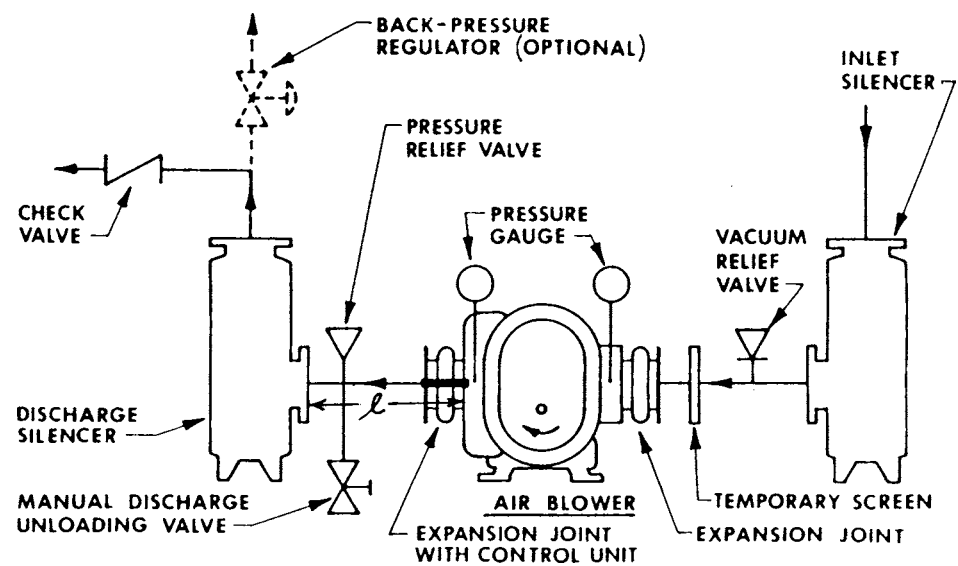


FIGURE 21.5 Suction and discharge arrangement for straight lobe compressor.

located at a specific distance from the compressor flange. Thus, the distance I between the discharge silencer and the compressor flange should be selected as follows to prevent resonance from occurring:

The distance I in m (ft) should be bigger or smaller than

$$\frac{nc}{4f}$$

where $n = 1, 3, 5, \dots$

c = velocity of sound in the gas in m/s (ft/s)

f = excitation frequency in Hz = $4 \times$ rotational speed in rev/s for two-lobe compressor = $6 \times$ rotational speed in rev/s for three-lobe compressor

k = ratio of specific heats

g = gravitational constant = 9.81 m/s^2 (32.16 ft/s^2)

R = gas constant in $\text{N} \cdot \text{m/K}$ ($\text{ft} \cdot \text{lb}/^\circ\text{R}$)

21.9 Bibliography

Hanlon, P. C., *Compressor Handbook*, McGraw-Hill, New York, 2001.

This page intentionally left blank

CHAPTER 22

Recent Developments in Separating Liquid from Gases

22.1 Introduction

The removal of liquids and solids from a gas stream is very important for most industries. The removal of these contaminants from the gas stream can reduce costly maintenance and downtime of the following three equipment:

1. Compressors
2. Turbines
3. Burners

Hydrocarbons and solid contaminants in the gas can also induce foaming in some cases. The lubricating oil used in some compressors leaks into the discharge gas in some applications. This oil contaminates the equipment located downstream of the compressor. A hydrocarbon film is deposited on heat exchangers due to this oil contamination. The thickness of this film increases during operation. This results in the following adverse consequences:

- Decrease in the effectiveness of the heat exchanger
- Increase in the plant energy consumption
- Increase in the risk of hot spots and leaks from the equipment

The following technologies are used to separate liquids and solids from the gas:

- Gravity separators
- Centrifugal separators
- Filter vane separators
- Mist eliminator pads
- Liquid/gas coalescers

22.2 Removal Mechanisms

There are different mechanisms used to remove liquids and solids from gases. They can be divided into the following categories:

1. *Gravity settling.* This method relies on using the weight of the droplets or particles (i.e., gravitational force) to separate them from the gas flow. This occurs when the weight of these droplets and particles exceeds the drag created by the flowing gas.
2. *Centrifugal separation.* This mechanism is more efficient than the previous one. It occurs when the centrifugal force applied on the droplets and particles exceeds the drag created by the flowing gas. The centrifugal force acting on these particles can be significantly higher than the gravitational force.
3. *Inertial impaction.* This mechanism relies on forcing the gas flow through a network of fibers and impingement barriers. The gas flows in this case through a tortuous path around these obstacles. The solid particles and liquid droplets flow through a straighter path. They collide with these obstacles. The velocity of the liquid droplets and solid particles drop due to this impact. Some of these liquid droplets and solid particles tend to coalesce. They eventually fall to the bottom of the vessel or remain trapped in the fiber medium.
4. *Diffusional interception or Brownian Motion.* This mechanism of separation occurs with very small aerosols. Their size is normally less than $0.1\text{ }\mu\text{m}$ (i.e., 10^{-7} m). The separation of these aerosols occurs when they collide with gas molecules. These collisions deviate these aerosols from the flow path of the gas around barriers. This increases the chance of having these aerosols strike the fiber surface. These aerosols can then be removed when the fiber surface is replaced.

The droplets and particles are normally measured in microns. One micron is $1/1000$ of a millimeter ($39 \times 10^{-6}\text{ in}$). Figure 22.1 illustrates the size of various materials in microns.

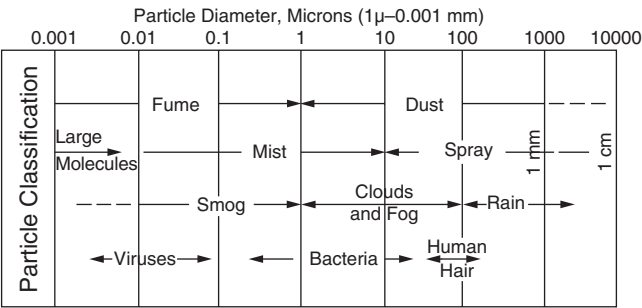


FIGURE 22.1 Size of various materials in microns. (Source: Reprinted with permission from Pall Corporation.)

22.3 Liquid/Gas Separation Technologies

22.3.1 Gravity Separators

Gravity separators are also known as knockout drums. This equipment relies on gravitational forces to control separation. The efficiency of the liquid/gas separation increases with the following:

- Decrease in the gas velocity
- Increase in the size of the vessel

Gravity separators are normally designed to remove droplets greater than 300 μm . This is due to the large size of the vessel required to achieve settling. The gravity separator is normally used as a first-stage scrubber. Gravity separators are used in conjunction with other separation equipment in applications requiring high separation efficiency.

22.3.2 Centrifugal Separators

Centrifugal separators are also known as cyclone separators. The centrifugal force that acts on an aerosol in these units can be several times larger than gravity. Cyclone separators are normally used for removing aerosols larger than 100 μm in diameter. However, a properly sized unit can be used for removing aerosols as low as 10 μm . The efficiency of cyclones is very low in applications requiring the removal of mist particles smaller than 10 μm . A combination of knockout drums and cyclones is recommended for applications involving waxy or coking materials.

22.3.3 Mist Eliminators

Mist eliminators rely on a separation mechanism known as inertial impaction. Mist eliminator pads normally consist of fibers or knitted meshes. They can remove droplets down to 1 to 5 μm . However, the vessel containing them is normally relatively large. This is because the gas velocity in it must be low to prevent liquid re-entrainment.

22.3.4 Filter Vane Separators

Vane separators consist of a series of baffles or plates within a vessel. This equipment controls separation using inertial impaction as well. The advantage of this equipment is that it can operate at higher velocities than mist eliminators. This is due to a more effective liquid drainage system. This feature reduces the chance of liquid re-entrainment. However, these units can only remove droplets larger than 10 μm . This is due to the relatively large tortuous path between the plates. These units are normally used to retrofit mist eliminators in applications where the gas velocity exceeds the design velocity.

22.3.5 Liquid/Gas Coalescers

The liquid/gas coalescer cartridges combine the features of both mist eliminator pads and vane separators. However, they are not usually specified for removing bulk liquids. A high-efficiency coalescer (Fig. 22.2) is normally placed downstream of a knockout drum or impingement separator in bulk liquid systems. The gas flows in these units

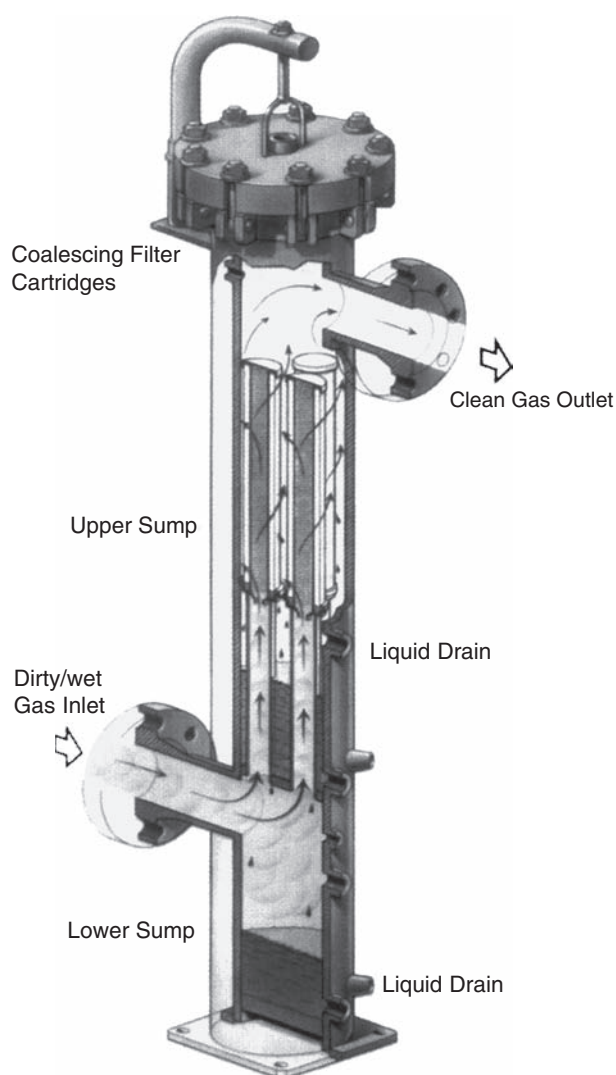


FIGURE 22.2 High-efficiency coalescer. (Source: Reprinted with permission from Pall Corporation.)

through a very fine pack of bound fibrous material with a wrap on the outer surface. This is done to promote liquid drainage. The cartridge in these units can trap droplets down to 0.1 μm . The gas velocity in this equipment can be much higher than mist eliminators and vane separators. This is due to the drainage of the coalesced droplets from the fibrous pack. Liquid/gas coalescers do not experience the following problems:

- Liquid re-entrainment
- Increase in pressure drop across the assembly

Technology	Droplet Size Removed
Gravity separator	Down to 300 μm
Centrifugal separator	Down to 8–10 μm
Mist eliminator pad	Down to 10 μm
Vane separator	Down to 10 μm
High-efficiency L/G coalescer	Down to 0.1 μm

TABLE 22.1 Types of Liquid/Gas Separators

22.3.6 Selection of Liquid/Gas Separation Equipment

Table 22.1 summarizes the capability of the various types of liquid/gas separators. A coalescer should be selected for applications containing aerosols smaller than 5 μm . The removal of very fine aerosols from gases provides the following benefits in compressor systems:

- Reduction in the operation and maintenance cost
- Increase in reliability

22.4 Formation of Fine Aerosols

Fine liquid aerosols can enter the gas system by the following methods:

- Condensation from saturated vapor
- Atomization of a liquid flow through a restriction (spray effect)
- Liquid re-entrainment

Recent studies confirmed that the aerosol droplets are smaller than 5 μm in the following applications:

- Choke flow through valves and other restrictions
- Vapors at their dew points

Figure 22.3 illustrates measurements performed to determine the concentration of liquid aerosols in a natural gas stream. The samples were taken from this gas stream downstream of vane separators (combination of a gravity separator and a horizontal filter barrier. It is equivalent to a mist eliminator pad). These results prove that in many applications, large quantities of aerosols can pass through this type of separation equipment. This is because it does not have the capability to capture very small aerosols. Thus, a liquid/gas coalescer should be selected for this application.

22.5 Ratings and Sizing of Separation Equipment

Coalescers have several advantages over filters. This is because they perform the following functions:

- Filtration of fine solid particles
- Coalescence of liquid aerosols from the gas stream

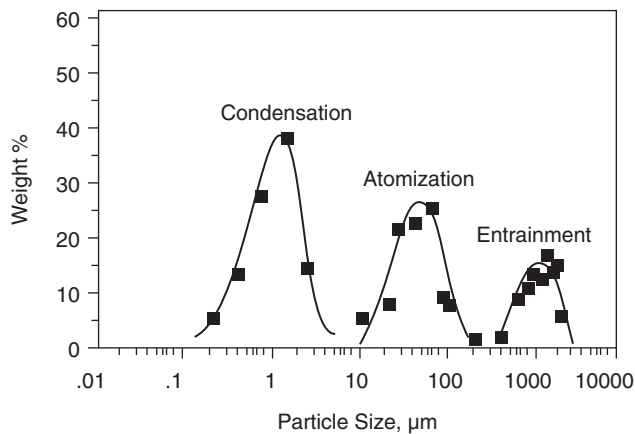


FIGURE 22.3 Aerosol size distribution. (Source: Reprinted with permission from Pall Corporation.)

The effectiveness of a coalescer depends on its size and rating. The following problems will occur if the coalescer is undersized:

- Continuous liquid re-entrainment.
- Very low liquid separation efficiency.
- The coalescer will become vulnerable to any process changes.

Figure 22.4 illustrates the importance of proper coalescer sizing. This figure shows that the coalescer performance deteriorates quickly when it is challenged by a high

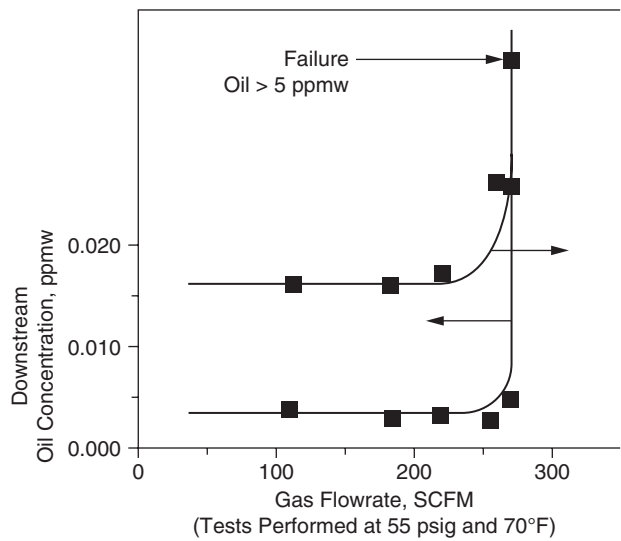


FIGURE 22.4 Coalescer efficiency change versus gas flow rate. (Source: Reprinted with permission from Pall Corporation.)

liquid flow. This can also occur if the gas stream has high concentration of aerosols. Thus, the characteristics of coalescers are different from most other separation equipment. The performance in most remaining separation equipment diminishes gradually when the unit operates beyond its rated capacity.

22.6 Bibliography

Pall Corporation, Scientific & Technical Report. GAS-4310b, "Recent Developments in Liquid/Gas Separation Technology," 1994.

This page intentionally left blank

Dynamic Compressors Technology

23.1 Introduction

Dynamic compressors operate based on the following principles:

- Increasing the velocity of the gas
- Converting this increase in velocity into pressure increase

These units are also known as turbocompressors. Centrifugal machines are used in more than 80% of applications involving dynamic compressors. Axial-flow machines are used in the remaining 20% or fewer of applications. They are normally used in applications requiring higher flows and lower pressures than centrifugal compressors.

23.2 Centrifugal Compressor Overview

Centrifugal compressors provide relatively reliable, and trouble-free service. They can compress most gases. They are used in a wide range of flow and pressure applications. Figure 23.1 illustrates a single-stage centrifugal compressor. It includes integral step-up gearing. It can also be direct-driven. Figure 23.2 illustrates multistage centrifugal compressors configuration. Centrifugal compressors can have a horizontally split casing (Fig. 23.3). They can also have a vertically split casing (barrel-type compressors) as shown in Fig. 23.4. There is also a wide range of nozzle configurations for these compressors.

These machines are designed to handle the following conditions:

- Intake volumetric flow rate from 500 to 200,000 m³/h (294–117,000 ft³/min)
- Discharge pressure up to 16 MPa (2352 psi)

Barrel-type compressors are designed for higher pressures. They have operated successfully in these applications. These compressors are normally driven by any of the following equipment:

- Steam turbine
- Gas turbine
- Electric motor (directly or using gears or variable speed drives)

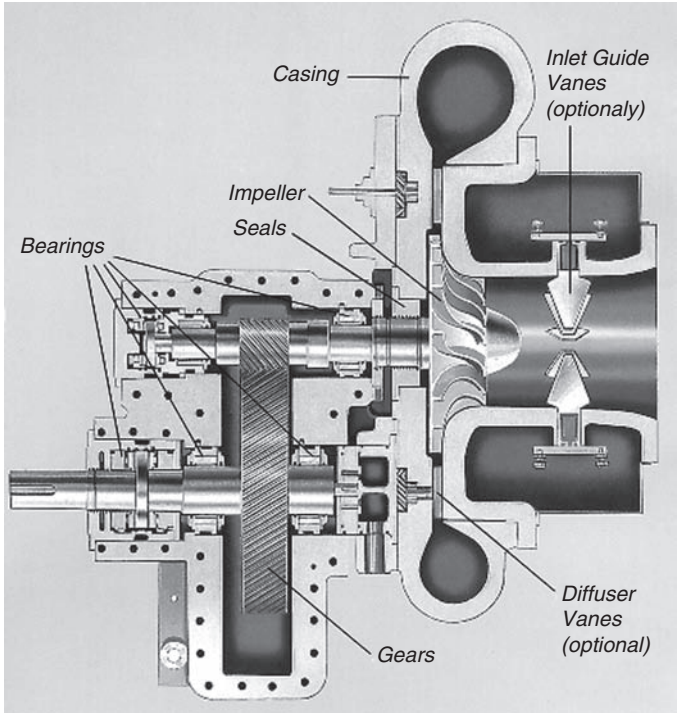


FIGURE 23.1 Single-stage centrifugal compressor with integral step-up gearing. (Source: Dresser-Rand Company, Olean, N.Y.)



FIGURE 23.2 Multistage centrifugal compressor in a petrochemical plant. (Source: Elliott Company, Jeannette, Pa.)

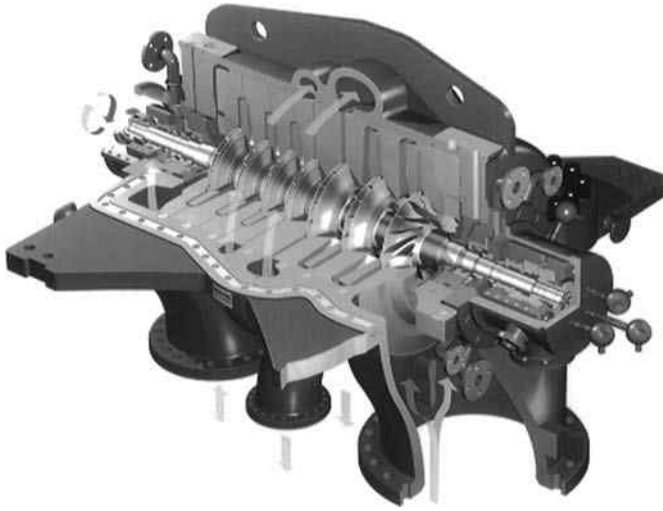


FIGURE 23.3 Centrifugal compressor with horizontally split casing construction.
(Source: Mannesmann-Demag, Duisburg, Germany.)

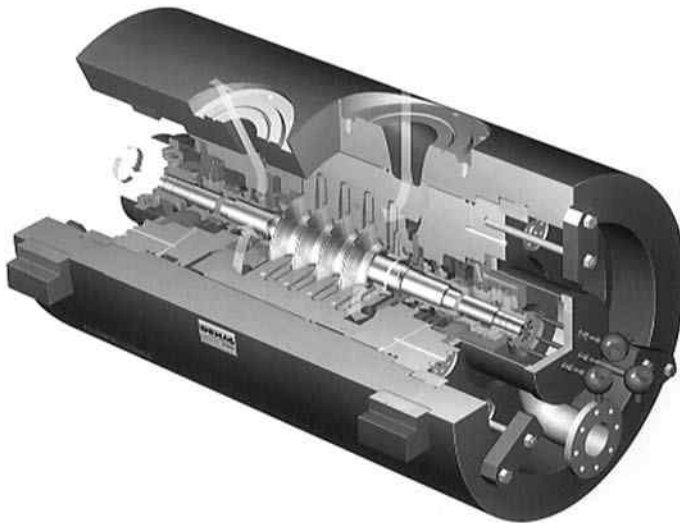


FIGURE 23.4 Centrifugal compressor with vertically split (also called radially split) design.
(Source: Mannesmann-Demag, Duisburg, Germany.)

The following shaft seals are used for these compressors:

- Labyrinth seals
- Mechanical contact seals
- Floating seals
- Dry gas seals

Centrifugal compressors can be used to compress any gas.

23.3 Axial Compressors Overview

Axial-flow compressors (Figs. 23.5 and 23.6) are designed to handle large volumetric flow rates. The size of their casing is relatively small. Their power requirement is relatively low

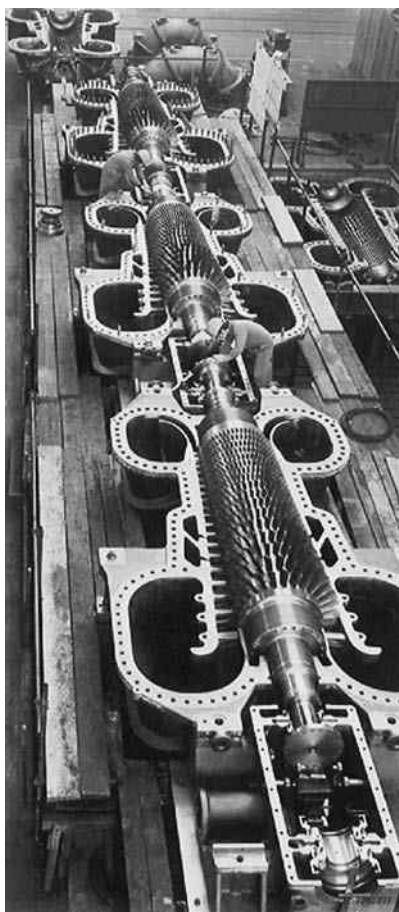
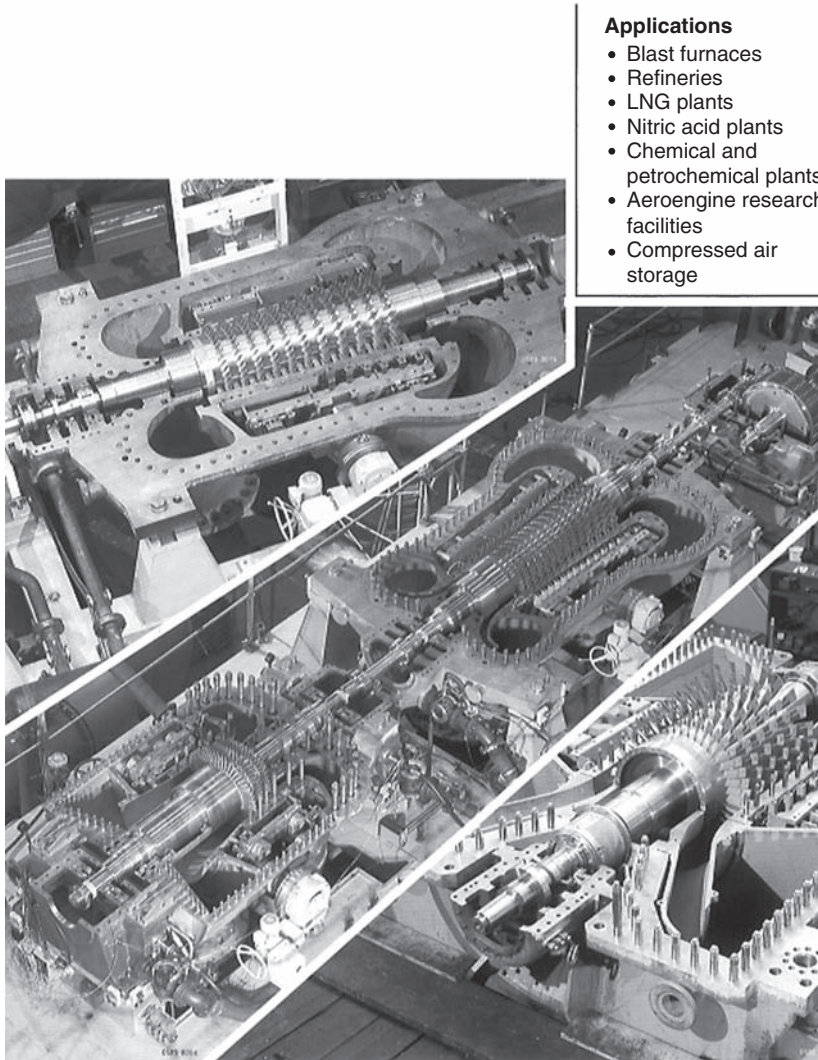


FIGURE 23.5 Axial-flow compressor set for an aircraft test bed in France. These machines can be used to generate compressed air or vacuum. The installation comprises six identical axial compressors. Capacity: $244,000 \text{ N} \cdot \text{m}^3/\text{h}$ in compression; $38,000 \text{ N} \cdot \text{m}^3/\text{h}$ in vacuum mode. (Source: Reprinted with permission from MAN Diesel & Turbo SE.)

**Applications**

- Blast furnaces
- Refineries
- LNG plants
- Nitric acid plants
- Chemical and petrochemical plants
- Aeroengine research facilities
- Compressed air storage

FIGURE 23.6 Typical axial-flow compressor.

(i.e., their isentropic efficiency is very high; it is around 89%). They are designed to handle the following conditions:

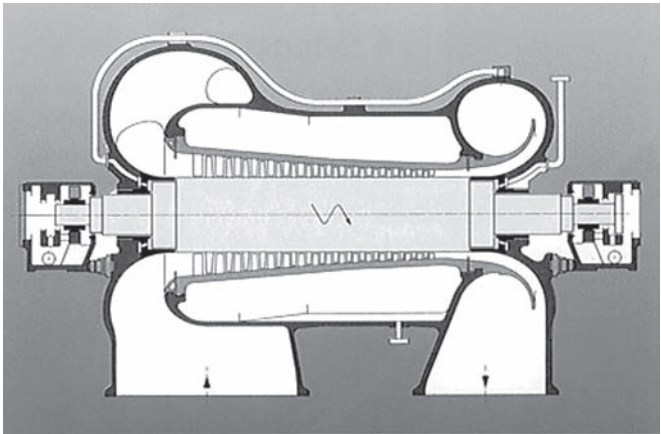
- Discharge pressure lower than 15 bar (218 psi)
- Intake volumetric flow rate between 40,000 and 1,000,000 m³/h (23,500 and 588,500 ft³/min)

Axial-flow compressors are used in the following applications:

- Gas turbines
- Blast furnace air

- Air separation services
- Nitric acid plants
- Natural gas liquefaction, etc.

Figures 23.7 and 23.8 illustrate the typical performance maps for these compressors.



Series A with fixed stator blades (FIXAX).

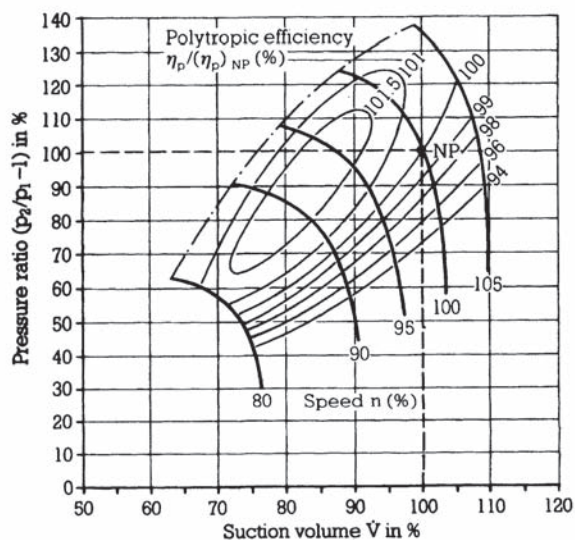
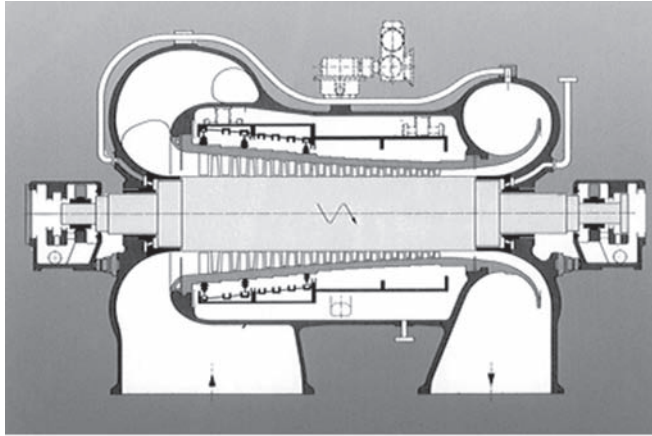


FIGURE 23.7 Performance maps for axial compressors with speed variation. (Source: Reprinted with permission from MAN Diesel & Turbo SE.)



Series A with adjustable stator blades (VARAX).

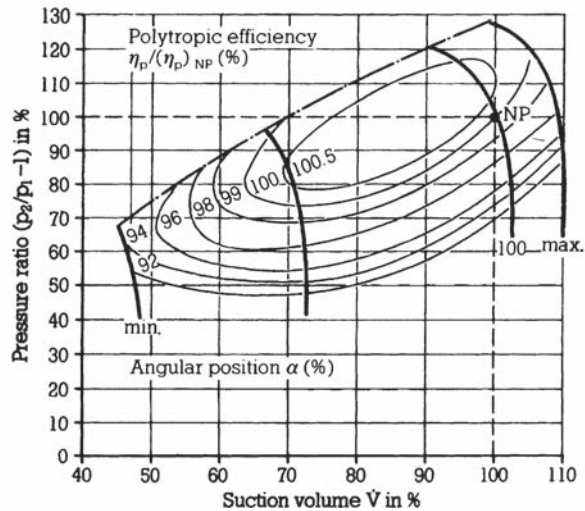


FIGURE 23.8 Performance maps for axial compressors with adjustable stator blades.
(Source: Reprinted with permission from MAN Diesel & Turbo SE.)

23.4 Bibliography

Bloch, H. P., *A Practical Guide to Compressor Technology*, 2nd ed., John Wiley & Sons Inc., Hoboken, New Jersey, 2006.

This page intentionally left blank

CHAPTER 24

Simplified Equations for Determining the Performance of Dynamic Compressors

This chapter provides the fundamental equations used to determine the following three parameters for centrifugal and axial gas compressors:

1. Brake horsepower
2. Operating speed
3. Discharge temperature of the centrifugal and axial gas compressors

24.1 Nonoverloading Characteristics of Centrifugal Compressors

The majority of impellers used on centrifugal compressors have backward-leaning vanes. The discharge pressure of these impellers decreases gradually as the capacity increases, while the compressor operates at constant speed. Thus, it is impossible to overload the prime mover of these compressor. This is because the head and brake horsepower will decrease significantly as the capacity increases.

24.2 Stability

Dynamic compressors experience a surge. This phenomenon is described in detail in subsequent chapters. The stability of a centrifugal compressor is defined as the percent change in capacity between the rated capacity and the surge point at rated head (Fig. 24.1).

24.3 Speedy Change

Figure 24.1a shows that the head varies significantly with a shift change in speed. Figure 24.2 illustrates the head versus flow relationship for operation that does not involve change in speed.

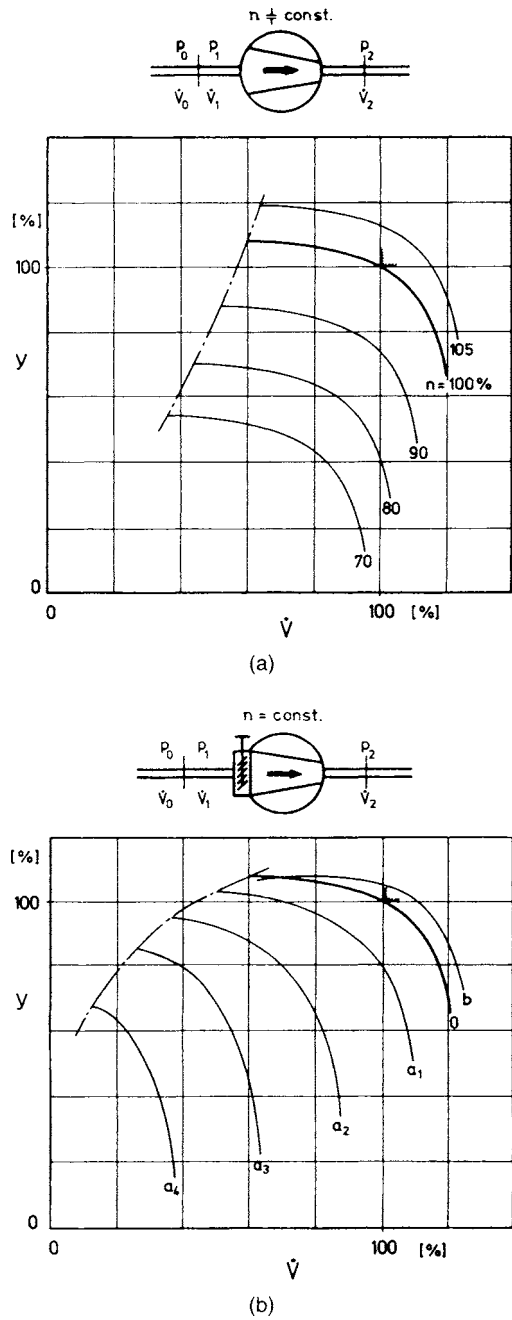


FIGURE 24.1 Performance of dynamic compressors: (a) variable speed; (b) variable inlet guide vanes; (c) suction valve throttling. (Source: Sulzer, Ltd., Winterhur, Switzerland.)

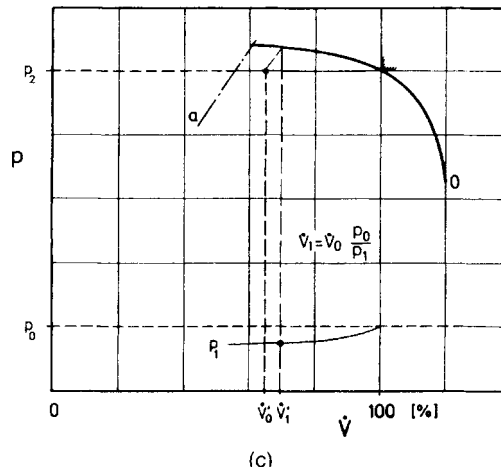


FIGURE 24.1 (Continued)

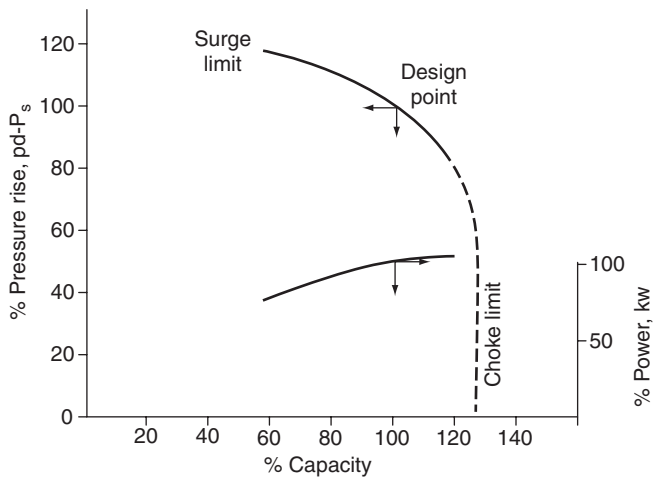


FIGURE 24.2 Head capacity curve for a centrifugal compressor. (Source: Dresser-Rand Company, Olean, NY.)

24.4 Compressor Drive

The following are the prime movers for centrifugal compressors:

- Steam turbine
- Electric motor
- Expansion turbine (expander)
- Combustion gas turbine

The selection of the driver is based on the following factors:

- Water requirements (steam consumption)
- Operating speeds
- Process control
- Process steam supply
- Fuel or energy costs
- Reliability

Variable inlet guide vanes or suction throttling are available for electric motors. Hydraulic couplings or variable-frequency drives are used with electric motors to provide variable-speed operation.

24.5 Calculations

The following three parameters are normally determined in centrifugal compressor calculations:

1. Shaft horsepower
2. Operating speed
3. Discharge temperature

The head is given by the following (just as for a liquid pump):

$$H = k_1 \int V dP$$

where H = head, ft (m)

V = specific volume, ft³/lb (m³/kg)

P = pressure, psia (bar, absolute)

K_1 = different constants, for English or metric conversions and expressions

The specific volume or density is constant for a liquid pump. Thus, the head is given by:

$$H = k_1 V (P_2 - P_1) = \frac{k_1 (P_2 - P_1)}{\rho}$$

where ρ is the density in kg/m³ or lb/ft³.

The specific volume is variable in compressors. The following equation can be used for polytropic compression:

$$PV^n = \text{constant}$$

Thus,

$$H = \frac{k_1 P_1 V_1}{(n-1)/n} \left[\left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right]$$

where n is the polytropic exponent of compression.

The head can also be expressed as follows:

$$H = \frac{ZRT_1}{(n-1)/n} \left[\left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right]$$

where R = gas constant = 154/MW or 8314/MW

T_1 = suction temperature, °R or K

Z = average compressibility

For simplification of calculation, the head can be expressed in this form:

$$H = k_1 P_1 V_1 \beta = ZRT_1 \beta$$

where $\beta = [(P_2/P_1)^M - 1]/M$

$$M = (n-1)/n$$

These equations indicate that the head and hence the horsepower vary follows:

1. Proportional to the absolute suction temperature
2. Inversely proportional to the molecular weight of the gas

Since there is a limited value for the head developed by a single impeller, gases having high molecular weight require fewer impellers than those having low molecular weight to reach the same discharge pressure (Fig. 24.3).

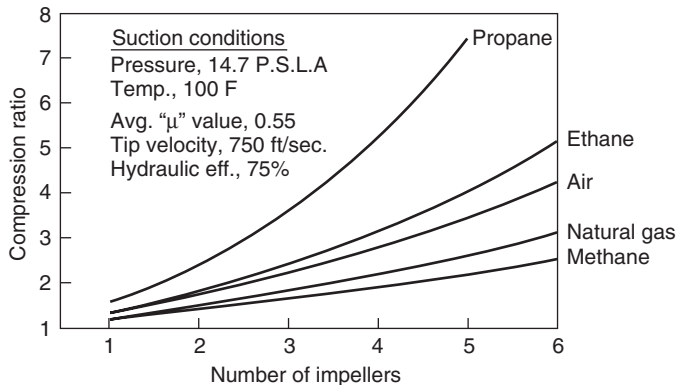


FIGURE 24.3 Compression ratio versus number of impellers. (Source: Dresser-Rand Company, Olean, NY.)

The hydraulic efficiency is given by the following:

$$\eta = \frac{\int V dP}{\Delta h} = \frac{\frac{P_1 V_1}{(n-1)/n} \left[\left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right]}{\frac{P_1 V_1}{(K-1)/K} \left[\left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right]}$$

$$\eta = \frac{(K-1)/K}{(n-1)/n}$$

where η = hydraulic efficiency

Δh = change in enthalpy, Btu/lb or kJ/kg

K = isentropic exponent (C_p/C_v)

The hydraulic efficiency is determined by tests. It depends generally on the capacity at the suction of the compressor.

The head developed by a centrifugal compressor stage (e.g., and impeller and diffuser) is given by the following:

$$H = \mu \frac{u^2}{g}$$

where μ = pressure coefficient

u = peripheral velocity, ft/s or m/s

g = gravitational constant, 32.2 ft/s² or 9.81 m/s²

The value of the pressure coefficient depends on the design of the stage. The average value for one stage of a multistage centrifugal compressor is 0.55. For a peripheral velocity of 235 m/s (770 ft/s), the head pre stage is 3050 m (10,000 ft). This allows determination of the number of stages required to develop the head of the compressor.

The power required by the compressor is given by:

$$\text{ghp} = \frac{W \Delta h}{33,000}$$

or

$$\text{kW} = \frac{mH}{3600}$$

where ghp = gas horsepower

W = gas flow, lb/min

kW = gas power, kW

m = mass flow, kg/h

H = differential head, m

However,

$$\Delta h = \frac{\int V dP}{\eta} = \frac{H}{\eta}$$

Therefore,

$$\text{ghp} = \frac{WH}{33,000\eta}$$

and

$$\text{kW} = \frac{mH_p}{3600\eta}$$

The frictional losses (e.g., bearings) are normally less than 1% of the gas horsepower. The discharge temperature for uncooled compressor is given by:

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^M$$

where $M = (n - 1)/n$

24.6 Bibliography

Hanlon, P. C., *Compressor Handbook*, McGraw-Hill, New York, 2001.

This page intentionally left blank

CHAPTER 25

Centrifugal Compressors— Components, Performance Characteristics, Balancing, Surge Prevention Systems, and Testing

25.1 Introduction

Centrifugal compressors are dynamic machines. The energy is transferred from the rotor to the gas. The gas is discharged from the impeller at a high velocity. The diffuser converts this increase in velocity into pressure.

25.2 Casing Configuration

Centrifugal compressors can be used for a wide range of applications. The number of impellers varies from one to ten in one casing. This number depends on the head and flow requirements. Figures 25.1 through 25.9 illustrate common casing configurations of centrifugal compressors.

25.3 Construction Features

Figure 25.9 illustrates the major elements of a centrifugal compressor. These elements include the following:

- Inlet nozzle
- Inlet guide vanes
- Impeller
- Radial diffuser

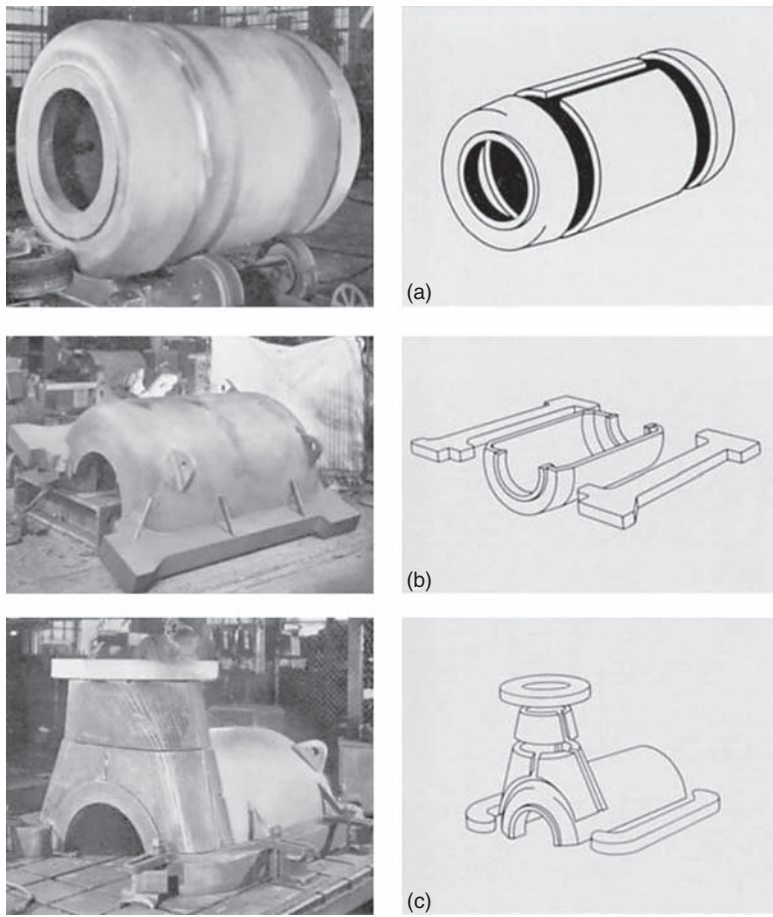
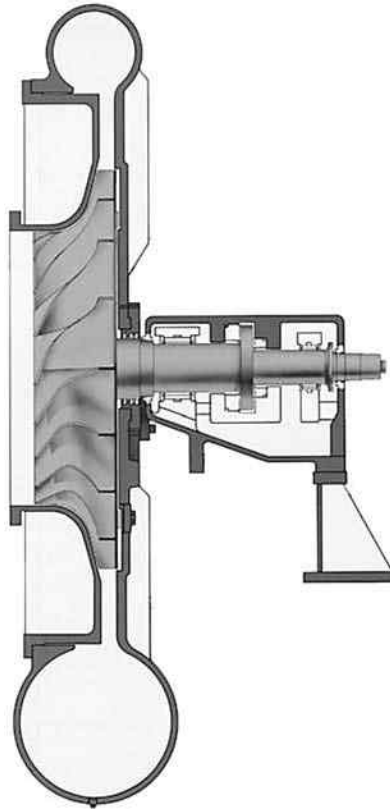


FIGURE 25.1 The fabricated compressor casing is a flexible design for accommodating the specific requirements of the application, including special nozzle arrangements. (a) The center section, a cylindrical shell of rolled and welded steel plate is joined by welding to two dished heads formed by hot spinning. Photo shows automatic girth welding of heads to shell. (b) Assembly is cut longitudinally into two equal halves and sturdy flanges are welded to each half of the shell. (c) Holes are burned in the shell and fabricated sections are welded together to form inlet, discharge, and sidestream nozzles.

- Return channel
- Collector volute
- Discharge nozzle

The inlet nozzle accelerates the gas. It directs the gas stream into the inlet guide vanes. These vanes could be either fixed or adjustable. The inlet nozzle is normally radially oriented on multistage compressors (Fig. 25.10). The inlet guide vanes are required in this case to distribute the flow evenly to the first-stage impeller. The inlet nozzle is generally installed axially in single-stage compressors. The inlet vanes may not be

FIGURE 25.2 Basic single-stage compressor. Typical construction is impeller overhung from the bearing housing. (Courtesy of Elliott Group, Ebara Corporation.)



required in this case. The gas leaving the impeller flows through the diffuser in a spiral manner. The return channel vanes (Fig. 25.11) straighten the flow before it enters the next impeller.

25.3.1 Diaphragms

The diaphragm is a stationary component. The shaded parts of Fig. 25.12 illustrate a diaphragm. It forms the following:

- Half of the diffuser wall of the previous stage
- Part of the return bend
- The return channel
- Half of the diffuser wall of the next stage

The flow leaving the diffuser of the last stage enters the discharge volute (Fig. 25.13). The volute is normally designed to have constant angular momentum ($RiVi = \text{constant}$). The gas leaving the volute enters the discharge nozzle. The velocity of the flow is reduced in the discharge nozzle. The flow leaving the discharge nozzle enters the process piping.

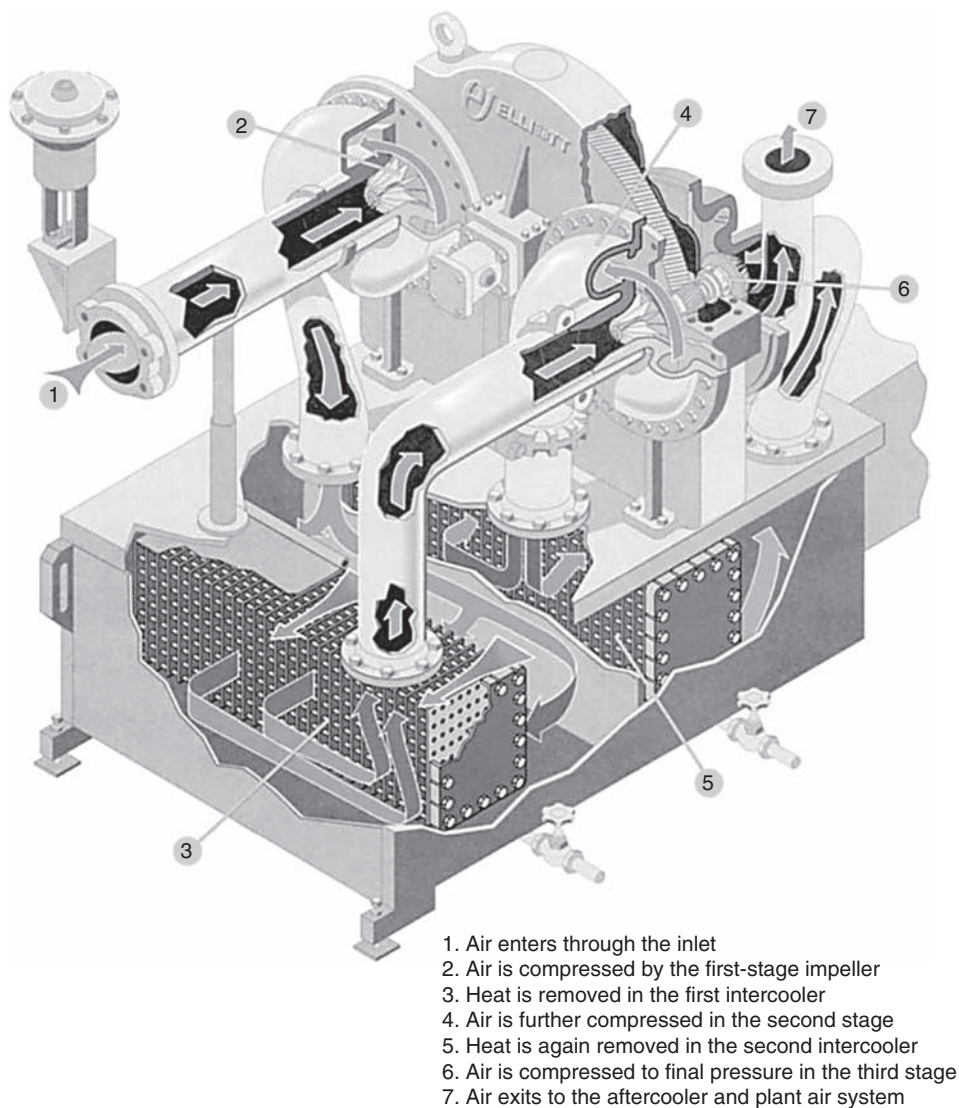


FIGURE 25.3 For high-speed single-stage compressors, maximum efficiency can be realized with intercooling between each stage. Also, cooling is required to keep operating temperatures below material limitations. (Courtesy of Elliott Group, Ebara Corporation.)

The flow velocity in the diffuser section is relatively high. It is more than 100 m/s. The surface finish of the diffuser section has a significant effect on the polytropic efficiency of the compressor. This is because the surface finish has a significant effect on the friction factor of the diffuser section.

Many applications experience dirt or polymer buildup on the surface of the diaphragm and impellers. This will give these surfaces a rough finish. Polymer buildup

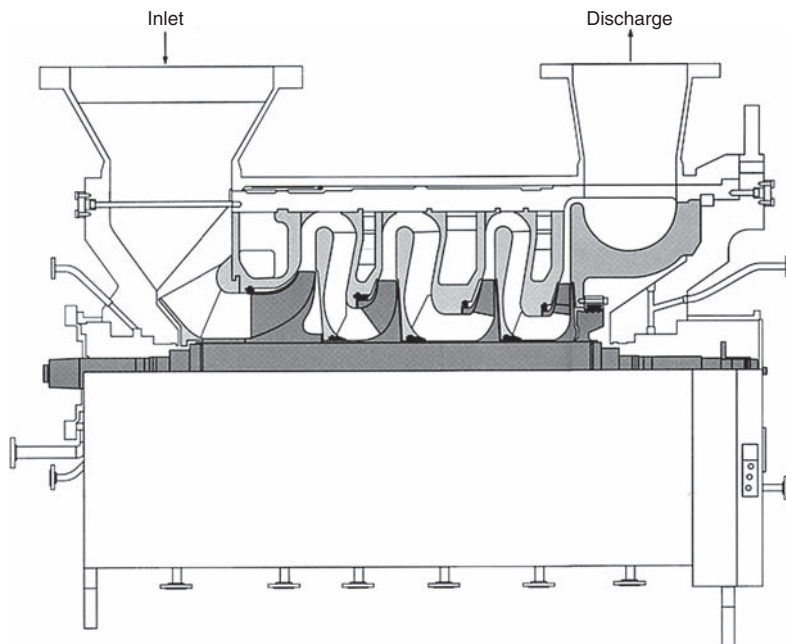


FIGURE 25.4 Basic straight-through multistage compressor with a balancing piston. This arrangement may employ 10 or more stages of compression. This arrangement is most often used for low-pressure rise process gas compression. Casing design shown is a barrel construction used for high-pressure or low-mol weight gases which provides limited leakage areas and thus better contains the process gas. (Courtesy of Elliott Group, Ebara Corporation.)

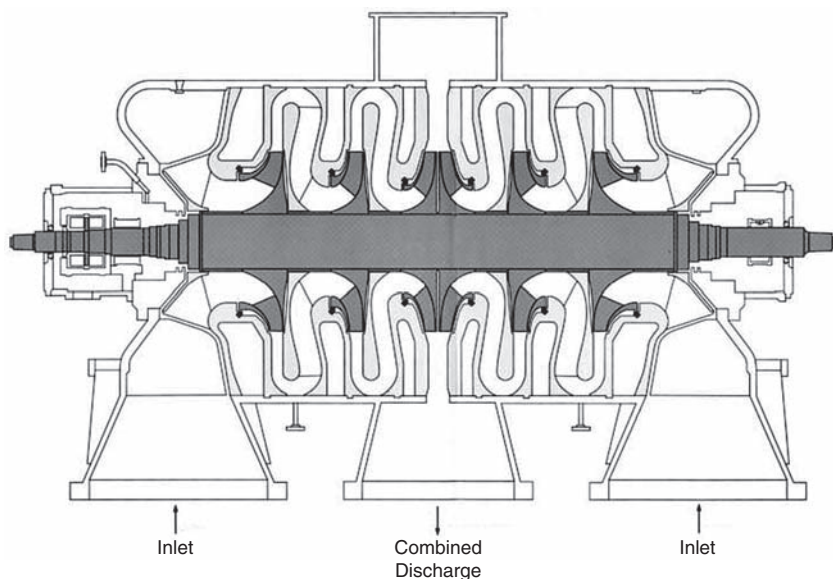


FIGURE 25.5 Double-flow compressor. This arrangement is used to double the maximum flow capability for a compressor frame. Since the number of impellers handling each inlet flow is only half of that of an equivalent straight-through machine, the maximum head capability is reduced accordingly. (Courtesy of Elliott Group, Ebara Corporation.)

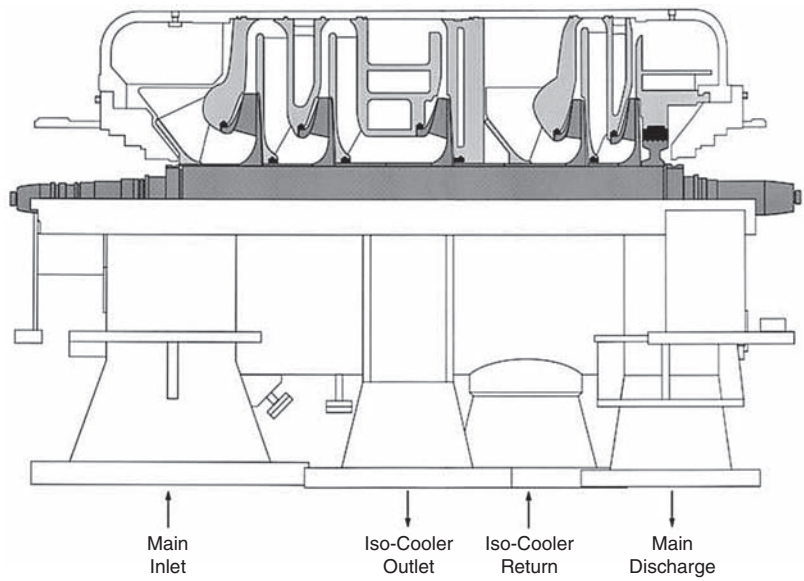


FIGURE 25.6 As in Fig. 25.3, cooling is required to keep operating temperatures below material or process limits as well as to improve operating efficiency. Iso-cooling nozzles permit the hot gas to be extracted from the compressor to an external heat exchanger, and then returned to the following stage at reduced temperature for further compression. (Courtesy of Elliott Group, Ebara Corporation.)

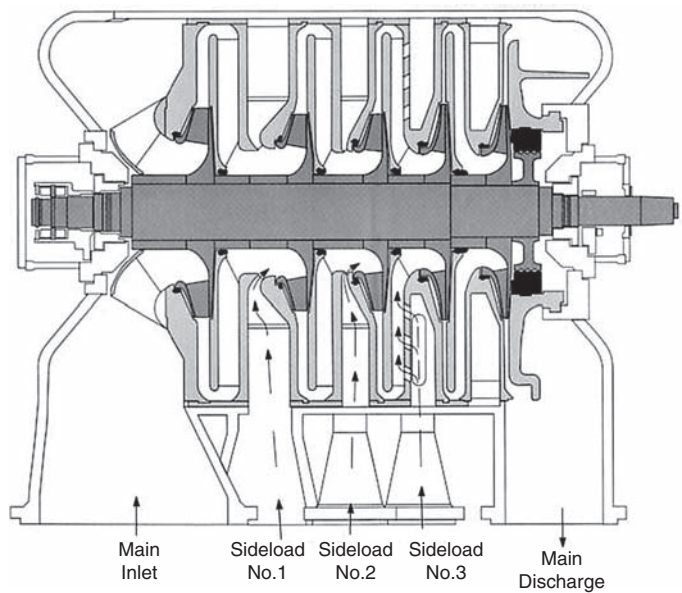


FIGURE 25.7 Side stream nozzles permit introducing or extracting gas at selected pressure levels. These flows may be process gas streams or flows from economizers in refrigeration service. Sideloads may be introduced through the diaphragm between two stages (sideload 3), or if the flow is high as in sideloads 1 and 2, the flow may be introduced into the area provided by omitting one or two impellers. (Courtesy of Elliott Group, Ebara Corporation.)

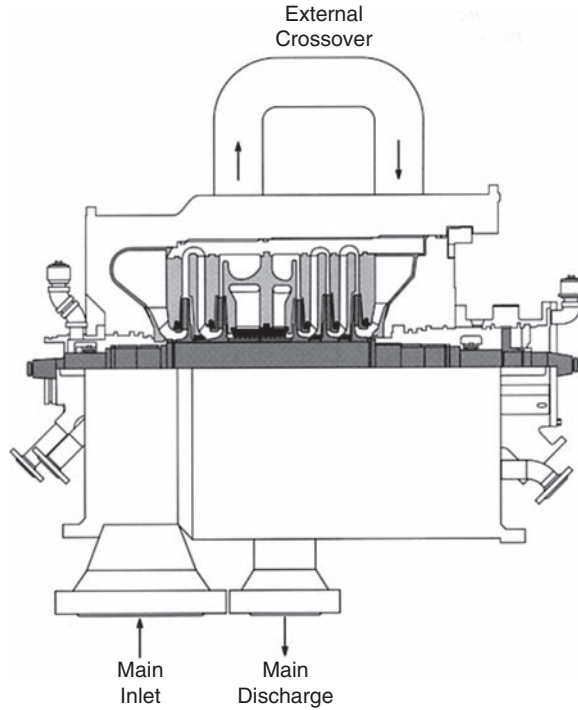


FIGURE 25.8 The back-to-back design minimizes thrust when a high-pressure rise is to be achieved within a single casing. Note that the thrust forces acting across the two sections act in opposing directions, thus neutralizing one another. (Courtesy of Elliott Group, Ebara Corporation.)

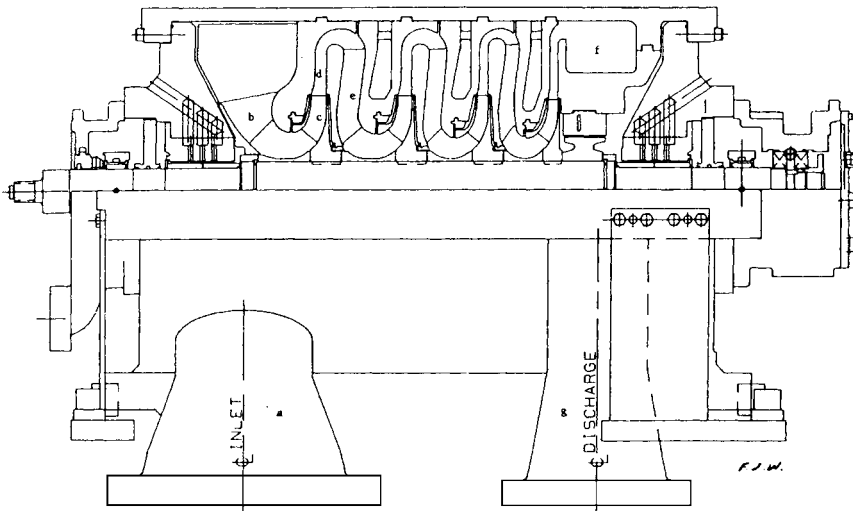


FIGURE 25.9 Major elements of a multistage centrifugal compressor: (a) inlet nozzle, (b) inlet guide vanes, (c) impeller, (d) radial diffuser, (e) return channel, (f) collector volute, and (g) discharge nozzle.

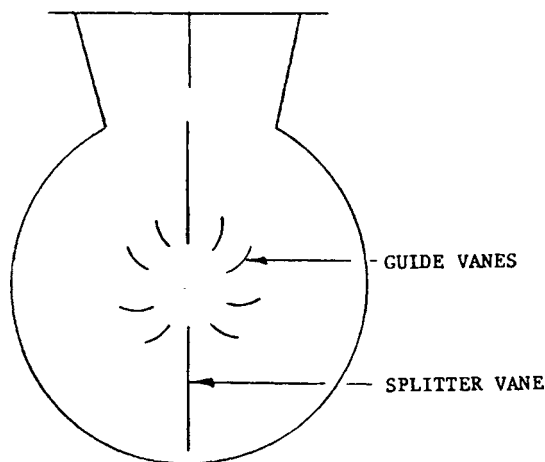


FIGURE 25.10 Multistage compressor inlet showing splitter vanes and guide vanes.

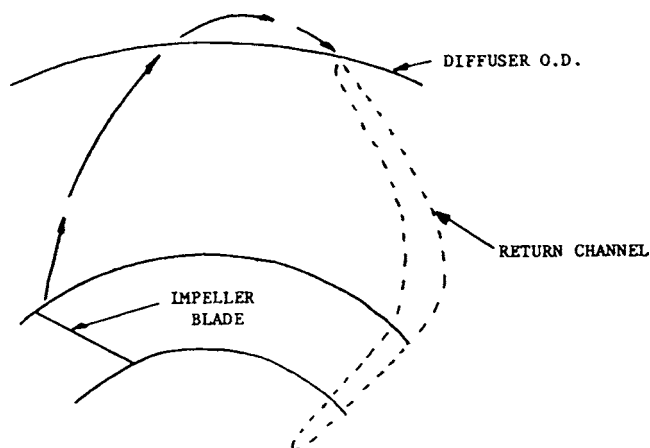


FIGURE 25.11 Flow path of gas from tip to return channel.

has also caused severe restriction in the diffuser passage in some applications. The pressure losses across the diffuser section increases due to both conditions. This will result in a reduction in the compressor isentropic efficiency (also known as compressor polytropic efficiency). The chemical process required to generate polymerization is not well understood. However, experience has shown that polymers form under certain conditions. They bond tenaciously to the metal of the compressor components. The following factors have been found to enhance the fouling process:

- Temperature above 90°C (194°F)
- Increase in pressure
- Increase in the concentration of reactable hydrocarbons in the process gas
- Increase in the roughness of the component surface finish

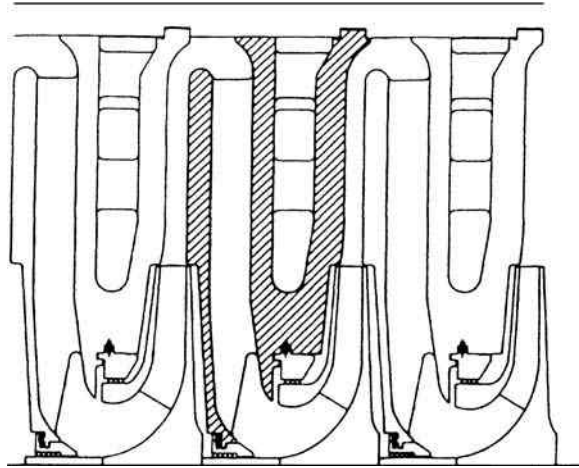


FIGURE 25.12 Multistage centrifugal compressor diaphragm.

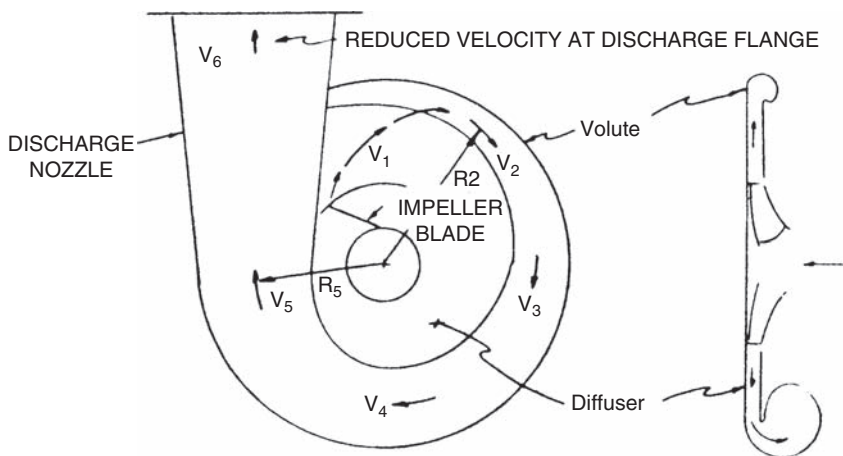


FIGURE 25.13 Discharge volute.

Water injection is used to reduce the operating temperature of the process gas. Water can be injected at each stage of the compressor. It should be injected through atomizing spray nozzles. The amount of water injected into the compressor should bring the gas to just below the saturation level. Nonstick coating should be applied to all the critical components inside the compressor. This coating can enhance and preserve the surface finish of these components. The material of this coating includes the following:

- Fluorocarbon-based (Teflon)
- Corrosion-resistant coating such as electroless nickel

Some manufacturers apply multiple coatings on the critical components of the compressor. The layers of coatings include the following:

- Barrier coating
- Inhibitive coating
- Sacrificial coating

This combination of coating layers has provided the best long-term service. Figure 25.14 illustrates the effects of coating on the internal components of the compressor.

The compressor performance is maintained by providing a wash system. This system involves the injection of either of the following two solutions into the compressor:

- Water with detergent
- Hydrocarbon solvents

These solutions wet the aerodynamic surfaces inside the compressor. This provides the following two functions:

- Prevents the attachment of the polymers to the surfaces
- Washes the compressor surfaces when the polymers have bonded to them

The amount of solution (wash liquid) injected should be limited to 3% of the gas mass flow rate. This is done to prevent erosion of the components. The solution should be injected at every stage of the compressor. Higher amounts of the solution should be injected near the discharge of the compressor. This injection is performed online. The

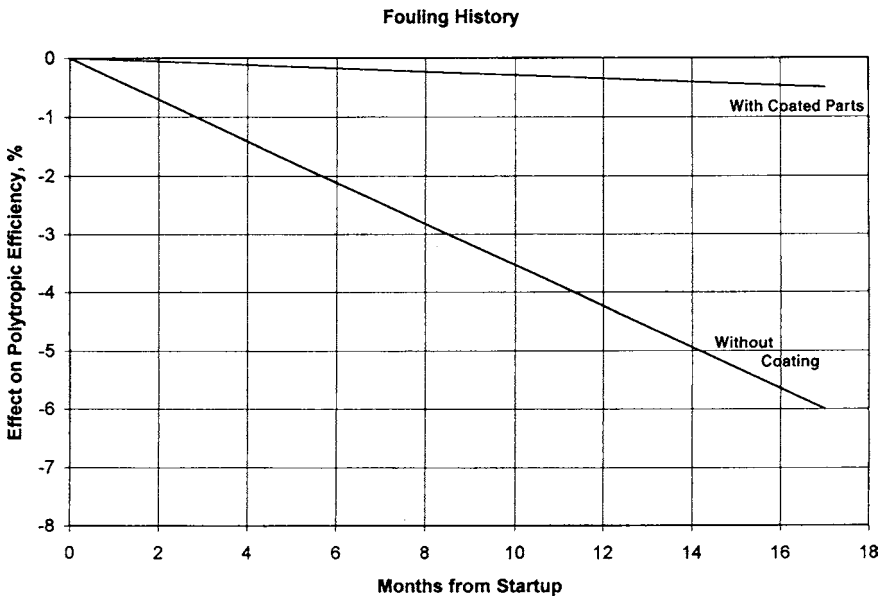


FIGURE 25.14 Effect of coated and non-coated surfaces on an ethylene feed gas compressor.

compressor can also be cleaned by injecting a special solution into it off-line. This solution should be allowed to soak into the polymers for a few hours. The compressor should then be rinsed with water. This method has proven to provide satisfactory results. The compressor can also be cleaned by hand using a water and detergent solution when it is disassembled. This method provides the best overall results. The compressor polytropic efficiency is normally restored to a value slightly lower than the original value following this cleaning.

25.3.2 Interstage Seals

The pressure increases across successive stages of the compressor. Seals are required around the impeller eye and rotor shaft. These seals prevent backflow from the discharge to the suction end of the compressor. The compressor performance is affected by the condition of these seals.

Figure 25.15 illustrates an aluminum labyrinth. This is the simplest and most economical shaft seal. It is commonly used between the stages of the compressor. This seal consists of a series of thin strips or fins. These strips are normally mounted in the diaphragm and remain stationary. The tip of the fins is maintained within a close clearance from the rotor.

The labyrinth seal acts as a series of orifices. There is a pressure drop across each orifice. The energy lost due to this pressure drop across each orifice is converted into heat. This energy conversion into heat is caused by the turbulence that occurs between any two orifices. The gas flow can be reduced by minimizing the size of the openings. Figure 25.16 illustrates a labyrinth seal clogged with dirt. The leakage flow across this seal has increased. This is due to the decrease in turbulence between the orifices. Figure 25.17 illustrates a worn labyrinth seal. The leakage flow across this seal has increased due to the following:

- Increase in the seal clearance
- Reduction in the turbulence between the orifices of the seal

The compressor polytropic efficiency will drop when the labyrinth seal becomes clogged or worn. Thus, these seals should be replaced at the earliest opportunity to restore the compressor polytropic efficiency.

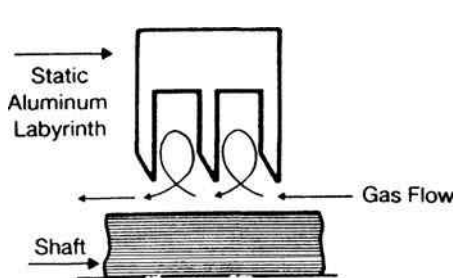


FIGURE 25.15 Aluminum labyrinth in new condition. Tight clearance and flow turbulence create resistance to leakage flow.

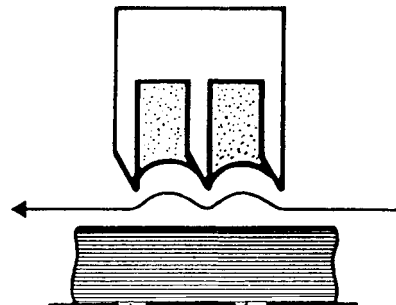
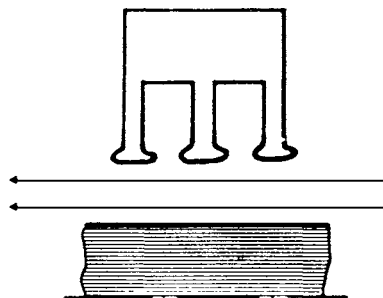


FIGURE 25.16 Fouled labyrinth. Turbulence is reduced and leakage flow is increased.

FIGURE 25.17 Rubbed labyrinth. Clearance is reduced and turbulence reduced, resulting in increased leakage.



Labyrinth seals have been typically made from aluminum. This is due to the following reasons:

- Aluminum is compatible with most gases.
- Aluminum is a ductile material. It prevents rotor damage if rubbing occurs.

Plastics such as Arlon CP (PEEK) and Torlon have been used successfully in applications involving corrosive gases. The seal clearances were not increased when these materials were used. This is because these materials have the same metallurgical properties as aluminum.

Catastrophic failure of the compressor could occur if a hard material is used for the labyrinth seals. This material includes stainless steel and cast iron. Aluminum labyrinth seals were replaced in one application with cast iron seals. The clearances of the seals were not increased. The rotor rubbed against the cast iron labyrinth seals when it passed through the critical speeds. A dry whirl occurred. The rotor vibrations were severe. The retainer belts of the bearings backed out due to this vibration. The bearing housing dropped off the compressor case. The damage was extensive in this case.

The compressor polytropic efficiency can drop by more than 7% due to wiped interstage seals. This result has been confirmed by calculation and field performance data. The following three operating modes enhance the damage to the labyrinth seals:

- Compressor surge
- Operating at the critical speeds
- Liquid ingestion into the compressor

The compressor may trip in some applications on high vibrations when it passes through the first critical speed. This is due to either of the following two conditions:

- High rotor unbalance
- Slow opening anti-surge valve

These high vibrations will increase the seal clearances. Compressor manufacturers have adopted several improvements to reduce the decrease in compressor polytropic efficiency caused by degradation of interstage seals. These improvements include the

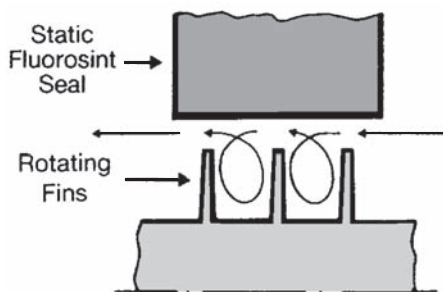


FIGURE 25.18 Abradable seal. Tight clearance and turbulence create resistance to leakage flow.

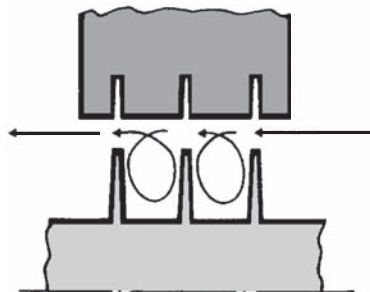


FIGURE 25.19 Rubbed abradable seal. Tight effective running clearance is unaltered and turbulence continues to create resistance to leakage flow.

use of abradable seals (Figs. 25.18 and 25.19) in the impeller eye and shaft seal area. The following are the advantages of abradable seals:

- Tighter clearances between the stationary seal and the rotating fins.
- The turbulence developed between the rotating fins creates a resistance to the leakage flow.
- The effect on the compressor polytropic efficiency is minimal following a seal rub.

The compressor polytropic efficiency is increased due to the usage of abradable seals. This is caused by the reduction in seal clearances. The recirculation flow through the impellers is reduced due to the reduction in seal clearances. The compressor polytropic efficiency is increased when the following seals are replaced with abradable design:

- Impeller eye seals
- Interstage shaft seals
- Balance piston seals

The impeller thrust varies with the seal clearance (Fig. 25.20). Thus, abradable seals control the impeller thrust because they reduce the seal clearance.

The installation of the labyrinth seal fins on the rotor prevents the buildup of deposits between them. This is due to the centrifugal force. Thus, this feature prevents the reduction in the compressor polytropic efficiency associated with the buildup of deposits between the fins. Conventional labyrinth seals have the fins mounted on the stationary part of the seal. This design results in a decrease in the compressor polytropic efficiency due to buildup of deposits between the fins (Fig. 25.16).

The rubbing of the aluminum fins of a conventional labyrinth seal increases the radial clearance of the seal. This is because the fins deform due to the rubbing (Fig. 25.17). They take the shape of a “mushroom.” This increase in the radial

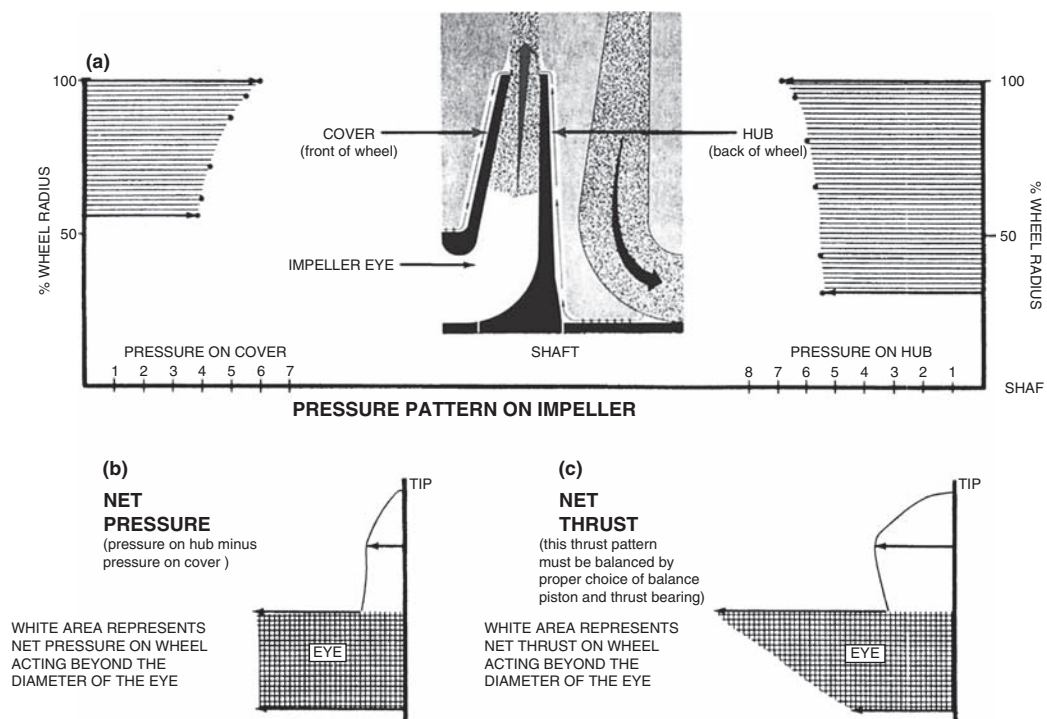


FIGURE 25.20 (a) Diagram showing the pressure pattern on the impeller. (b) Net pressure. (c) Net thrust.

clearance of the seal increases the leakage across the seal. It also results in the following two changes:

1. A decrease in the compressor polytropic efficiency
2. Variations of the compressor thrust loading

These problems are avoided when abraded seals are used. This is because the rotating fins will not be damaged due to rubbing with the stationary part of the seal. Thus, the clearances of the seal will not be affected by this rubbing of the rotating fins with the stationary part of the seal (Figs. 25.18 and 25.19). The variations in compressor polytropic efficiency and thrust loading will not occur in this case.

The increase in the compressor polytropic efficiency due to using abradable seals depends on many factors. These factors include the compressor size. The compressor flow rate increases with the square of the impeller diameter. However, the seal clearance increases almost linearly with the impeller diameter. The seal clearance depends also on the following two factors:

1. Bearing clearances
2. Manufacturing tolerances

Thus, the compressor leakages become a smaller portion of the total flow as its size increases. Therefore, the effect of reducing these leakages on the compressor polytropic

efficiency diminishes as the compressor size increases. In fact, abradable seals provide the most benefit for smaller, higher-pressure compressors.

25.3.3 Balance Piston Seal

Figure 25.21 illustrates a balance piston (or a center seal). Its purpose is to compensate for the axial thrust imposed on the rotor. This thrust is created by the pressure difference across the compressor. The balance piston employs the pressure difference across the compressor to balance the axial thrust. This provides the following benefits:

- The selection of a smaller thrust bearing
- Reduction in the friction losses

The balance piston employs a labyrinth seal. There is leakage across this seal. This leakage is sent back to the compressor intake. Thus, there is a differential pressure across the balance piston. This leakage is sent to other section of the compressor in some applications. This is done to increase the compressor polytropic efficiency. The balance piston leakage is sent to atmosphere in air compressors.

The integrity of the balance piston seal is critical. This is because this seal is sealing the gas at the discharge pressure of the compressor. Any damage to this seal will increase the following parameters:

- Leakage rates
- Thrust loads
- Frictional losses
- Compressor power requirement

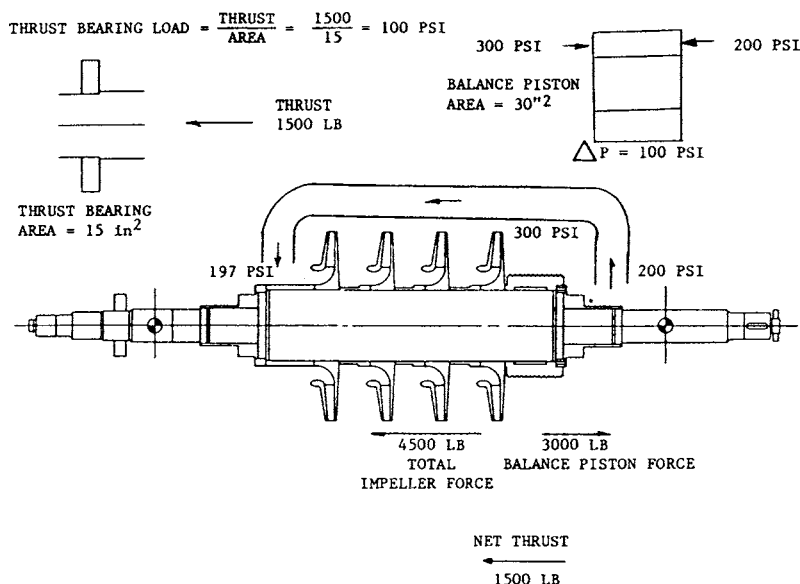


FIGURE 25.21 Schematic of compressor thrust. Pressure drop in the balance line is normally 6.9 to 20.6 kPa (1 – 3 psi).

25.3.4 Impeller Thrust

The differential pressure across the impeller generates the impeller axial thrust. This thrust is determined by the product of the differential pressure across the rotor and the impeller area. The total thrust acting on the rotor is the sum of the individual thrusts acting on each impeller.

25.4 Performance Characteristics

Figure 25.22 illustrates the characteristics of a centrifugal compressor. These characteristics are determined by the geometry of the impeller and diffuser. The impeller imparts the kinetic energy on the gas by centrifugal forces. The diffuser converts this increase in gas velocity into an increase in pressure.

Figure 25.23 illustrates the head curve for a compressor stage. The following are the three important aspects of this curve:

- Slope of the curve
- Stonewall (or choke)
- Surge

FIGURE 25.22 Velocity/pressure development for a typical radial inlet impeller.

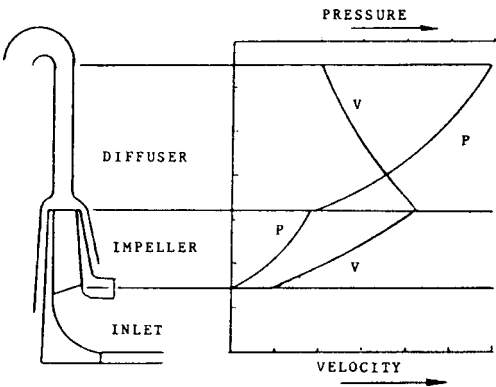


FIGURE 25.23 Head curve for a compressor stage.

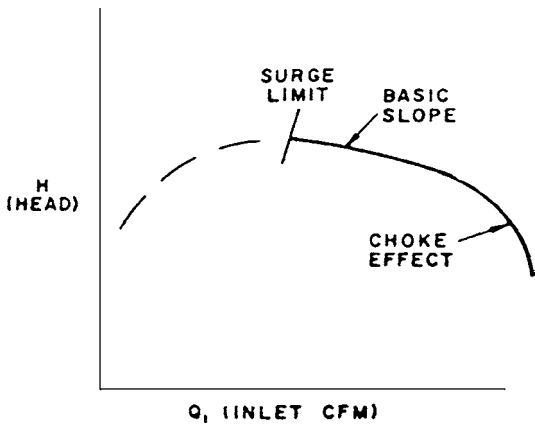
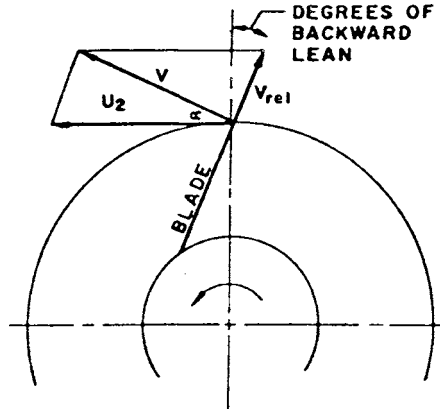


FIGURE 25.24 Vector diagram of the gas velocity relative to the impeller blade. The slope of the characteristic curve is strongly influenced by this relationship.



25.4.1 Slope of the Centrifugal Compressor Head Curve

Figure 25.24 illustrates the vector diagram of the gas velocity relative to the impeller blade. V_{rel} represents the gas velocity relative to the impeller blade. U_2 represents the absolute speed of the blade tip. V represents the absolute velocity of the gas. V is given by the following equation:

$$\vec{V} = \vec{U}_2 + \vec{V}_{rel}$$

Figure 25.25 illustrates the radial (V_R) and tangential (V_T) components of the absolute gas velocity (V). The head output for a radial inlet impeller (shown in Fig. 25.22) is proportional to the product of U_2 and V_T . V_{rel} decreases with the flow in a typical backward-leaning bladed impeller. This causes an increase in V_T . Thus, the head output will increase. This is because it is proportional to the product of V_T and U_2 (U_2 is a constant. It is given by this equation: $U_2 = r \cdot \omega = \text{constant}$, where r is the radius of the impeller and ω is the rotational speed of the impeller). Figure 25.26 illustrates the effects of a change in the flow rate. The following changes occur when the gas velocity, $V_{\text{gas velocity}}$, increases (i.e., the flow through the impeller increases):

- V_{rel} increases
- V_T decreases
- Head output decreases

FIGURE 25.25 Vector diagram of the gas velocity shown in Fig. 25.24 in radial and tangential components.

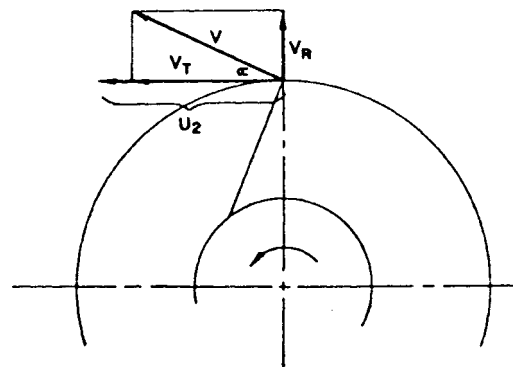
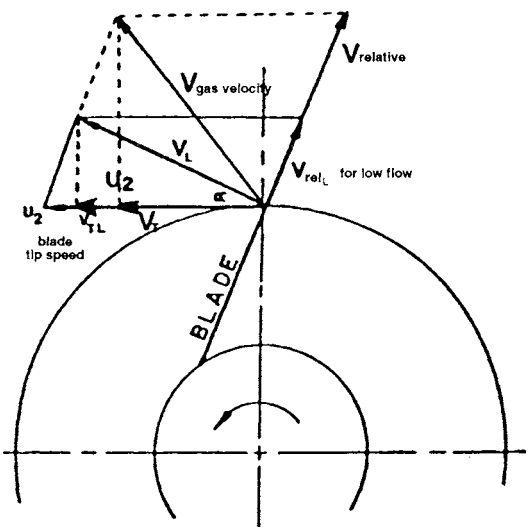


FIGURE 25.26 The effect of a change in flow rate on the vector diagram at the impeller O.D. is shown. Note that V_T decreases as flow increases (V_{rel} increases) for a backward-leaning impeller blade. This gives the backward-leaning impeller the characteristic negative-sloping head curve shown in Fig. 25.23.



This decrease in the head with increasing flow generates the basic slope of the centrifugal compressor performance curve (Fig. 25.23).

Figure 25.27 illustrates the characteristic curves for the following three basic configurations of blade profiles:

1. Forward leaning
2. Radial
3. Backward leaning

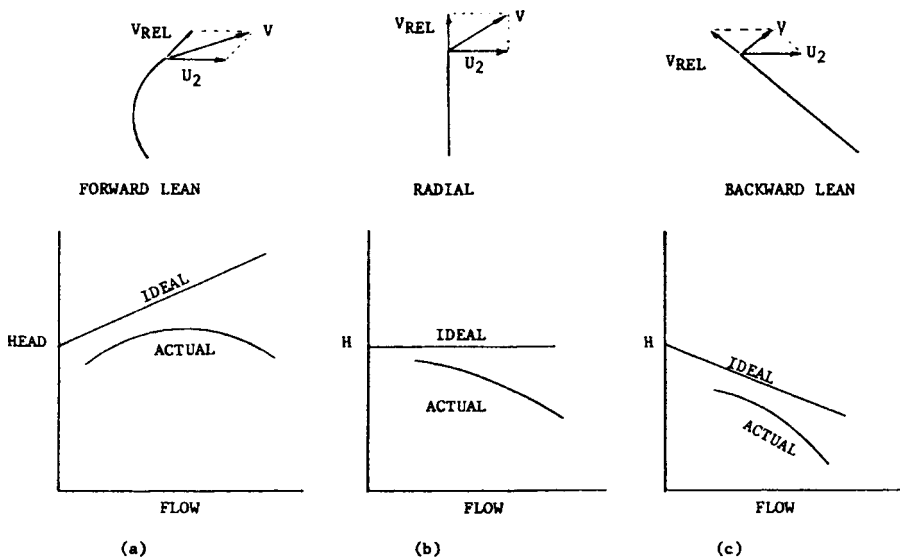


FIGURE 25.27 Three basic head curve shapes for centrifugal compressors.

The forward leaning blade profile provides the following:

- Positive sloping head curve (i.e., the head increases with the flow)
- The maximum head output

This is due to the increase in V_t with the flow. The radial blade profile has a theoretical constant (flat) head curve. This is because V_t remains constant when the flow changes. However, the backward-leaning blade profile has the highest overall stage efficiency (Fig. 25.28). The forward-leaning blade profile has the lowest overall stage efficiency. Thus, most modern centrifugal compressors use backward-leaning bladed impellers. The compressor polytropic efficiency increases with the increase in the backward-lean profile. However, the head drops at a higher rate with the increase in the angle of the backward lean profile (Figs. 25.27c and 25.28). The designer selects the blade angle and tip width to meet the required head and efficiency characteristics of the application.

25.4.2 Stonewall

Figure 25.29 illustrates the stonewall phenomenon (also known as choke). This condition results in a rapid drop in head as the flow is increased. This phenomenon occurs because the velocity of the gas approaches the speed of sound in the gas (Mach number approaches 1.0). The Mach number is given by the following equation:

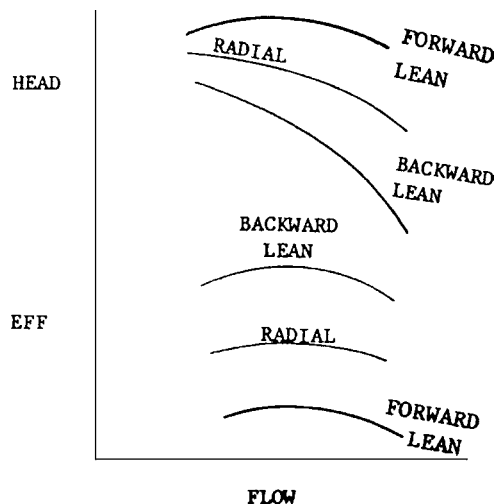
$$M = V/C$$

where M = the Mach number

V = the velocity of the gas

C = the speed of sound in the gas

FIGURE 25.28 The effect of impeller blade angle on head.



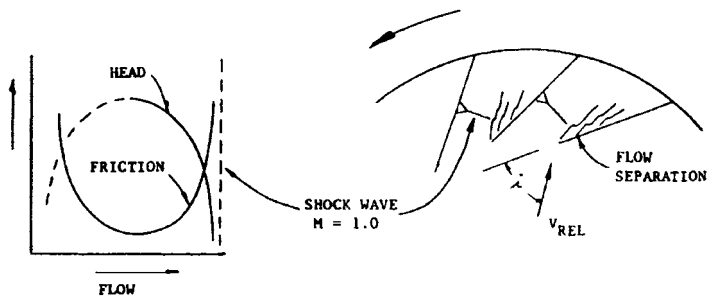


FIGURE 25.29 Stonewall. Flow is limited in impeller throat due to flow separation and developing shock wave.

Compressor damage can occur if it is operated near the stonewall condition (at a very high flow rate). The impeller inlet geometry controls the stonewall effect of a centrifugal compressor stage having a vaneless diffuser.

Figure 25.30 illustrates the following velocities:

- U_1 : represents the tangential velocity at the leading edge of the blades.
- V : represents the absolute velocity of the inlet gas. This gas has completed a 90° turn. It is now moving radially (assuming that the compressor does not have prewhirl vanes). The name radial inlet is derived from the radial velocity of the gas at the inlet of the impeller.
- V_{rel} : represents the gas velocity relative to the blade. The relationship between these parameters is given by the following equation:

$$\vec{V} = \vec{U}_2 + \vec{V}_{rel}$$

V_{rel} is parallel to the blade angles when the compressor operates at the design flow. The magnitude of V increases as the flow increases beyond the design flow of the compressor. This will result in an increase in the magnitude of V_{rel} . The change in the magnitude of V will change the angle of impingement between V_{rel} and the blade. V_{rel} impinges now on the blade at a negative angle. This condition is known as negative

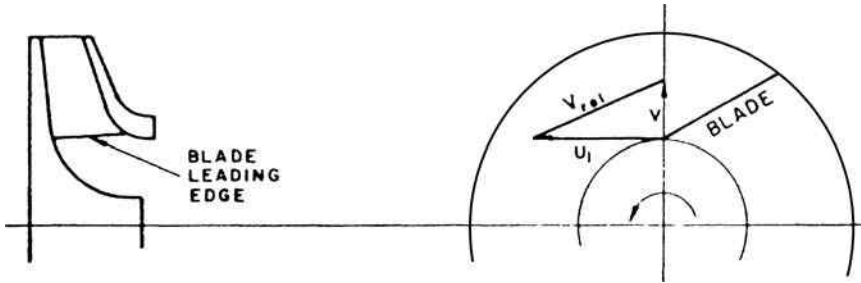


FIGURE 25.30 Flow vectors for impeller design condition.

angle of attack. The stonewall phenomenon is partially caused by high negative angles of attack. This is because of the following:

- Boundary layer separation (see Appendix for details)
- Reduction in the effective area in the blade pack

This reduction in the effective area and the high value of V_{rel} results in the following:

- Flow conditions at Mach 1
- Corresponding shock wave (Fig. 25.29)

25.4.3 Surge

The surge flow occurs at the peak head of the compressor (Figs. 25.23 and 25.29). The surge phenomenon is very damaging to the compressor. It must be avoided. Flow reversal occurs during the surge process. The damage caused by the surge process increases with the increase in compressor discharge pressure.

The magnitude of V_{rel} varies proportionally to the flow. Thus, the magnitude of V_{rel} will drop when the flow is reduced at constant speed. This results in the following changes:

- A decrease in the flow angle α (i.e., the angle between $V_{gas\ velocity}$ and V_T). This change is illustrated in Figs. 25.26 and 25.31.
- An increase in the incidence angle (i.e., the angle between $V_{gas\ velocity}$ and the blade). This increase is illustrated in Fig. 25.31. The flow path of a given gas particle from the tip of the impeller to the outside diameter of the diffuser increases when the flow angle, α , decreases (Fig. 25.31). This will increase the frictional losses that the flow experiences in the diffuser. These frictional losses continue to increase as the flow and flow angle (α) decrease. There is a flow angle, α , at which the frictional losses increase at a higher rate than the head.

FIGURE 25.31 Flow through a diffuser.

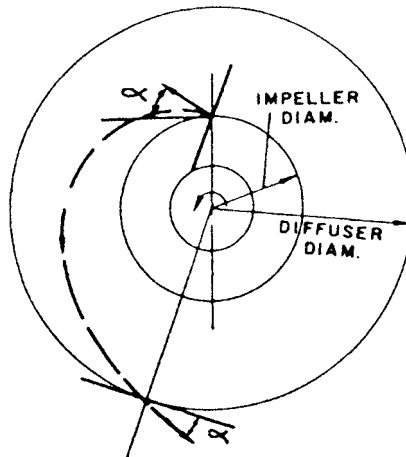
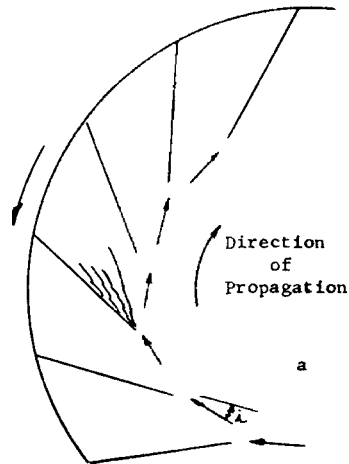


FIGURE 25.32 Rotating stall.



The poor angle of incidence, i , (Fig. 25.32) contributes to the high losses associated with low flow. This is because it can result in flow separation at the low-pressure side of the blade leading edge. This flow separation normally starts at one or more blades simultaneously. It shifts continuously around the impeller blades. This phenomenon occurs at relatively low speeds. The full surge normally occurs just after this phenomenon. The compressor experiences the full surge suddenly at higher speeds. There is a sudden change from stable operation to flow separation on all the blades and a complete flow reversal at these speeds.

The slope of the head-flow curve in the low-flow region (the surge zone) is positive. This is due to the following:

- Flow separation
- Higher frictional losses

The flow is unstable in the surge zone. This is because any decrease in flow in this zone will result in a lower pressure at the discharge of the compressor. However, the area downstream of the compressor is still at the previous pressure. This pressure is higher than the new pressure developed by the compressor. Thus, a flow reversal will occur at this stage. The designer can vary the initiation of the surge zone. This is done by adjusting the diffuser area to increase V_{rel} and the flow angle α . However, the increase in V_{rel} generates higher frictional losses. Thus, the designer must balance between the desired surge zone and the optimum-stage efficiency during the design process.

The point of initiation of the surge zone can be changed. This is done by adding vanes in the diffuser (Fig. 25.33). These vanes shorten the flow path of the gas through the diffuser. This reduces the frictional losses. The reduction in frictional losses results in the following:

- Increase in the head developed by the compressor
- Increase in the compressor polytropic efficiency

However, the operating range of the compressor is reduced. The incidence angle to the vanes, i , changes rapidly during off-design operation. Flow separation occurs also in this condition. This also reduces the operating range of the compressor.

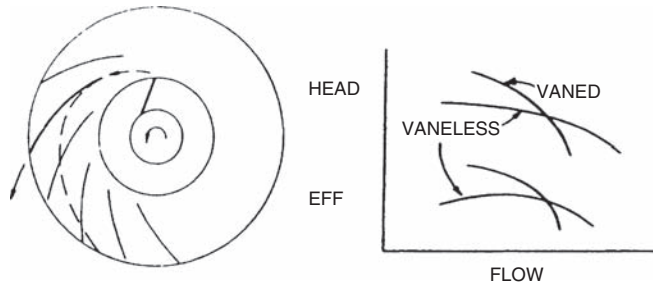


FIGURE 25.33 Diffuser vanes.

Figure 25.34 illustrates a simple compressor system. This system is used to describe the sequence of events that occur during the surge process. It consists of a motor-driven compressor and a large tank. The compressor delivers air to the tank. The entire system is at ambient conditions when the compressor is at the idle state. The pressure in the tank starts to increase from point 1 when the compressor is started. The flow entering the tank drops steadily due to the increase in resistance. Point 2 is eventually reached. The pressure at this point causes a high back pressure on the compressor. The flow through the compressor is significantly reduced. Most of the energy supplied by motor is dissipated by frictional losses. It is not used to build the head. These frictional losses are caused by the following:

- Flow separation at the low-pressure side of the blade leading edge. This is caused by poor incidence angle.
- Longer diffuser passage.

These losses continue to increase as the flow decreases. The slope of the head curve will reverse eventually. The head output of the compressor drops when the flow reaches point 3. However, the pressure in the tank is still at point 2. This results in a reverse flow from the tank to the compressor. This reverse flow continues until the pressure in the tank drops below the discharge pressure of the compressor. The flow will reverse direction again. This flow continues to repeat itself until the compressor

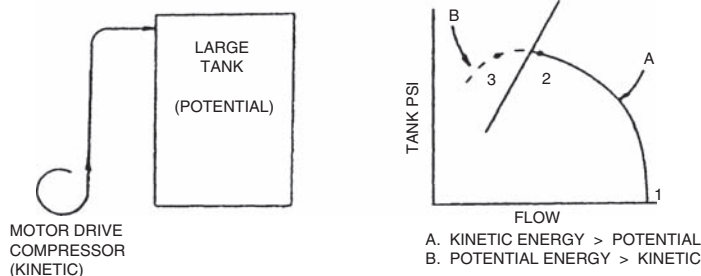


FIGURE 25.34 Surge. Once the pressure in the tank exceeds the capability of the compressor to produce head, reverse flow occurs.

destroys itself. The surge process is very damaging to the compressor. It generates the following in the compressor:

- Severe vibrations
- Significant overheating
- Significant pressure variations

The surge process should be avoided to protect the compressor.

25.4.4
Off-Design Operation

The shape of a compressor performance characteristics curve can change significantly during off-design operation. Figure 25.35 illustrates the variation of the following parameters with changes in the inlet conditions:

- Discharge pressure
- Gas horsepower (GHP) that is, compressor power requirement
- Characteristic head curve

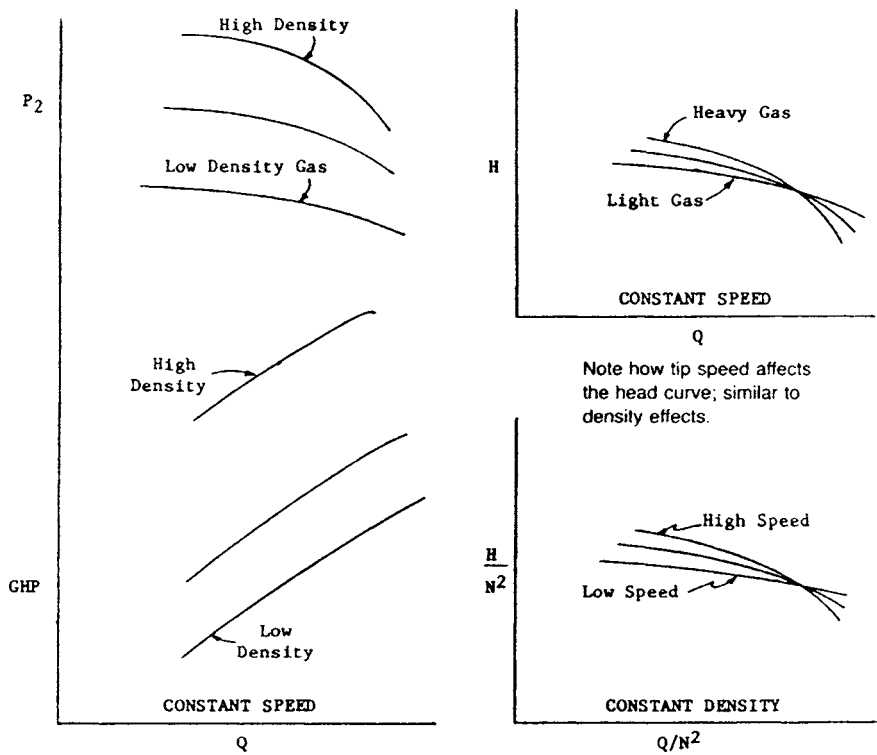


FIGURE 25.35 The effect of varying inlet conditions at constant speed for a single-stage compressor. For multistage compressor, the curve shape and operating range is further compounded by volume ratio effects. See Fig. 25.37.

These graphs indicate that a variable speed drive is required for applications having the following conditions:

1. Constant discharge pressure is required.
2. Inlet pressure, temperature, or molecular weight is changing.

Figure 25.36 illustrates the variation in head curve with the speed. The decrease in head (with increasing flow) is steeper at higher speeds. This is due to the significant increase in turbulence when the flow increases at higher speeds. This increase in turbulence will lead to a higher pressure drop in these cases. Figure 25.37 illustrates the variation of the head and efficiency with changes in the suction density for multistage compressors. These variations are also known as volume ratio effects.

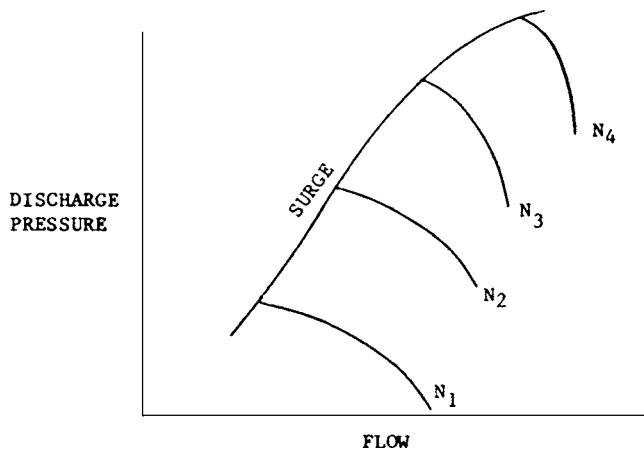


FIGURE 25.36 The effect of speed change on compressor performance curve.

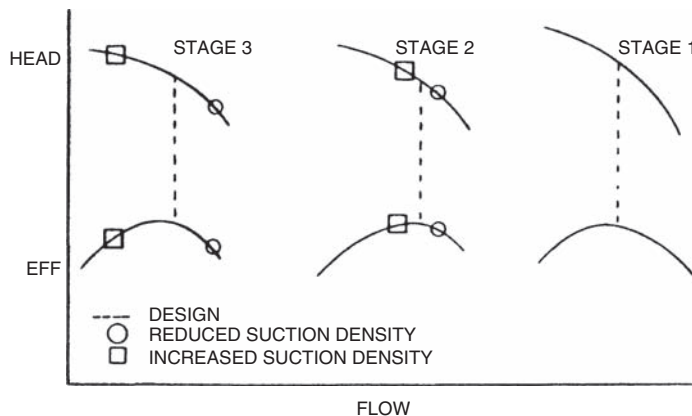


FIGURE 25.37 Volume ratio effects.

The head characteristics depend on the acoustic velocity of the gas. A reference constant is used to determine an “equivalent tip speed.” This reference constant is normally air at 26.67°C (80°F). Air is chosen because it is used in most developmental testing. For example, we know the following:

- The sonic velocity of air at 26.67°C (80°F) is 347.7 m/s (1140 fps)
- The sonic velocity of propylene at -40°C (-40°F) is 225.7 m/s (740 fps)

The equivalent tip speed of a compressor stage that operates at a mechanical tip speed of 250 m/s (819.67 fps) on propylene at -40°C (-40°F) is

$$U_{eq} = 250 \times (347.7 / 225.7) = 385.14 \text{ m/s (1262.74 fps)}$$

Thus, the shape of the stage characteristic head curve at 250 m/s (819.67 fps) on propylene at -40°C (-40°F) is the same as air at 26.67°C (80°F) and 385.14 m/s (1262.74 fps).

This equation is not accurate. This is because there is also the effect of the impeller tip volume ratio. This effect is based on the gas density. It causes a variation in the head and compressor power requirement. The head curves obtained from tests on air and propylene under the same condition will be slightly different than the curves obtained using the equivalent tip speed equation. However, for all practical purposes, the results obtained from the equivalent tip speed equation are considered to be the same as the ones obtained experimentally. The following parameters will change with variation in the gas density:

- Pressure increase across each stage of the compressor
- Volume ratio

Thus, the volumetric flow rate entering a stage will be different than the one entering the previous stage. This effect will be compounded on the following stages. This will result in premature choke and surge.

25.5 Rotor Dynamics

The rotor dynamics of any rotating machinery plays a major role in determining its long-term reliability. Rotor dynamics has many facets. However good balancing is the primary concern of rotor dynamics.

25.6 Rotor Balancing

The purpose of rotor balancing is to line up the mass centerline of the rotating parts and the centerline of the journals. It is done by improving the mass distribution of the rotor and its components. This will reduce the unbalanced forces in the rotor. Balancing is a process that involves adding or removing weights to obtain even distribution of forces around the rotor.

The unbalance should be corrected in the same axial planes along the rotor in which the unbalance has occurred. Correction of unbalance in different axial planes along the

rotor may generate vibration at speeds different from the speed at which the rotor was originally balanced. Thus, the most desirable balancing in high-speed turbo machinery is at the design operating speed.

Rotors are divided into the following categories:

1. *Stiff shaft rotors*. These rotors operate at speeds significantly lower than the first lateral critical speed. These units can be balanced at low speeds. The balancing should be done in two correction planes. This balancing will remain effective at the design operating speed of the rotor.
2. *Quasi-flexible rotors*. The operational speed of these rotors meets the following criteria:
 - a. It is above the first lateral critical speed.
 - b. It is below the higher lateral critical speeds.

The following features should be taken into consideration during the balancing process of these rotors:

1. Model shapes or model components of unbalance (Fig. 25.38)
2. Static and couple unbalance

These rotors can be balanced at low speeds. This is because the balancing techniques can achieve the following objectives:

1. Correct the static and couple unbalance
2. Reduce the residual modal unbalance

The balancing at low speeds of these rotors is sufficient to achieve adequate balancing at the design operating speed. This category includes most multistage compressor rotors.

3. *Very flexible rotors*. These rotors operate at a higher speed than two or more major lateral critical speeds. Several changes in their modal shape occur as the speed is increased to the operating range. This is due to their flexibility. High-speed balancing is required for these rotors. It involves making necessary correction in the weight distribution in numerous balance planes.



FIGURE 25.38 Rotor lateral critical speed mode shapes: (a) first critical mode shape, (b) second critical mode shape.

25.7 Surge Prevention Systems

The surge prevention systems are known also as anti-surge control system. These systems maintain a minimum volumetric flow rate through the compressor. This is done to prevent surge from occurring. The minimum volumetric flow rate through the machine is achieved by bleeding flow from the discharge of the compressor. This flow is discharged to either of the following two:

1. Atmosphere
2. Recirculated back to the compressor inlet

This flow must be cooled in the latter case. The temperature of the flow should be reduced to the normal inlet temperature of the compressor. Most applications require a simple surge prevention system. This system includes normally a flow element. This device measures the flow rate at the inlet to the compressor. The surge prevention system maintains a minimum flow rate through the compressor. This prevents the surge from occurring. However, some applications involve the following two:

1. Variable speed
2. Variations in the gas inlet conditions to the compressor

These applications require a more sophisticated surge prevention system. These systems prevent surge under all conditions. They achieve this objective by modulating a control valve using signals for the following parameters:

- Pressure
- Temperature
- Speed
- Combination of different parameters

Provisions must be made to prevent surge during startup and trip-out of the machine. The control system must open the anti-surge valve within 3 seconds of a driver trip-out (i.e., the driver trip-out must be interlocked to open the anti-surge valve). The machine should be allowed to coast to a full stop with this valve open. Otherwise, the machine could surge while the speed is dropping. The surge could occur due to a fast reduction in the flow while the pressure is still relatively constant. The flow in centrifugal compressors is proportional to the speed. Thus, the flow will drop quickly when the speed drops. However, the pressure downstream of the compressor is still at the discharge pressure that it had before the trip. This condition could very well result in surge. Figure 25.36 illustrates how a reduction in flow at constant discharge pressure will result in surge. Figure 25.39 illustrates the mechanical damage caused by surge. The surge prevention systems are especially important for applications having the following machines:

- Axial compressors
- High-horsepower centrifugal compressors



FIGURE 25.39 Damage caused by compressor surge.

This is because the damage caused by surge in these units can be extensive. Figure 25.40 illustrates the basic components of a typical anti-surge control system. The following is a description of the function of each component in this system:

1. FE. This is a flow element. It is usually on orifice or a venturi. This component is normally located at the compressor suction. It determines the flow rate by measuring the difference of static pressure across the flow measuring element.

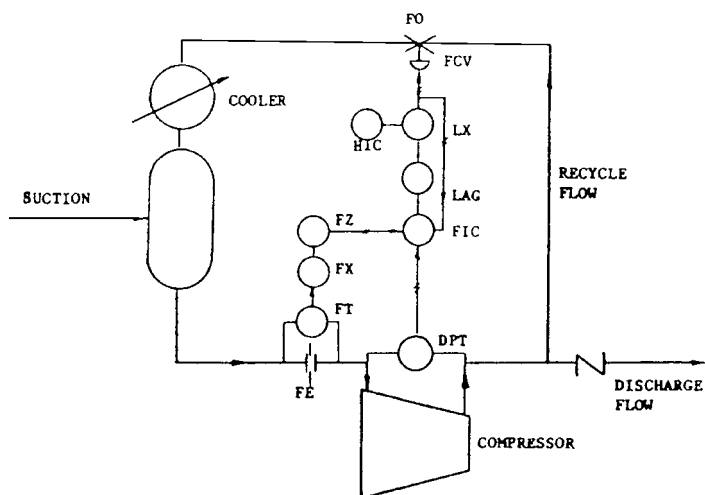


FIGURE 25.40 Typical anti-surge system.

2. FT. This flow transmitter performs the following functions:
 - a. Measures the pressure drop across the flow element.
 - b. Transmits the flow signal. This signal is proportional to the measured differential pressure.
3. FX. This is a ratio station. It receives the signal from the flow transmitter. The ratio station multiplies this signal by a constant. This constant is the slope of the control line.
4. FZ. This is the bias station. It receives the signal from FX. This bias station biases the surge control line.

The surge control line is placed parallel to the compressor surge line (Fig. 25.41). The maximum allowable differential pressure across the compressor for a given flow is given by the following equation:

$$\Delta P = Ch + b$$

where ΔP = the maximum allowable differential pressure across the compressor for a given flow. The anti-surge valve will open when the measured differential pressure across the compressor reaches ΔP

C = the slope of the surge control line (ratio signal)

h = the differential signal across the inlet orifice. This value is measured by FT

b = the bias of the control line

5. FIC. This is the surge controller. It controls the flow. This controller compares the measured differential pressure across the compressor (DPT output) and the ΔP output from FZ. The compressor will be operating in the safe zone (to the right of the surge control line) when the calculated ΔP is greater than the measured ΔP . The surge controller opens the anti-surge valve when the calculated ΔP becomes equal to or less than the measured ΔP . The controller throttles the opening of the anti-surge control valve to maintain the compressor operation on the surge control line. The control system must be able to provide rapid response. This is essential for applications involving rapid changes in the flow.

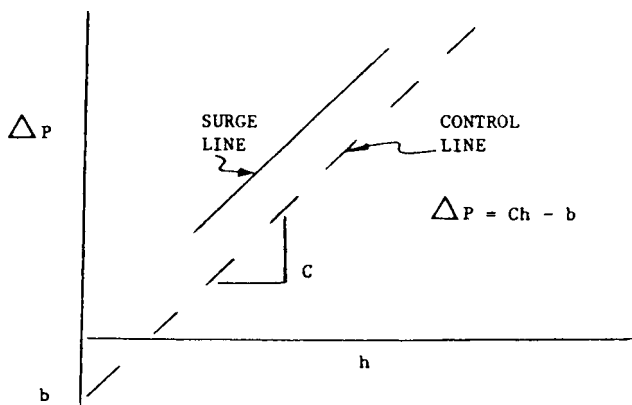


FIGURE 25.41 Surge control line.

6. LAG. This device allows the surge controller to perform the following functions:

- a. Opens the anti-surge (recycle) valve quickly
- b. Closes the anti-surge valve slowly

This feature improves the stability of the control system. It achieves this objective by minimizing hunting between the control system and recycle valve (hunting is defined as the repeated opening and closing of the recycle valve. It can lead to instability and complete destruction of the control system and compressor).

7. LX. This device is a low-signal selector. It receives two inputs. However, it delivers one output. The following are its inputs:

- a. A 100% signal to open the control valve
- b. The surge controller output signal

The output of this device is sent to the recycle valve and back to the surge controller. It provides a feedback signal to the surge controller. This device prevents windup of the controller. Windup of the controller occurs when the control valve becomes unaffected by the signal from the controller. The following example illustrates how windup can occur. A blockage in the recycle line will result in the following condition:

- a. The flow control valve reaching the fully open position.
- b. The signal from the controller has not reached 100%. The controller will in this case continue to increase the output in an attempt to open the flow control valve further. However, since the valve is already fully open, it cannot open further. The controller output will continue to increase until it reaches 100%. The controller output will start to decrease its output when the process requires a reduction in the opening of the flow control valve. However, the flow control valve remains in the fully open position until the controller output reaches the same value that it had when the flow control valve became fully open. There is a range of variation in the controller output when the flow control valve remains unaffected by the controller output. This phenomenon is known as controller windup. It can be eliminated by setting the controller output to 100% when the flow control valve reaches the fully open position. The windup can also be eliminated by preventing the controller output from exceeding the value that has resulted in the full opening of the flow control valve (100% signal valve). This is the purpose of the low-signal selector (LX). It prevents the controller output signal from exceeding the value that has resulted in the full opening of the flow control valve (100% signal valve). Thus, the low-signal selector eliminates the controller windup. The effectiveness of the anti-surge control system will be significantly reduced if the controller windup is allowed to occur.

8. FCV. This is the anti-surge recycle valve. The anti-surge control system opens this valve to prevent surge. The flow is recycled through the valve from the compressor discharge to the compressor inlet. This valve should be sized at 105% of the design flow at the design pressure increase across the compressor.

25.8 Surge Identification

The ideal method to identify the location of aerodynamic instability involves monitoring the dynamic pressure at the following locations:

- Near the impeller inlet
- In the diffuser

Flow instability can occur in either of these locations. Flow separation at the inlet of the impeller blades causes flow instability at the impeller inlet. This instability occurs normally at the peak head on the compressor head curve. The stall occurs in the diffuser section in some applications. This is because the pressure in the diffuser is lower than the compressor discharge pressure. This instability does not normally occur at the peak head of the compressor head curve.

The detection of surge does not require sophisticated instrumentation. The instability can be heard clearly near the compressor in many applications. However, it can be subtle in some applications. Operators should listen very closely and identify the instability in these applications.

The inlet stall condition may not be heard near the compressor discharge. The instability may not also be observed by monitoring the flow and pressure in the control room if the instrumentation has a slow response. However, hard surge can be detected in the control room in these cases.

The surge phenomenon can be very damaging for the compressor. The surge line is established by allowing the machine to experience surge for only one or two surge pulses. The unit can be damaged extensively if the surge phenomenon is allowed to continue. The surge can be prevented by having a quick opening recycle or blowoff valve. This valve must be able to open fully within 1 to 2 seconds. The closure time of this valve must be within 10 to 20 seconds. This will enhance the stability of the anti-surge control system.

25.9 Liquid Entrainment

Liquid entrainment in the gas stream can damage the compressor extensively. The condensation of liquids in the recycle line can create blockage in the line. This will reduce the effectiveness of the anti-surge system. The liquid droplets entrained with the gas cause impingement damage in the first stages of the compressor. These droplets tend to vaporize inside the compressor (due to the increase in temperature). This will increase the density of the gas in the last stages of the compressor. The operating point of the compressor will shift on the head-flow curve. This phenomenon is illustrated in Fig. 25.37. Thus, surge can occur due to liquid entrainment.

The vaporization of the liquid droplets inside the compressor will also increase the velocity of the gas in the last stages of the compressor. This increase in the gas velocity above the design velocity results in mechanical damage in these areas. Thus, liquids must be prevented from entering the compressor during normal operation and upset conditions. The following recommendations should be implemented to prevent compressor damage due to liquid entrainment in the gas:

1. Install a knockout drum at the compressor inlet in applications that may involve liquid entrainment in the gas entering the compressor. This equipment removes the liquid from the gas entering the compressor. It is also known as vapor-liquid separation or flash drum. Refer to Chap. 22 for details.
2. Keep the conditions at the compressor inlet above the liquefaction points of any gas constituent.
3. Heat trace, bleed off, or purge stagnant lines that may experience liquid condensation due to the cooldown of stagnant gas during shutdown periods.
4. The gas flowing in the recycle lines should reenter the main gas stream upstream of the knockout drum.
5. Install drains and level indicators in all the low spots of the piping and vessels located upstream or downstream of the compressor that may have liquid accumulation. This will allow draining of these liquid as required.
6. Ensure that all the liquids that may have condensed upstream or downstream of the compressor during the shutdown period are drained before starting the compressor.

25.10 Instrumentation

Figure 25.42 illustrates the instrumentation required for a compressor system. Two independent instruments are required as a minimum for each location involving measurement of temperature or pressure. The following components should not be installed between the pressure tap points and the compressor flanges:

- Valves
- Strainers
- Silencers
- Other sources of pressure drop

The pressure taps located near a bend should not be in the bend plane. They should be normal to the bend. The hole of the static pressure tap should be between 0.635 cm (0.25 in) and 1.27 cm (0.5 in). It should also be deburred and have a sharp edge (Fig. 25.43).

25.11 Cleaning Centrifugal Compressors

Centrifugal compressors require frequent cleaning. This is due to the buildup of dirt, polymer, or other substances on the internal components of the compressors. The compressor performance will drop significantly if it is not cleaned frequently. The cleaning frequency is determined by the drop in efficiency. The compressor performance is normally restored to “like new” following the cleaning process.

Centrifugal compressors can also be cleaned online. This is done by injecting liquid cleaning agents into the gas at the compressor inlet during normal operation. However, online cleaning has proven to be less effective than offline cleaning. Cleaning the

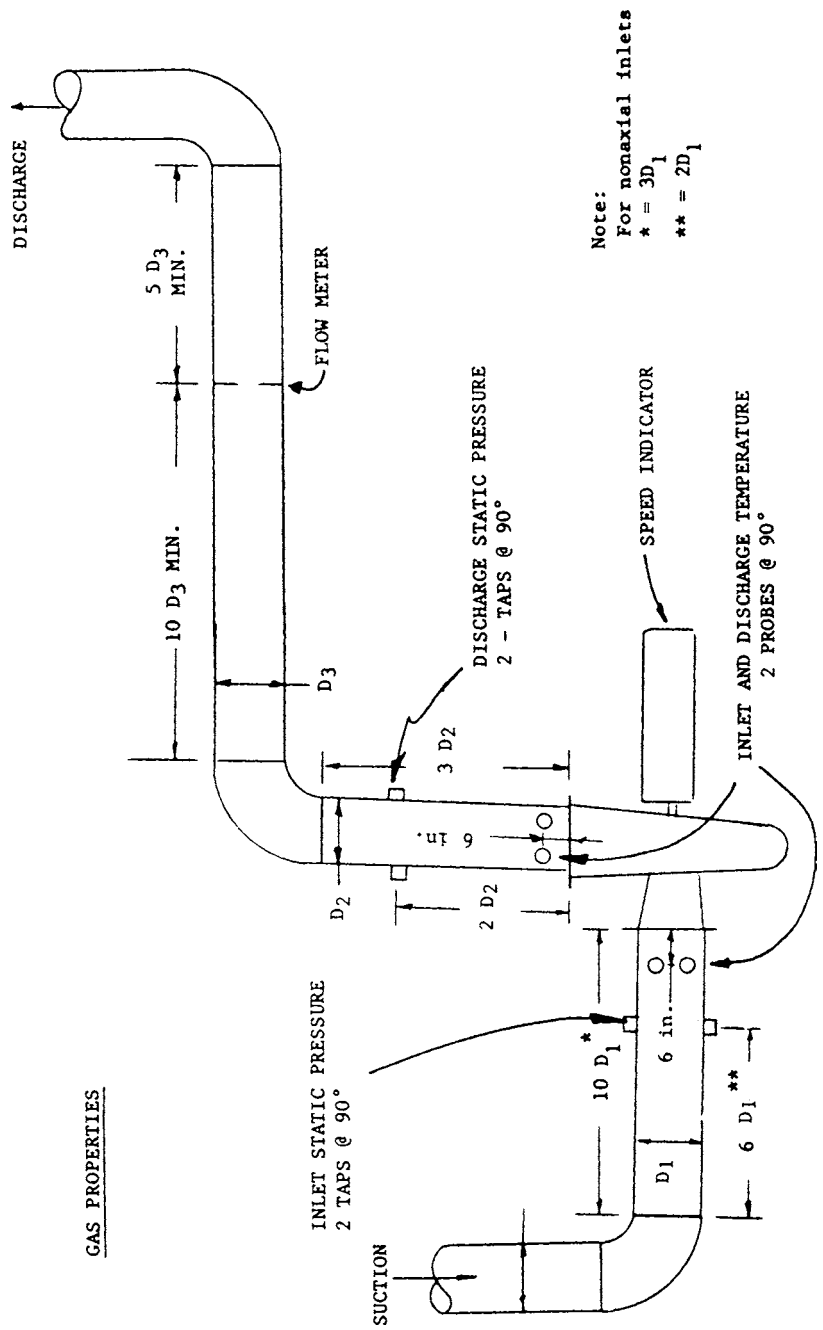


Figure 25.42 Typical performance test setup.

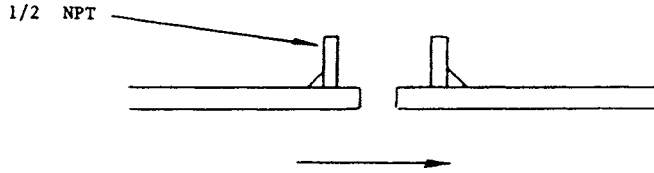


FIGURE 25.43 Static pressure tap. The hole should be $\frac{1}{4}$ to $\frac{1}{2}$ in. in diameter. It should be deburred but have a sharp edge.

compressor online by injecting mild abrasives such as cooked rice or walnut shells is not recommended. This is due to the following reasons:

1. The abrasive material injected can erode the protection coatings placed on the internal components of the compressor. This will enhance the erosion and corrosion rates of these components.
2. The abrasive material injected can penetrate the oil used in the compressor seals in some applications. This will damage these seals and contaminate the oils.

25.12 Bibliography

Hanlon, P. C., *Compressor Handbook*, McGraw-Hill, New York, 2001.

25.13 Appendix: Boundary Layer

25.13.1 Definition

The *boundary layer* is the layer of fluid in the immediate vicinity of a flow boundary that has had its velocity relative to the boundary affected by viscous shear. Boundary layers can be either laminar or turbulent. This depends on the following:

- Length of the boundary layer
- Viscosity of the fluid
- Velocity of the flow near the boundary layer
- Boundary roughness

25.13.2 Description of the Boundary Layer

The flow has zero relative velocity at the boundaries. However, the velocity gradient is steep from the boundary into the flow. The boundary shear forces reduce the velocity of the flow in the boundary layer. The boundary layer is very thin at the upstream end of an object at rest in a uniform flow. However, the thickness of the boundary layer increases with the distance from the upstream point. This is due to the continual action of the boundary shear stress which slows down additional fluid particles.

25.13.3 Separation: Wake

The thickness of the boundary layer drops when the pressure decreases in the downstream direction. However, the thickness of the boundary layer increases rapidly when the pressure increases in the downstream direction (i.e., *adverse* pressure gradient profile). The boundary shear stress and the adverse pressure gradient decrease the momentum in the boundary layer. The flow in the boundary layer stops when the decrease in its momentum reaches a specific value. Figure 25.44a illustrates this phenomenon. It is called *separation*. The boundary streamline separates from the

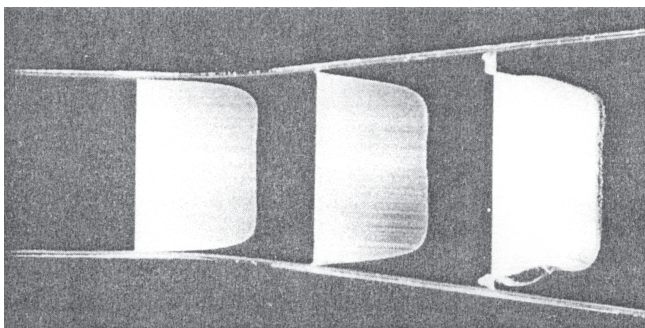
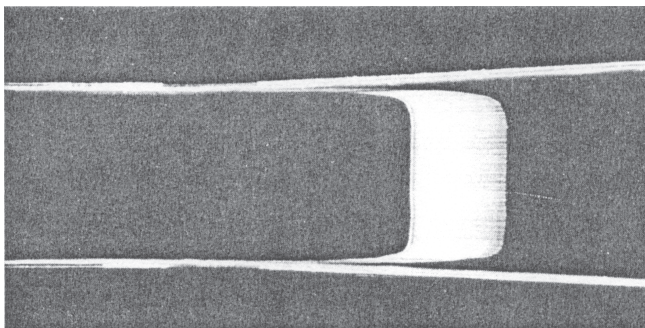
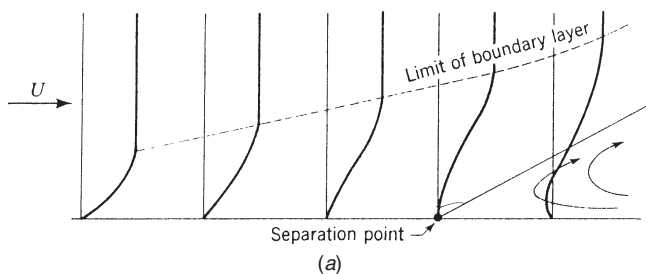


FIGURE 25.44 (a) Effect of adverse pressure gradient on boundary layer. Separation. (b) Boundary layer growth in a small-angle diffuser. (c) Boundary layer separation in a large-angle diffuser. [Source: Parts (b) and (c) from the film "Fundamentals of Boundary Layers," by the National Committee for Fluid Mechanics and the Education Development Center.]

boundary at the separation point. The adverse pressure gradient causes reverse flow near the wall downstream from the separation point. The region downstream from the separation point is called wake. Figure 25.44*b* illustrates the growth of the boundary layer in a small-angle diffuser (very small adverse pressure gradient). Figure 25.44*c* illustrates the boundary layer separation and reverse flow near the boundaries of a large-angle diffuser (high adverse pressure gradient).

25.13.4 Bibliography

Streeter, V. L., *Fluid Mechanics*, 7th ed., McGraw-Hill, New York, 1979.

This page intentionally left blank

CHAPTER 26

Compressor Auxiliaries, Off-Design Performance, Stall, and Surge

26.1 Introduction

Axial and centrifugal compressors are used in gas turbines. Axial compressors are normally used in gas turbines that have a rating higher than 3 MW. Centrifugal compressors are normally used in gas turbines that have a rating lower than 3 MW. Figure 26.1 illustrates a low and a high-pressure axial-flow compressors used in an aeroderivative gas turbine. The following is a description of their auxiliaries, off-design performance, stall, and surge.

26.2 Compressor Auxiliaries

A filter is installed at the inlet of an axial-flow compressor to protect it from particulate damage. A silencer is also installed at the inlet to reduce noise. There is a pressure drop across each of these equipment. Figure 26.2 illustrates the enthalpy-entropy (h - s) diagram across the compressor.

The tip of modern compressor blades operates in the supersonic or transonic (near the speed of sound—Mach 1) flow regime. The front rows of the blades operate in the transonic while the back rows operate in the subsonic flow regime. The temperatures in the back rows of the compressor are much higher than the front rows due to the increase in pressure. The discharge temperature of a modern gas turbine compressor is in the range 300 to 480°C (572–890°F). Figure 26.3 illustrates a typical rotor of an axial-flow compressor.

26.3 Compressor Off-Design Performance

Compressors often operate at a state different from their design state. The deviations from the design state include the following:

- Changes in the ambient conditions
- Part load, or overload operation

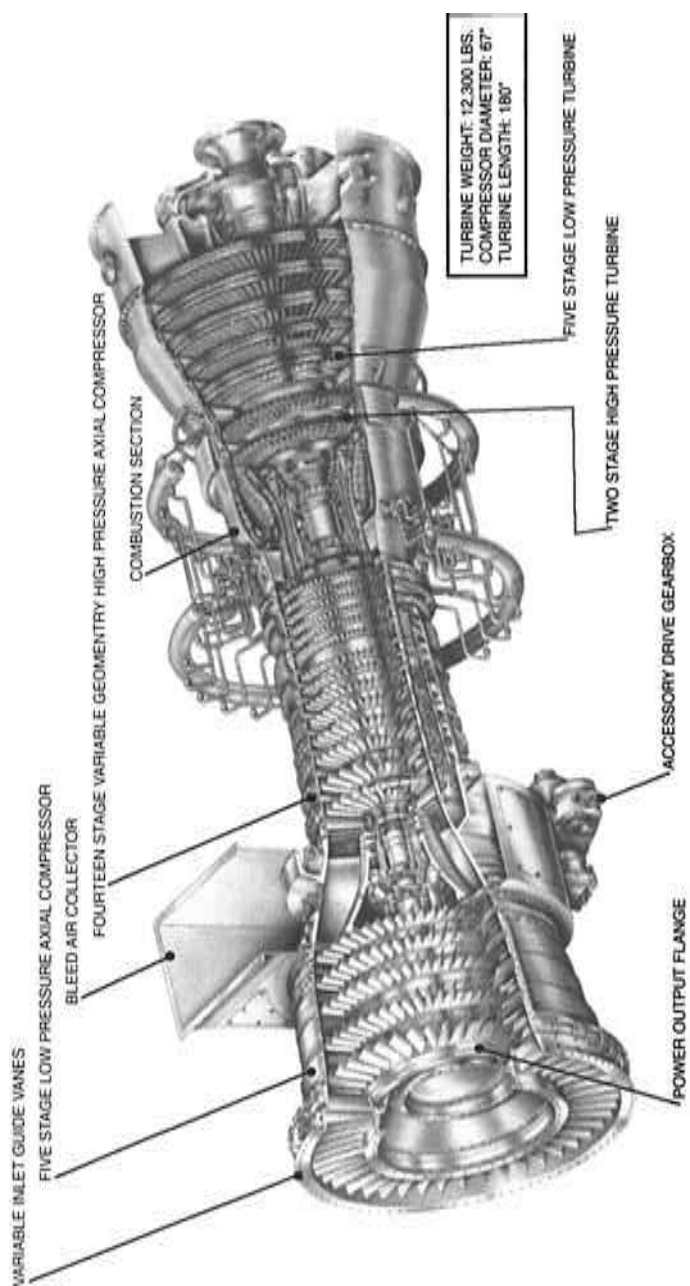


Figure 26.1 The LM600Q aeroderivative gas-turbine engine.

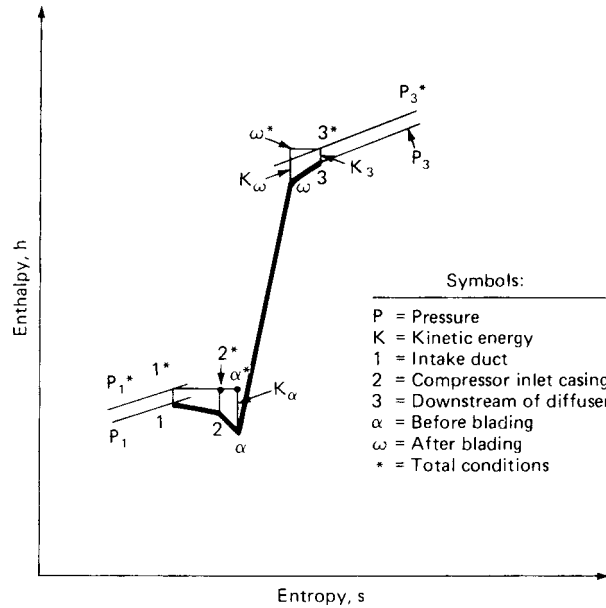


FIGURE 26.2 An h - s diagram for the compressor process.

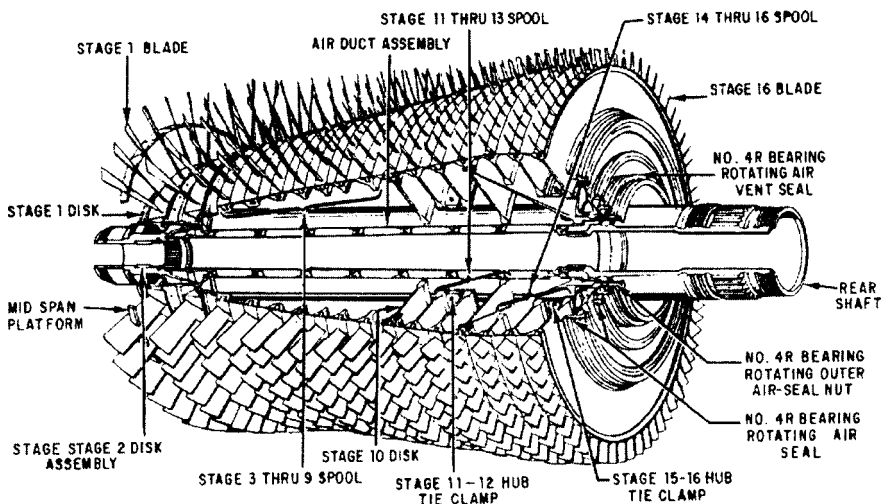


FIGURE 26.3 LM2500 compressor features.

- Transient operation including start-up and shutdown
- Operation anomalies

Compressors must be able to accommodate these situations. They are very sensitive to deviations from design conditions. Careful considerations must be made during the design and operation phase of the machine to prevent failures. Since the pressure increases along the compressor, a slight disturbance can cause a stall in the flow. This could lead to a pressure recovery that cannot be eliminated even if the disturbance is removed. The consequences can be very serious if the compressor operates in any non-stable zone. The compressor blades could be destroyed due to high transient stresses.

Thus, the compressor designer determines the limits of the stable operating zone. Measurements are also taken to confirm the design calculations. Figure 26.4 illustrates the compressor characteristics in the form of pressure ratio (r_c) versus the mass flow rate ratio ($\dot{m}/\dot{m}_0$). The following is a detailed description of the flow mechanisms in the zone close to the stability limit.

26.3.1
Low Rotational Speeds

During start-up, the rotational speeds and pressure ratios are low. Stall could occur on the blade profiles in the front stages of the compressor due to unfavorable flow conditions. The stall phenomenon does not occur in all the blade profiles simultaneously. However, it occurs in the zones that rotate at around one-half of the compressor's speed. This phenomenon is known as *rotating stall*. There are localized reversed flows within the stall zones. This will cause a decrease in the compressor mass flow.

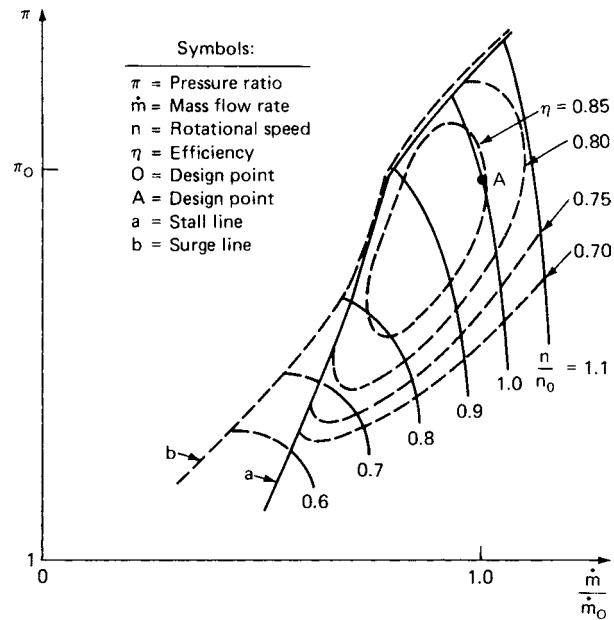


FIGURE 26.4 An r_c - m diagram. (Source: Traupel, W. *Thermische Turbomaschinen*, Vol. 2, 3rd ed. Springer, 1982.)

However, the flow remains constant. These cases are known as *local instabilities*. These stall zones will spread if the pressure ratio increases while the rotational speed of the compressor remains constant. They can eventually block the entire stage cross-section and cause sudden variations in the mass flow between the values that will lead to stall, rotating stall, or even flow reversal (negative flow). This phenomenon is known as *compressor surge*.

26.3.2 High Rotational Speeds

A rotating stall can occur at high rotational speeds of the mass flow drop or the pressure ratio increase. This stall is initiated normally in the last stage of the compressor. It is established along the whole length of the blade leading to a merge between the rotating stall and surge phenomenon as shown in Fig. 26.4.

The compressor should not be operated in the rotating stall or surge zones. This is because the blades could be damaged due to high transient stresses. Thus, the nominal operating pressure is around 15% to 20% lower than the surge line. Deviations from nominal conditions during operation do not normally lead to surge. However, the compressor blades should be cleaned regularly because fouling of the blades lowers the surge line.

The compressor would cross the rotating stall zone during start-up and shutdown if no special precautions were taken. Compressor bleed valves are used to shift the start-up line of the compressor away from the stall line. The bleed valves are open when the pressure is low. They close when the rotational speed reaches 90% of the operational speed due to increased pressure inside the compressor. The variable stator guide vanes achieve the same effect. This method has the advantage of reducing the starting power of the unit. However, it is more expensive.

26.4 Performance Degradation

The performance degradation in a gas turbine can be either recoverable or non-recoverable. The recoverable power losses are mainly caused by compressor fouling. They can be restored by off-line water washing of the compressor. Non-recoverable losses are normally caused by wear in the components. They can be restored by a major disassembly and repair of the components. These are explained below:

- *Recoverable.* Compressor fouling occurs during operation. It results in a loss of power output. In extreme cases, it can cause surge. The compressor can be cleaned while it rotates at low speed by injecting water-containing cleaning solvents. The machine should continue to operate following the injection of the mixture until dry. In most cases, the power lost due to fouling will be regained using this technique. The use of abrasive materials such as crushed nutshells to clean the unit at full speed is not recommended. The reason is that this technique erodes the coating layer placed on the turbine blades. The nutshells can also enter into the instrumentation ports and the hydraulic governing system causing potential impairments. Wet cleaning at full speed is not effective because the drops evaporate in the compressor. Fouling of the turbine occurs when the fuel contains ash. Crude and heavy oils contain a large amount of ash. Some ash contains corrosive components such as alkaline and heavy metals. Magnesium and silicon must be added to the fuel to reduce the rate of corrosion. However,

these additives increase the amount of ash generated. They also buildup a hard coating on the turbine blades. This coating changes the profile of the blades resulting in a reduction in efficiency and flow capacity. Low-speed cleaning is effective in removing this coating if it is water-soluble or hygroscopic.

- *Non-recoverable.* Non-recoverable losses for aeroderivative gas turbines are similar to heavy frame machines. They are around 1 to 2% per 25,000 hour from “new and clean” condition.

26.5 Bibliography

Elliott, T., *Standard Handbook of Powerplant Engineering*, 2nd ed., McGraw-Hill, New York, 1997.

CHAPTER 27

Dynamic Compressors Performance

27.1 Description of a Centrifugal Compressor

Figures 27.1 and 27.2 illustrate the major components of a multistage centrifugal compressor. This machine has the following components:

- A. An outer casing
- B. Diaphragm bundle
- C. Shaft
- D. Impellers
- E. Balance drum
- F. Thrust collar
- G. Rotor hub (it is used to drive the rotor)
- H. Journal bearings
 - I. Thrust bearing (to hold the rotor in position axially)
- L. Labyrinth seals
- M. Oil film end seals

The gas enters the compressor through a suction nozzle. It then moves through an annular chamber known as the inlet volute. The gas then flows uniformly from all directions into the center (Fig. 27.3). A fin is installed at the opposite side of the chamber from the suction nozzle. It prevents the formation of gas vortices. The gas flows into the suction diaphragm. It then enters the eye of the first impeller.

The impellers are made from two discs. They are known as the disc and shroud. These discs are connected by blades. The impellers are shrunk onto the shaft. They are also held by one or two keys. The gas is discharged from the impeller at a higher velocity and pressure. Figure 27.4 illustrates the pressure distribution around the impeller. The disc side of the impeller is exposed to the discharge pressure. A part of the impeller on the other side is also exposed to discharge pressure. However, the part near the eye of the impeller is exposed to the suction pressure. Thus, there is a thrust force that acts on the rotors toward the suction.

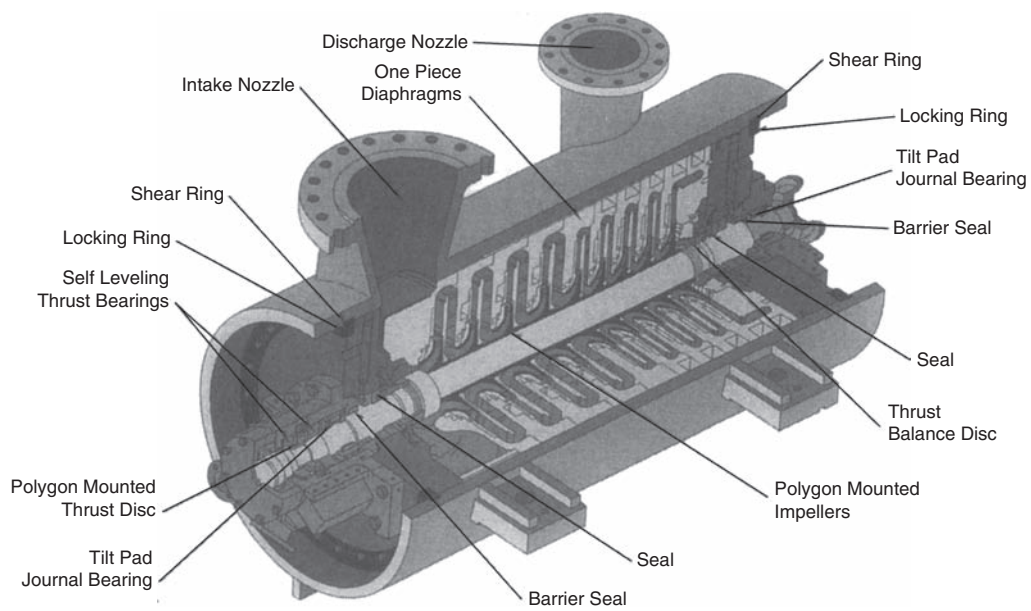


FIGURE 27.1 Major components of multistage centrifugal compressors. (Source: Dresser-Rand Company, Olean, N.Y.)

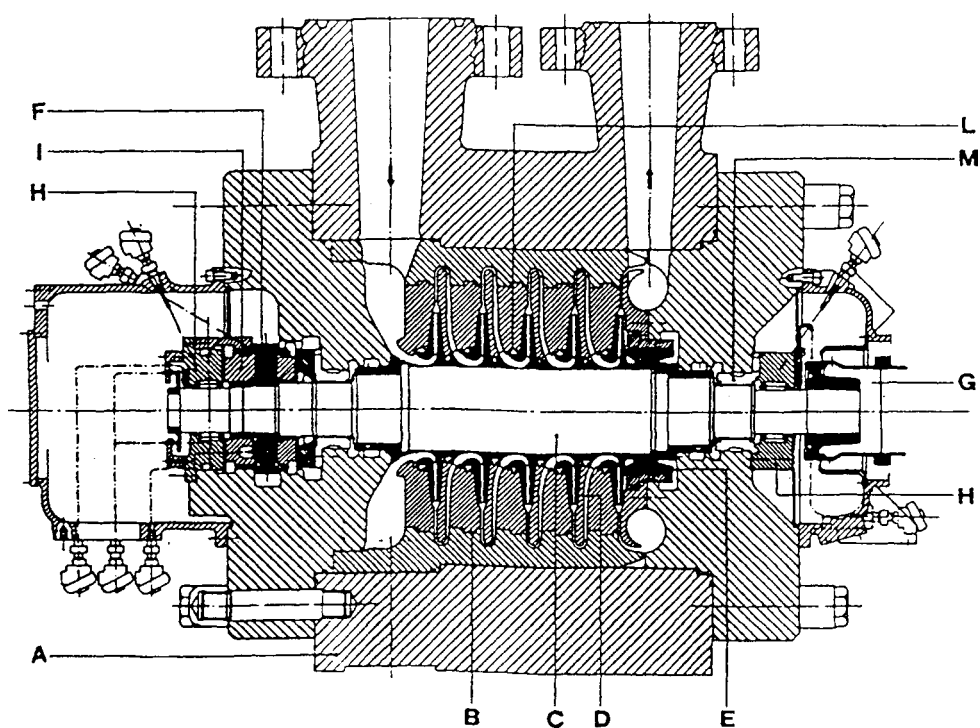


FIGURE 27.2 Sectional view of centrifugal compressor schematic.

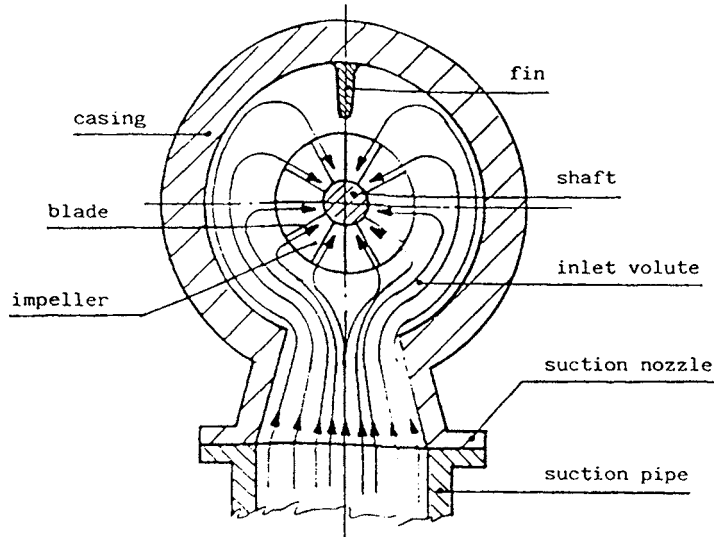


FIGURE 27.3 Qualitative view of the flow in the volute.

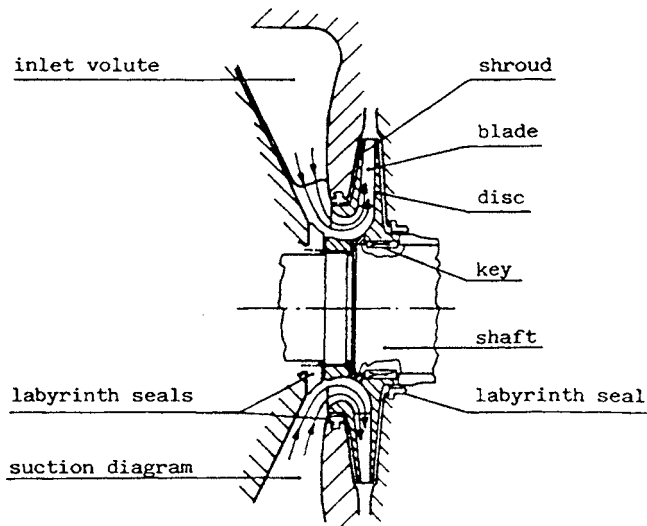


FIGURE 27.4 First-stage sectional view.

The gas flows next through a diffuser. This is a circular chamber of increasing cross sections (Fig. 27.5). The diffuser increases the pressure of the gas by dropping its velocity (first law of thermodynamics—conservation of energy). The gas then enters through the return channel. This component is a circular chamber bounded by two rings. These rings form the intermediate diaphragm. Blades are fitted in this diaphragm. They direct the gas toward the inlet of the next impeller. The blades straighten the spiral gas flow. This is done to obtain a radial outlet and axial inlet to the next impeller. The gas path is identical for each impeller.

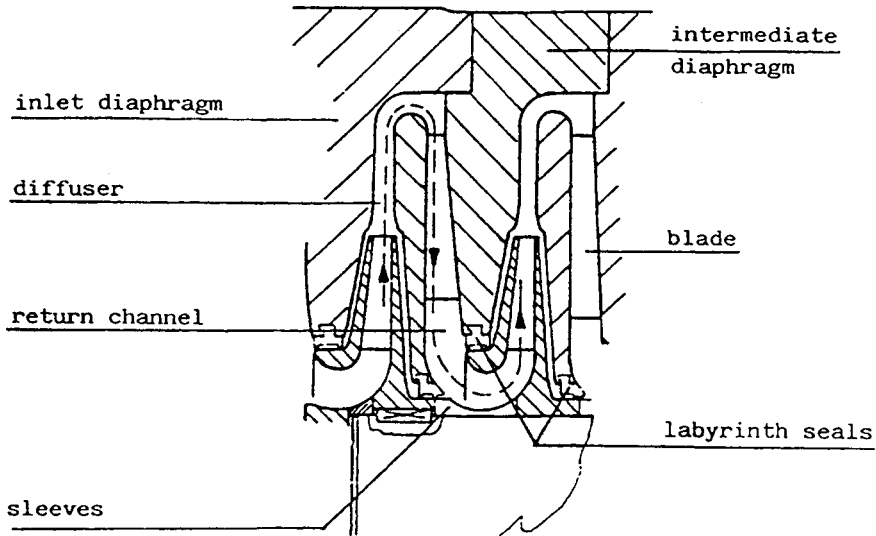
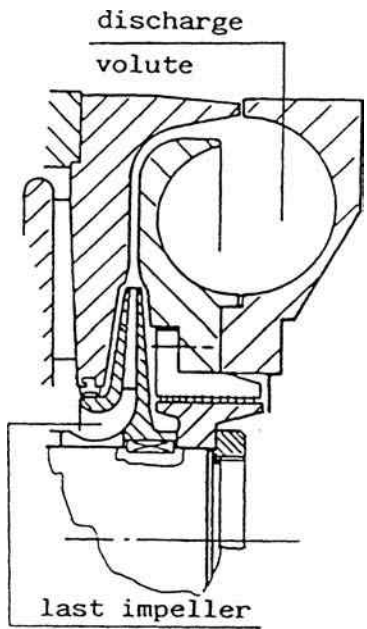


FIGURE 27.5 Labyrinth seals and diaphragms.

Figure 27.5 also illustrates the labyrinth seals. These seals are installed on the diaphragms. They minimize the internal gas leakage between the stages (a stage is the area of compression between two consecutive nozzles) of the compressor. The labyrinth seals consist of rings made of two or more parts. The gas exists from the last impeller of a stage and enters into a diffuser (Fig. 27.6). This diffuser directs the gas to an annular

FIGURE 27.6 Last impeller of a stage.



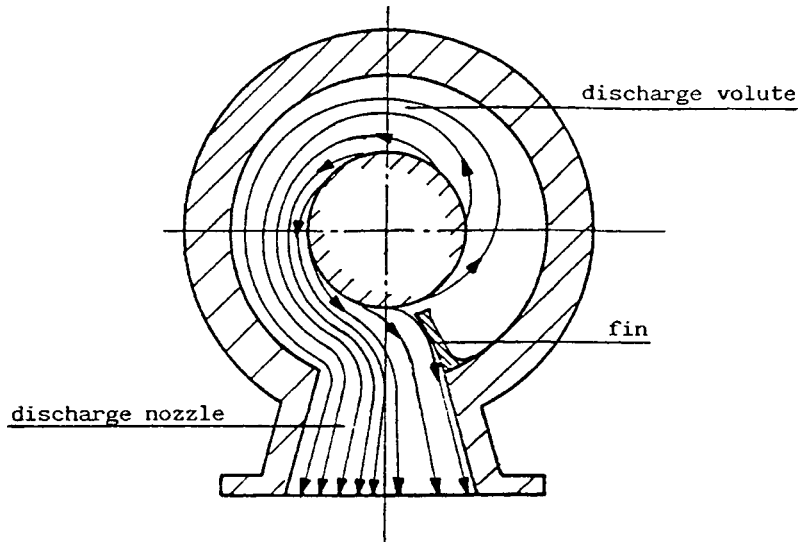


FIGURE 27.7 Discharge volute: qualitative view of the flow.

chamber known as the discharge volute (Fig. 27.7). The discharge volute collects the gas from the diffuser and sends it to the discharge nozzle. There is another fin near the discharge nozzle. It directs the gas to the discharge nozzle.

The balance drum (E) is mounted on the shaft (Fig. 27.2). It is located after the end impeller. The balance drum balances the total axial thrust produced by all the impellers. The discharge pressure of the last impeller acts on one side of the balance drum. The compressor inlet pressure is applied to the other side of the balancing drum through a balancing line (Fig. 27.8). This configuration eliminates the axial thrust produced by the

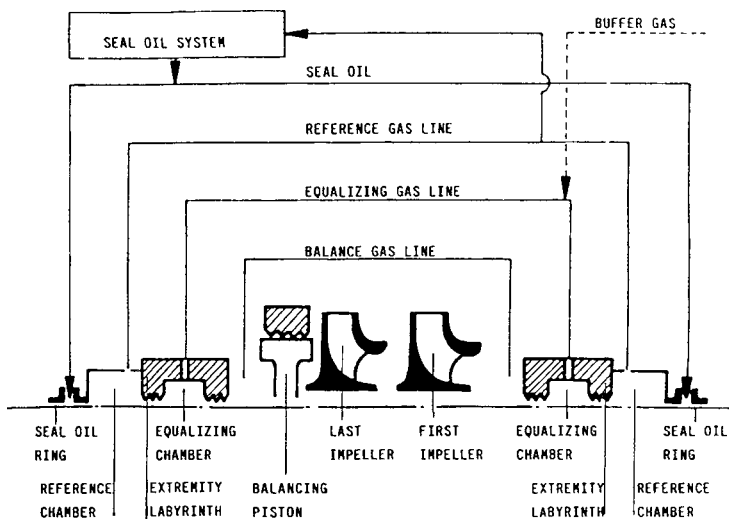


FIGURE 27.8 External connection of the oil system.

impellers. There is also another external connection between the balancing chambers (balancing line, Fig. 27.8) for the shaft-end oil seals. The purpose of this line is to reduce the pressure difference between the two oil seals. The gas chambers are located outside the shaft-end labyrinths. These chambers are also connected to obtain the same reference pressure for the oil seal system (Fig. 27.8). An inert gas is injected into the balancing chamber (buffer gas system) in applications requiring separation of the seal oil and process gas. The pressure of this buffer gas allows it to leak both inward and outward to create the seal.

27.2 Centrifugal Compressor Types

There are different configurations of centrifugal compressors. They are designed for specific applications and pressure ratings. They are classified as follows.

27.2.1 Compressors with Horizontally Split Casings

Horizontally split casings are made of two half casings. These casings are joined along the horizontal centerline. Compressors with horizontally split casings are used for applications having operating pressures below 70 bar. The following components are connected to the lower casing of these compressors:

- Suction and delivery nozzles
- Any side stream nozzles
- Lube oil pipes

This configuration provides access to all internal components by simply raising the upper casing. Centrifugal compressors having horizontally split casing can further be identified as follows:

- *Single-phase multistage compressors* (Fig. 27.9).
- *Multistage two-phase compressors* (Fig. 27.10). The two compression phases are arranged in series. The gas is cooled between the two phases to increase the efficiency of the machine.
- *Multistage three-phase compressors* (Fig. 27.11). These units are used in applications requiring compression of different gas flows to various pressure levels. These gas flows are injected and/or extracted from the machine during operation.
- *Two-phase compressors with a central double-flow rotor* (Fig. 27.12). The gas enters these units from the ends of the machine. It is discharged from the delivery nozzle. This component is positioned in the center of the casing. This design eliminates the axial thrust produced by the impellers.

27.2.2 Centrifugal Compressors with Vertically Split Casings

Vertically split compressor casings are made from a cylinder. Two covers are used to close the cylinder at the ends. This design is normally referred to as “barrel.” These compressors are used in high-pressure application (up to 70 MPa). Figure 27.13 illustrates a barrel-type compressor having a single compression phase. Figure 27.14 illustrates a barrel-type compressor having two compression phases in parallel within the casing.

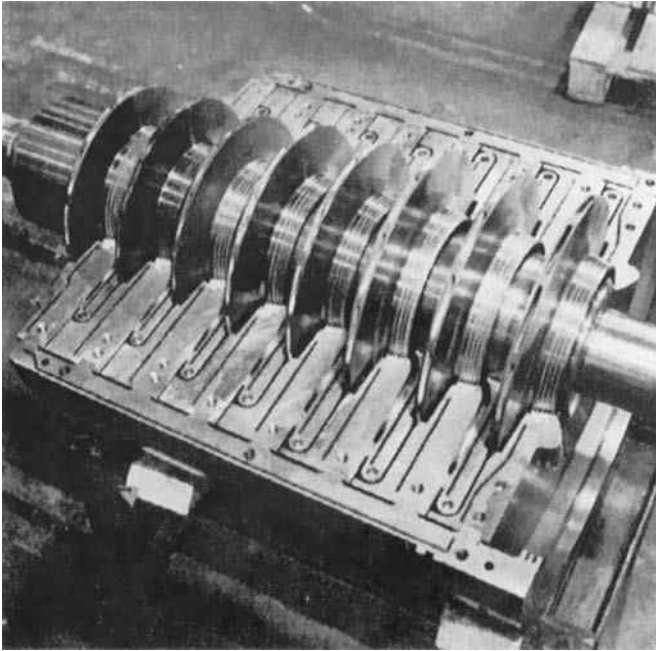


FIGURE 27.9 Horizontally split casing.

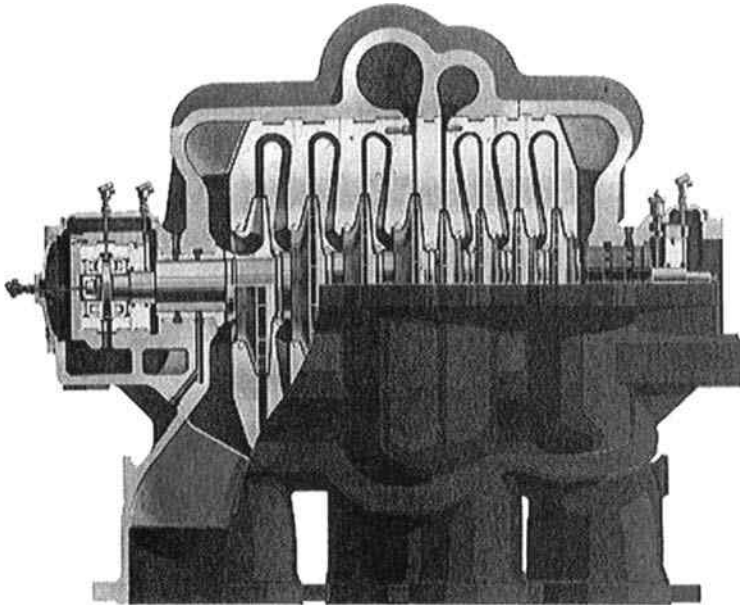


FIGURE 27.10 Multistage two-phase compressor.

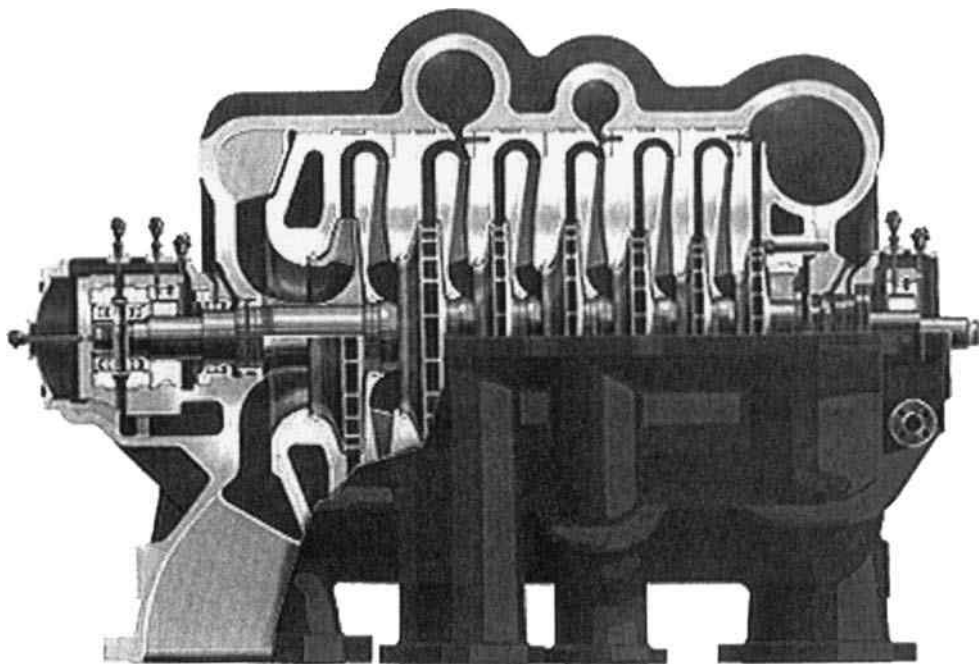


FIGURE 27.11 Multistage three-phase compressor.

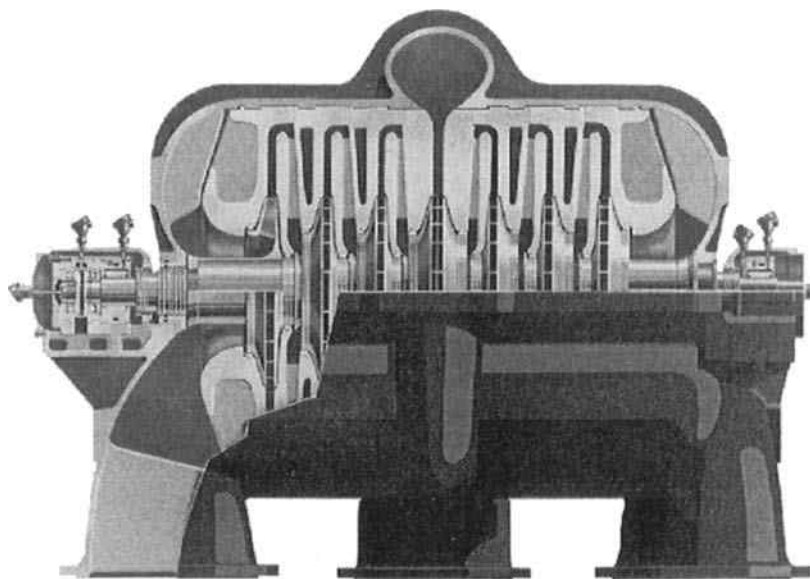


FIGURE 27.12 Two-phase compressor with a central double-flow impeller.

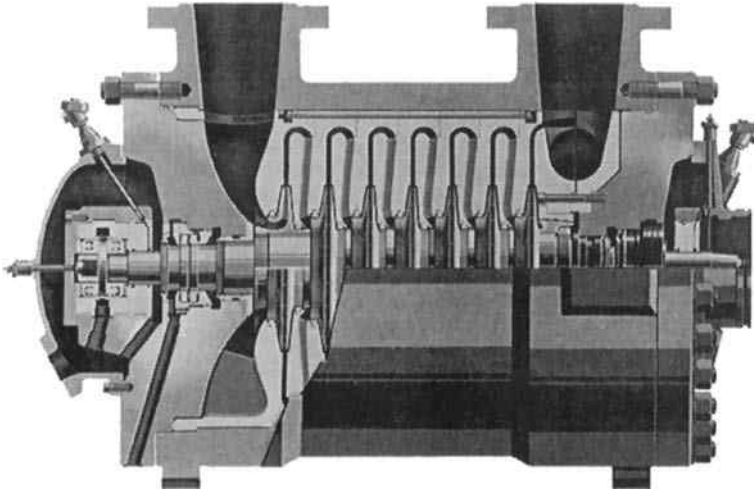


FIGURE 27.13 Barrel-type compressor with one compression phase.

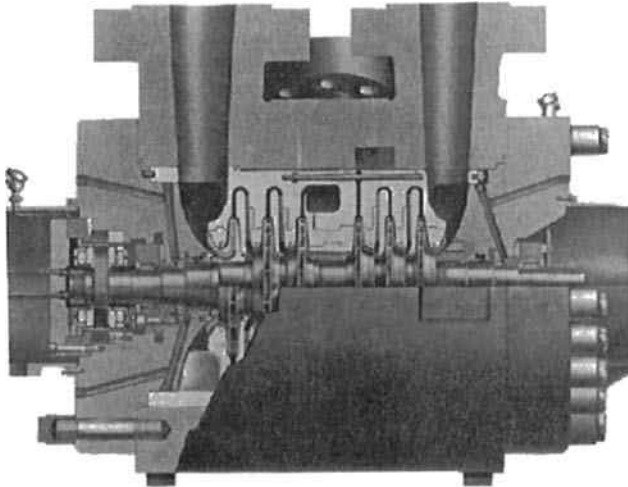


FIGURE 27.14 Barrel-type compressor with two compression phases.

27.2.3 Compressors with Bell Casings

Figure 27.15 illustrates a barrel compressor having a bell-shaped casing. These units are used for high-pressure applications.

27.2.4 Pipeline Compressors

Figure 27.16 illustrates a pipeline compressor. These machines have a bell-shaped casing. They also have a single vertical end cover. They are normally used for transporting natural gas. They generally have side suction and delivery nozzles. This is done to facilitate their installation on gas pipelines.

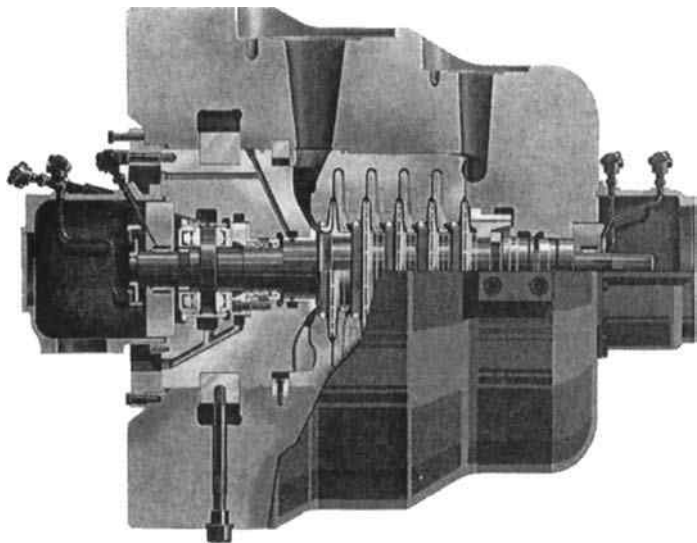


FIGURE 27.15 High-pressure barrel-type compressor.

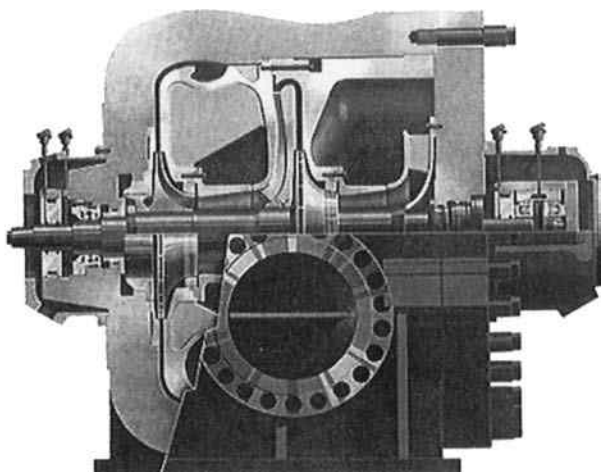


FIGURE 27.16 Pipeline compressor.

27.2.5 SR Compressors

Figure 27.17 illustrates an SR-type compressor. These units are used in relatively low-pressure applications. They have several shafts with overhung impellers. They normally have open-type (i.e., shroudless) impellers. This is done for the following three reasons:

1. High impeller tip speeds
2. Low stress levels
3. High-pressure ratios per stage

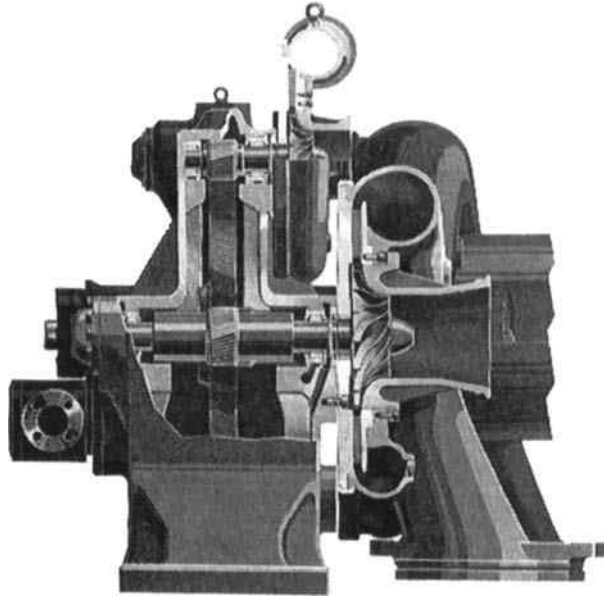


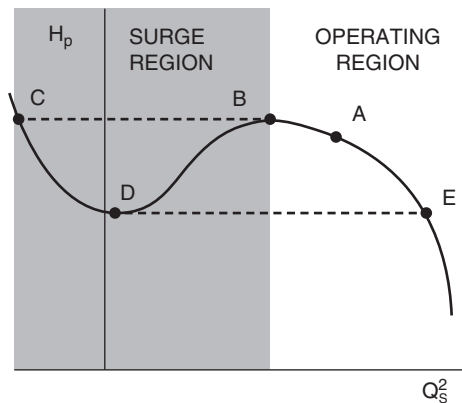
FIGURE 27.17 SR-type compressor

These units are normally employed in geothermal application, air or steam compression, etc.

27.3 Performance Limitations

Figure 27.18 illustrates a typical performance curve for a centrifugal- or axial-flow compressor. The performance of a compressor is limited not only by the maximum head that it can deliver at any given rotational speed and inlet condition but also by the maximum flow rate.

FIGURE 27.18 Performance curve illustrating typical performance control response. (Source: Compressor Controls Corporation, Des Moines, Iowa.)



27.3.1 Surge Limit

Consider a compressor operating at constant speed with fixed inlet conditions. The operating point of this compressor will be stationary. This is because it is operating initially at steady state. The operating point will move along the performance curve to the left if the network resistance downstream of the compressor increases. The compressor will eventually reach a point of minimum stable flow and maximum head. A potentially destructive phenomenon known as surge will occur if the compressor is operated to the left of this point. This point is known as the surge limit point. Figure 27.19 illustrates the surge limit line (SLL). This line represents the locus of surge limit points that occur at different rotational speeds.

The surge phenomenon can be visualized by considering the following example. Consider a system made from a compressor and discharge valve. During normal steady-state operation, the process gas will not accumulate over time in the plenum between the compressor and the valve. This mode of operation is represented by point A in Fig. 27.18. The flow rate through the valve will drop if it were closed slightly. The pressure in the plenum will increase. The new steady-state flow rate would be lower than the previous flow rate. The compressor would approach the surge limit (point B in Fig. 27.18) if the valve continued to close in small steps. The approach to point B would be noticed by a relatively large drop in flow and a slight increase in pressure. In other words, the slope of the compressor head-flow curve will drop near the surge limit point. A further closure of the valve at the surge point will decrease the flow. The compressor cannot achieve the pressure required to maintain the flow at a lower value than the one reached at the surge point. This causes a sudden drop in the flow. The flow (the flow is proportional to ΔP_o found on Fig. 27.20) will reverse direction within 20 to 80 milliseconds (Fig. 27.20). This change is shown as the dashed line BC in Fig. 27.18.

The operating point cannot be on the portion of the curve between B and D. This is because a higher pressure is required to deliver the flow through the valve. Note the drop in flow occurs before the drop in pressure (Fig. 27.20).

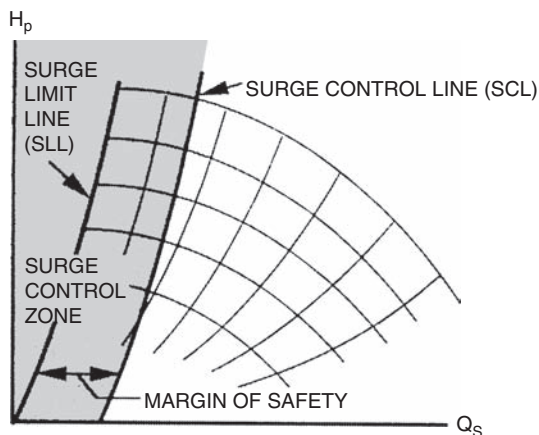


FIGURE 27.19 Compressor map showing the surge limit line (SLL), surge control line (SCL), and surge control zone. (Source: Compressor Controls Corporation, Des Moines, Iowa.)

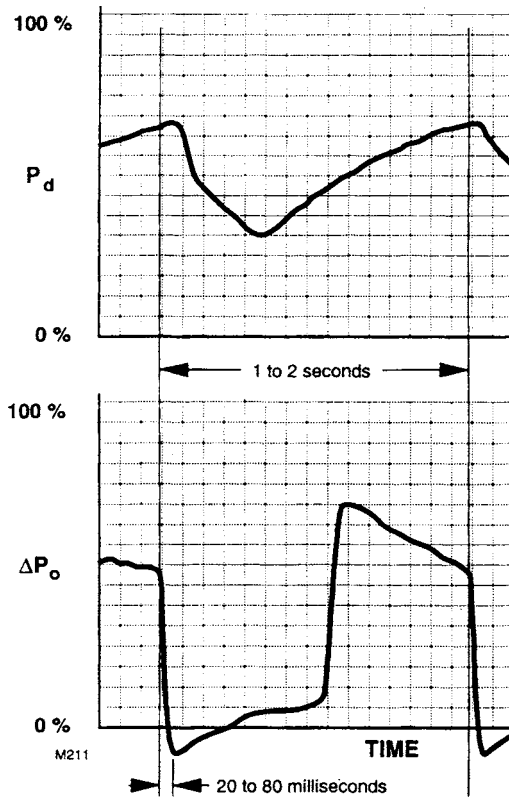


FIGURE 27.20 Pressure and flow variations during typical surge cycles. (Source: Compressor Controls Corporation, Des Moines, Iowa.)

The flow would be in the reverse direction (negative) at point C. However, energy is still being added to the gas. The pressure in the plenum between the compressor and valve would start to drop due to the discharge of the gas. The operating point of the compressor would move from C to D in Fig. 27.18. The compressor cannot stay at point D. This is because point D is a part of the highly unstable region (surge region). At point D, the compressor is still operating at constant speed and the valve is partially open. Thus, the operating point jumps quickly from point D to E. This is because this point belongs to the stable (safe) region of operation. Since the discharge valve is still at the same position that caused the surge initially, the operating point will move from point E to A. The compressor will repeat the cycle (A-B-C-D-E-A) again. This highly oscillatory pattern will continue until an action is taken to stop it. (e.g., the discharge valve is opened or the compressor fails). The severe oscillation of flow and pressure that occurs during the surge process will have the following consequences:

- High vibrations
- Increase in gas temperature
- High loads on the thrust bearing and impeller

Severe compressor damage and process upset will probably occur if the compressor is allowed to go through a few surge cycles.

27.3.2 Stonewall

Consider the same scenario of constant-speed compressor operation with fixed inlet conditions. The operating point of the compressor will move from point A toward the right along the performance curve if the network resistance decreases due to the opening of the valve. A point having maximum flow and minimum head will be reached eventually. A further decrease in the network resistance (opening of the valve) will not increase the flow rate. This point is known as the choke point or stonewall. This phenomenon is caused by the increase in gas velocity to the local acoustic velocity (Mach 1) at some point in the compressor.

Single-stage centrifugal compressors will not be damaged usually by the stonewall phenomenon. However, the rotor and blades of multistage centrifugal compressors could be seriously damaged by this phenomenon. A suitably designed antichoke controller should be used to manipulate an antichoke control valve. This controller will maintain sufficient system resistance to prevent choke.

27.3.3 Prevention of Surge

Surge occurs when the compressor cannot overcome the network resistance. Thus, surge can be prevented by decreasing the network resistance when the operating point approaches the SLL. This is accomplished by opening an anti-surge valve. A portion of the compressor flow will be recycled or discharged through this valve. The main disadvantage of this method is the decrease in the efficiency of the compressor. The energy that was used to compress the recycled gas will be wasted. Thus, the control system should be designed as follows:

- To open the anti-surge valve to the minimum flow required to prevent surge
- To open the anti-surge valve for the minimum duration required to prevent surge

The anti-surge control system involves maintaining an adequate margin of safety without causing an unnecessary reduction in efficiency. This is accomplished by maintaining the compressor operating point on or to the right of the surge control line (SCL) shown in Fig. 27.19.

27.3.4 Anti-Surge Control Systems

Anti-surge control systems are designed to maintain a minimum flow through the compressor to avoid surge. Figure 27.21 illustrates a single parameter surge protection system. These systems have been used extensively. They consist of the following:

- A flow measuring device
- An anti-surge valve

The anti-surge control systems will prevent surge if they are properly sized and installed. However, the single parameter surge protection systems are not suitable for the following applications:

- Suction throttle constant-speed compressors
- Compressors driven by variable-speed drivers

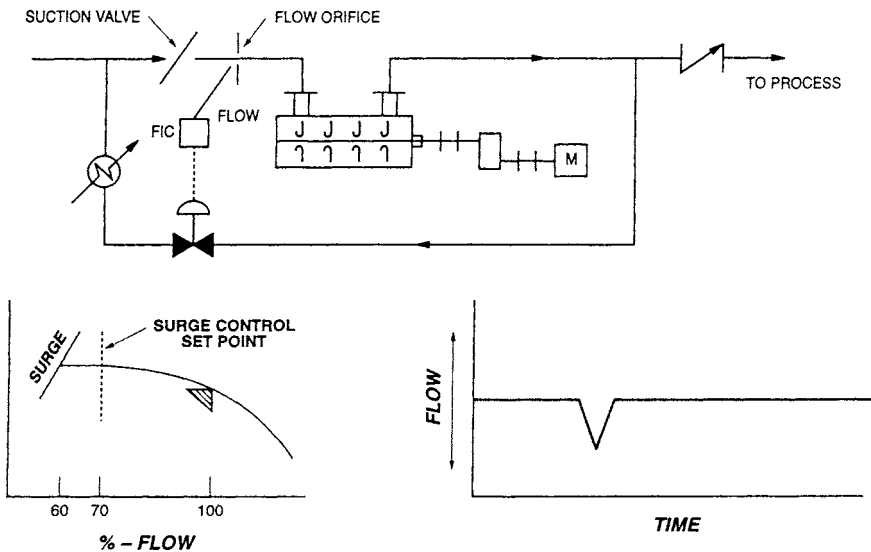
Motor Drive

FIGURE 27.21 A single parameter surge protection system.

The single parameters surge protection systems are not suitable for these applications because a large area of their operation will not be allowed. This will result in unnecessary recirculation and drop in efficiency.

Figure 27.22 illustrates a standard two parameter anti-surge control system. The instrumentation of this system measures the following parameters:

- Flow through the compressor
- Differential pressure across the compressor

This control system generates an SCL that is almost parallel to the actual surge line of the compressor. This method allows the compressor to operate over a wider range than the previous system. It would provide adequate protection for the compressor if it was properly sized.

Modern microprocessor systems are also used to prevent surge. The instrumentation of these systems measures the following parameters:

- Flow through the compressor
- Differential pressure across the compressor
- Compressor inlet temperature
- Compressor discharge temperature

The additional parameters measured by these systems compared with the previous systems allow the compressor to operate over a wider range. The SCL generated by these systems will be closer to the surge line generated by the previous method. This increases the efficiency of the compressor and minimizes the losses.

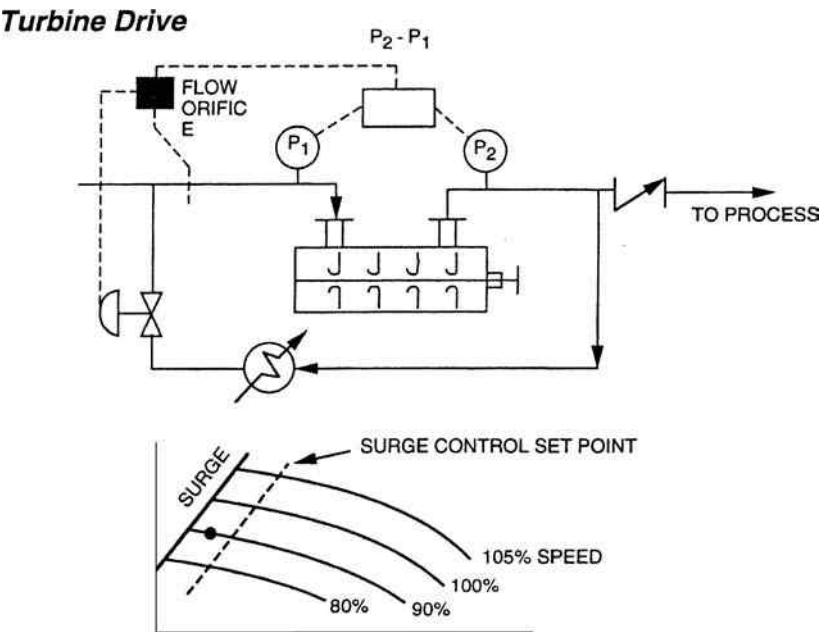


FIGURE 27.22 The standard two parameter or biased system.

27.4
Bibliography

Bloch, H. P., *A Practical Guide to Compressor Technology*, 2nd ed., John Wiley & Sons Inc., Hoboken, New Jersey, 2006.
Forsthoffer, W. E., *Forsthoffer's Rotating Equipment Handbooks, Vol. 3: Compressors*, 1st ed. Elsevier Ltd., Oxford, United Kingdom, 2006.
Halon, P. C., *Compressor Handbook*. McGraw-Hill, New York, 2001.

Compressor Seal Systems

28.1 Introduction

There is a wide variety of fluid seal systems. The design, sealing fluids, and sealing pressures used in these systems vary significantly depending on the application and manufacturer of the seal system. However, the function of a seal system can be defined as “to continuously supply clean fluid to each specific seal interface point at the required differential pressure, temperature, and flow rate.”

Figure 28.1 illustrates a typical seal system used for a centrifugal compressor. This system is designed for usage with clearance bushing seals. The following are the major sub-systems of the seal oil system:

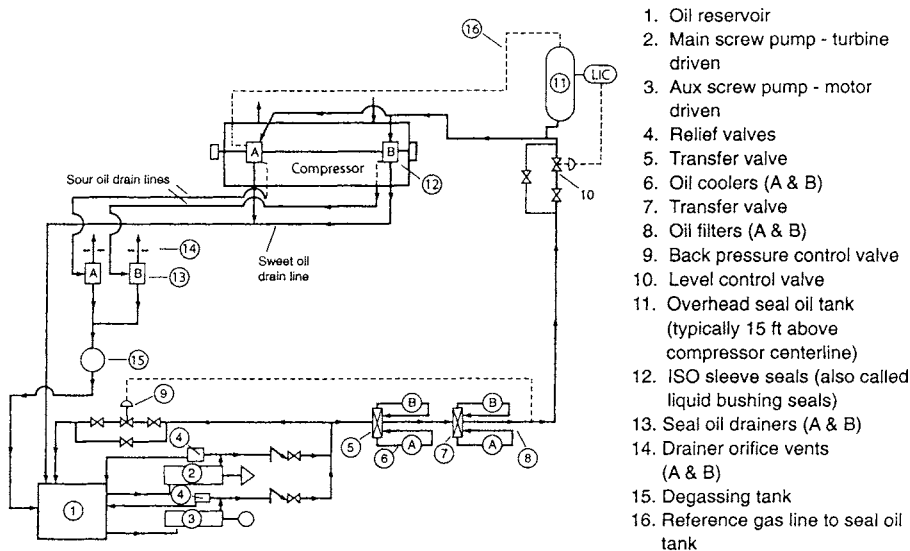
- The supply system
- The seal housing system
- The atmospheric drain system
- The seal leakage system

28.2 The Supply System

The supply system consists of the following equipments:

- Reservoir
- Pumping units
- Heat exchangers
- Transfer valves
- Temperature control valves
- Filters

The purpose of this subsystem is to supply a continuous, clean, and cool sealing fluid to the seal interfaces at the required differential pressure.



Note: component condition instrumentation and autostarts not shown

FIGURE 28.1 API 614 Lube/seal oil system for ISO-sleeve seals. (Courtesy of Elliott Group, Ebara Corporation.)

28.3 The Seal Housing System

Figure 28.2 illustrates a seal housing system. This system consists of the following seals:

- A gas side bushing
- An atmospheric bushing.

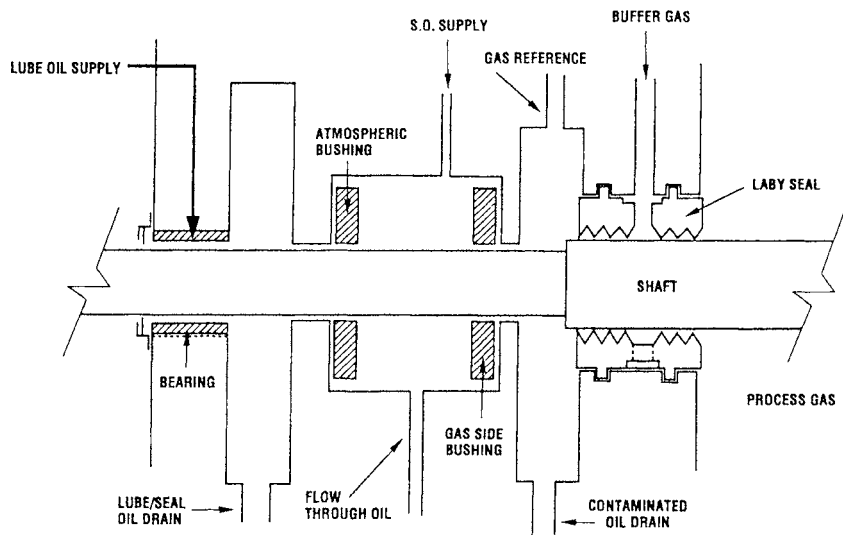


FIGURE 28.2 Bushing seal schematic. (Courtesy of M.E. Crane Consultant.)

The following are the purposes of the seal housing system:

- To positively contain the fluid in the compressor
- To prevent gas leakage from the compressor to the atmosphere

The seal fluid is introduced between the interfaces of both seals. This constitutes a double seal arrangement. The following are the purposes of the gas side bushing seal:

- To constantly contain the reference fluid
- To minimize sour oil leakage

The purpose of the seal created by the atmospheric bushing is to reduce the flow of the seal liquid to the amount required to remove frictional heat from the seal. Since the downstream pressure of this seal is normally atmospheric pressure, this seal can be conceptualized as a bearing. Some seal designs feed the oil from the atmospheric bushing directly into the bearing. The pressure downstream of the atmospheric bushing in this design will be maintained constant at around 137.4 kPa (20 psi). However, the upstream supply pressure in this design varies depending on the pressure required by the sealing media in the compressor. For example, if a seal system is designed to maintain a differential pressure of 40 kPa (5.82 psi) between the process gas inside the compressor and the seal oil supply to the gas side bushing, the seal oil supply pressure would be 40 kPa (5.82 psi) to both the gas side bushing and atmospheric bushing if the process gas pressure is 0 kPa. Thus, the gas side bushing and atmospheric bushing differential pressure would both be equal to 40 kPa (5.82 psi). However, if the process gas pressure were increased to 101 kPa (14.7 psi), the seal oil supply pressure would increase to 141 kPa (20.52 psi). This increase in the seal oil supply pressure will occur to maintain a 40 kPa (5.82 psi) differential pressure across the gas seal. The differential pressure across the gas side bushing will remain constant at 40 kPa (5.82 psi) in this case. However, the differential pressure across the atmospheric bushing will increase from 40 kPa (5.82 psi) to 141 kPa (20.52 psi). Thus, the main concern in any seal liquid system is ensuring adequate fluid flow to the atmospheric bushing under all conditions. The seal fluid leaving the seal chamber returns to the seal through two additional subsystems.

28.4 The Atmospheric Draining System

The oil leaving the atmospheric bushing through the lube/seal oil drain system returns to the seal oil reservoir. This oil enters the bearing system in some applications. There should be no gas in this oil. This is because it does not contact the process gas.

28.5 The Seal Leakage System

The fluid that enters the gas side bushing is normally controlled to a minimal amount. This is done to allow this fluid to be returned to the reservoir after it has been degassed. This fluid is discarded in some applications. The amount used of this fluid is normally limited to less than 75.8 L (20 gal) per seal per day. This liquid comes in contact with the shaft. A part of it is atomized because the shaft rotates at a high speed.

This atomized gas combines with the sealing gas. The gas mixture enters the leakage system. The seal leakage system consists of the following:

1. Automatic drainer
2. Vent system
3. Degassing tank (if furnished)
4. Supply system
5. Seal housing system

The following are the functions of each of these components:

- *The Drainer.* The mixture of oil and gas leaving the gas side seal enters the drainer. The liquid level in the drainer is controlled by any of the following methods:
 - An internal float
 - An external level control valve

The oil is drained back to the degassing tank or reservoir.

- *The Vent System.* The seal oil drainer vent system is referenced to a lower pressure than the pressure in the gas side seal. The vented gas from this system is routed back to any of the following:
 - Compressor suction
 - Suction vessel
 - A lower pressure source
- *The Degassing Tank.* The degassing tank is normally heated. The residence time in this tank is more than 72 hours normally. This provides enough time for the seal oil to degas. The degassed oil returns to the reservoir. This oil meets all the seal oil specifications (viscosity, flashpoint, dissolved gases, etc.) at this stage.
- *The Supply System.* The seal oil supply system provides sealing oil to the atmospheric bushing and gas side bushing. The downstream pressure of the atmospheric bushing is the atmospheric pressure. This pressure remains constant. However, the downstream pressure of the gas side bushing varies during operation. This pressure is referenced to the compressor process gas pressure. Thus, the seal oil supply system provides a constant differential pressure control between the gas pressure and the seal oil supply pressure. The flow through the gas side bushing remains constant. This is because the differential pressure across it remains constant. However, the flow through the atmospheric bushing seal varies. This is because the differential pressure across it varies. For example, the flow through the atmospheric bushing will be low when the compressor is at atmospheric pressure. This is because the differential pressure between the seal oil supply and the compressor pressure must remain constant. However, the flow through the atmospheric bushing will increase as the pressure inside the compressor increases. Some seal systems have had problems maintaining adequate oil flow through the atmospheric bushing when the compressor pressure is low.
- *The Seal Housing System.* The purpose of any seal is to contain the fluid in the vessel of the equipment (pump, compressor, turbine, etc.). There is a wide variety of seal types and designs. Figure 28.3 illustrates a typical mechanical

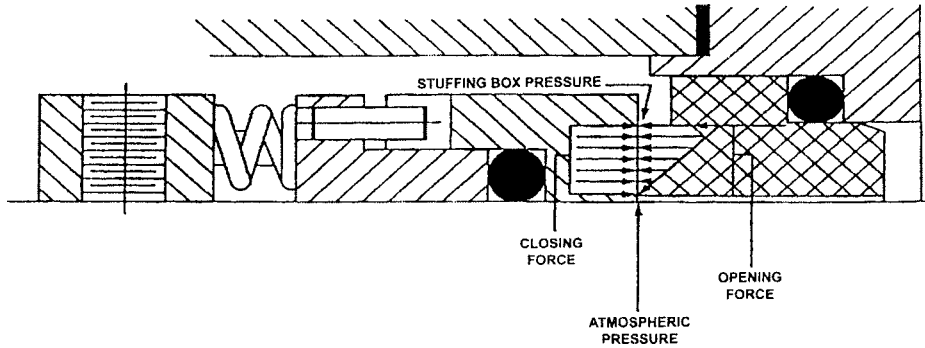
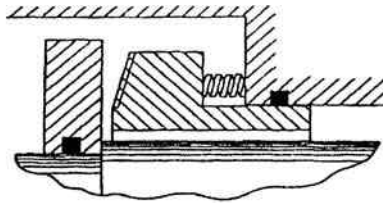


FIGURE 28.3 Typical pump single mechanical seal.

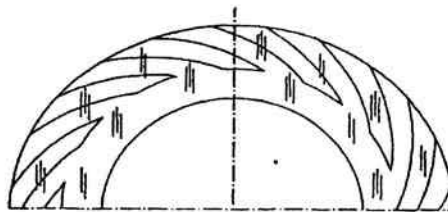
seal used for a pump. This seal has a stationary and a rotating part. The frictional heat generated between these two parts is removed by the liquid. A small amount of liquid vaporizes during this process. This vapor leaves the pump constantly across the seal face. Seal-less pumps are required for many applications to prevent the emission of toxic vapors. The following are the major types of seal combinations used in centrifugal compressors.

28.6 Gas Seals

Figure 28.4 illustrates a typical gas seal. This seal design is much simpler than the traditional liquid seal system. The principle of operation of a gas seal is to maintain a fixed minimum clearance between the rotating and stationary face of the seal. A lifting device



Typical Design For Curved Face — Spiral Groove Non-contact Seal;
Curvature May Alternately Be On Rotor



Typical Spiral Groove Pattern On Face Of Seal
Typical Non-contact Gas Seal

FIGURE 28.4 Typical gas seal. (Courtesy of John Crane Co.)

is used to maintain this minimum clearance between the two faces of the seal. Gas seals use the sealed gas or a clean buffer gas to create the seal. They do not require a liquid sealing system including pumps, a reservoir, and other components. However, the sealing gas must be supplied to the seal at the proper flow rate, temperature, and cleanliness. Thus, a highly efficient, reliable source of filtration, cooling, and supply of the sealing gas must be provided. Some gas seals rely on an inert buffer gas to create the seal. The supply source of this buffer gas must be very reliable. The compressor will be as reliable as this supply source. The cleanliness of the sealing gas is essential. The effectiveness of the sealing devices will be reduced significantly due to buildup of debris entrained with the sealing gas. This will result in a rapid damage to the seal faces.

28.7 Liquid Seals

Liquid seals have been commonly used in compressors. The liquid introduced in the seal performs the following functions:

- Ensures adequate sealing of the gas
- Removes the frictional heat of the seal

All compressor liquid seals consist of two seals. The sealing liquid is introduced between the sealing faces. Figure 28.5 illustrates a liquid seal. The differential pressure across the gas side (inboard) and atmospheric side (outboard) seals provides sufficient flow through the seals to remove the frictional heat at the maximum operating speed. The gas side seal shown in Fig. 28.5 is a contact seal. This seal is similar to the seals used in most pump applications. This design provides reliable operation and minimum oil leakage (20–40 L, 5–10 gal per day per seal). The compressor can operate continuously for more than 3 years with this seal design.

28.8 Liquid Bushing Seals

Figure 28.6 illustrates a typical liquid bushing seal. This seal design can be used for the gas side or atmospheric side seal application. However, it is used commonly in compressors for atmospheric bushing applications.

The flow through a bushing seal is governed by the same principle as an orifice. Thus, the clearance between the shaft and the bushing surface should be minimized to reduce the leakage. This clearance should be around 0.0005 cm diametrical clearance per centimeter of shaft diameter (0.0005 in diametrical clearance per inch of shaft diameter).

Liquid bushing seals are also used for gas side seal applications. However, their leakage rate will be significantly higher than that of a contact seal in these applications. This is because liquid bushing seals act as an orifice. Thus, they must be designed to minimize the differential pressure across the bushing when used for gas side seal applications. A differential pressure control system must be utilized to achieve this objective. This system must be able to establish and maintain accurately a specific oil/gas differential pressure under all operating conditions. The typical design differential pressure across a gas side bushing seal is around 34 kPad (5 psid) to 68 kPad (10 psid). A level control system is usually used to maintain an accurate control of this differential pressure.

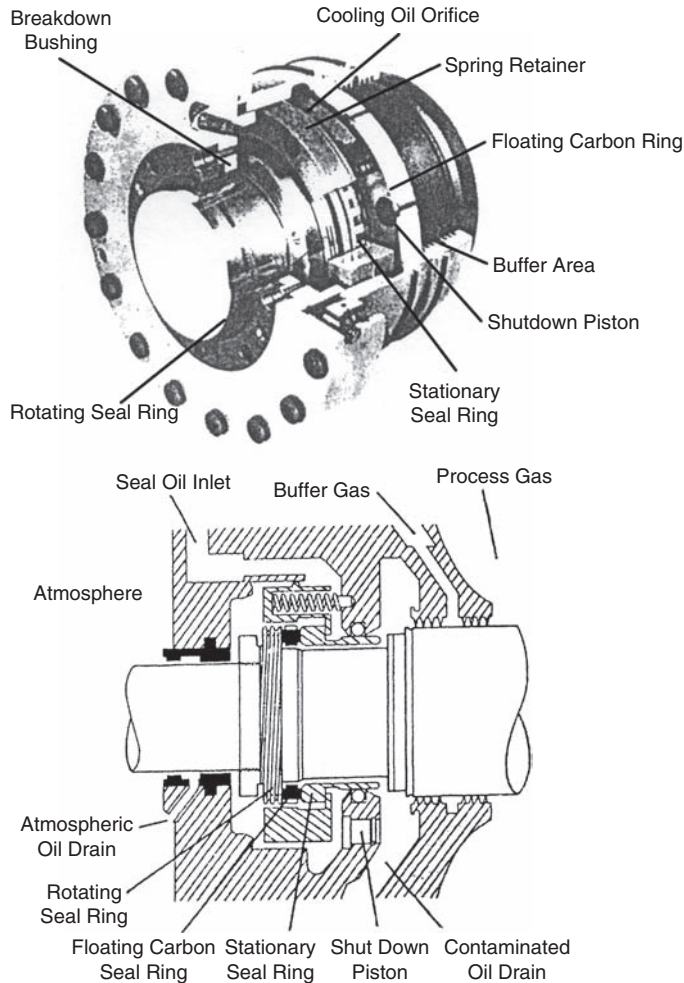


FIGURE 28.5 Iso carbon seal. (Courtesy of Elliott Group, Ebara Corporation.)

The adequate operation of a bushing seal (Fig. 28.6) depends totally on maintaining a liquid interface between the seal and shaft surface. The gas will leak outward through the seal if this liquid interface fails. The seal system must be in operation whenever there is pressurized gas inside the compressor. The ingress of process gas into the supply system should always be prohibited. This can be accomplished by implementing one of the following:

- Continuous supply of buffer gas.
- Installation of a check valve in the seal oil supply header. This check valve should be located as close as possible to the seals.
- The compressor casing should be isolated and vented immediately upon failure of the seal system.

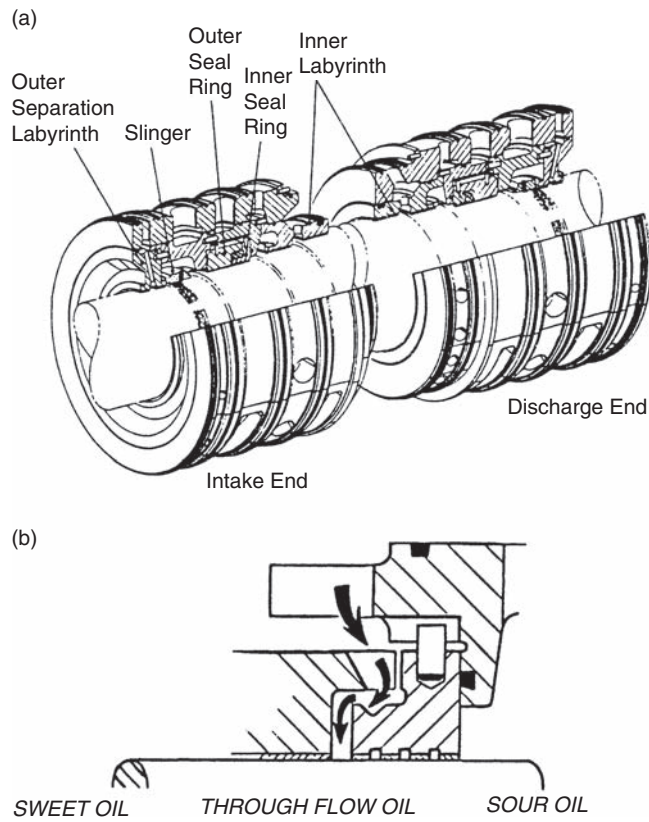


FIGURE 28.6 Bushing seal—(a) oil film seal; (b) seal oil flow. (Courtesy of Dresser-Rand.)

The supply piping of the seal oil must be thoroughly checked for debris before restarting the compressor if the second or third method listed above is implemented. Many problems have occurred in bushing seals due to this problem.

28.9 Contact Seals

Figure 28.7 illustrates a typical compressor contact seal. The design of these seals is similar to pump seals. The differential pressure between the seal oil supply and the reference gas should be maintained between 240.5 kPa (35 psi) and 343.5 kPa (50 psi) for these seals. This differential pressure is required to remove the heat generated by friction in the seal. The oil leakage rate can be maintained between 1.9 L (5 gal) and 38 L (10 gal) per day per seal.

Contact seals are used in applications having a maximum speed of 12,000 r/min. The reason for this limitation is that the heat generated by the rubbing of the sealing surfaces increases significantly with the speed. Bushing seals are used above this speed. The reason for this is that these seals have a lower rubbing speed. This is due to

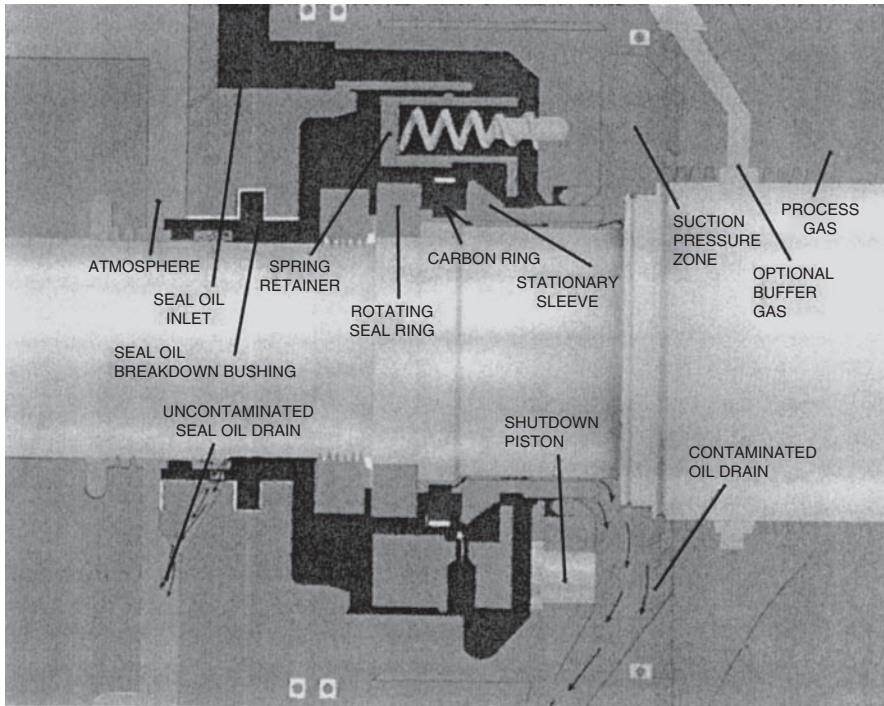


FIGURE 28.7 Compressor contact seal. (Courtesy of Elliott Group, Ebara Corporation.)

their smaller diameters. The maximum differential pressure across contact seals is 1.37 MPa (200 psi). This limit is determined by the materials of the seal. Thus, contact seals are not used commonly for atmospheric seal applications. This is because they are differential pressure limited. However, these seals are used commonly for gas side seal applications.

28.10 Restricted Bushing Seals

Figure 28.8 illustrates a restricted bushing seal. This seal employs a small pumping ring. The purpose of this ring is to generate a flow in the opposite direction of the bushing liquid flow. This is done to reduce the relatively large leakage that occurs in bushing seals. This seal design can prevent the leakage flow. However, these seals are normally designed to have a small leakage flow at maximum operating speed. They are used exclusively for gas side service.

The following is a summary of the basic types of liquid seals used in compressors:

- Open bushing type (used for either gas or atmospheric side seal applications)
- Contact type (used for gas side seal applications)
- Restricted bushing type (used exclusively for gas side seal applications)

Compact Design — allows shorter bearing spans for higher critical speeds of the compressor rotor.

Sleeve (impeller) with interference fit under bushing — protects shaft and simplifies assembly and disassembly. Requires only a jack/puller bolt ring.

Spacer fit at initial assembly — no field fitting of parts.

ITEM DESCRIPTION

- 1.....Shaft
- 2.....Impeller
- 3.....Stator
- 4.....Stepped Dual Bushing
- 5.....Bushing Cage
- 6.....Nut
- 7.....Shear Ring
- 8.....Oil/Gas Baffle
- 9.....Spacer Ring

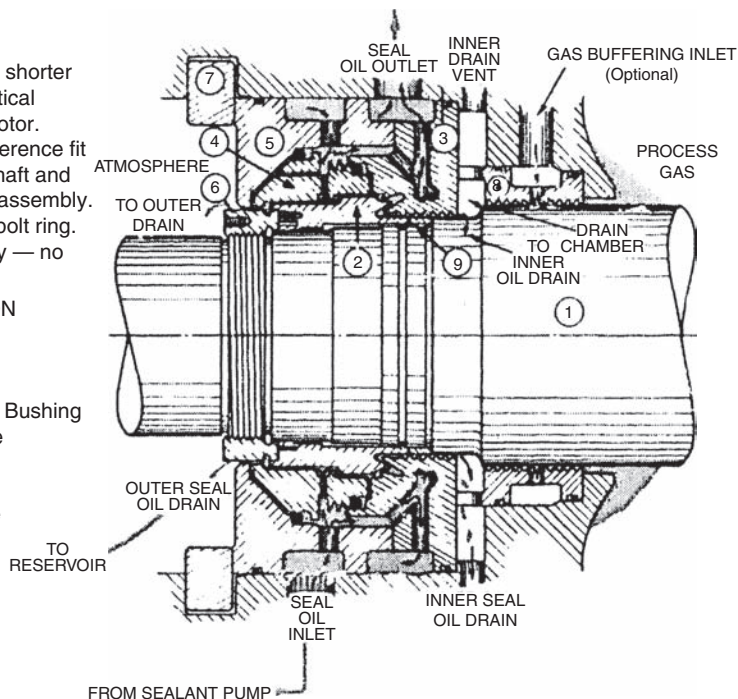


FIGURE 28.8 Turbo-compressor “trapped bushing seal.” (Courtesy of A.C. Compressor Corp.)

28.11 Seal Supply Systems

The following is an examination of different types of seal systems. Each system employs different types of compressor seals. The function of each seal system will be discussed in detail.

Example 28.1 Contact-Type Gas Side Seal—Bushing-Type Atmospheric Side Seal with Cooling Flow

This seal system (Fig. 28.9) employs a contact seal on the gas side and a bushing seal on the atmospheric side of each compressor end. The supply fluid on each end of the compressor is referenced to the suction pressure. Different reference pressures are used in different compressor applications. The reference pressure is normally taken from either the balance drum end or the high-pressure end of the compressor. This is done to ensure that there is an adequate differential pressure between the oil and the gas. Thus, the low-pressure end of the compressor may experience a slightly higher differential pressure between the oil and the gas than the high-pressure (reference end) end of the compressor. The oil enters the seal chamber through the supply line. A differential pressure control valve maintains the differential pressure across the gas side contact seal. The seal oil flows in the following directions:

- Through the seal chamber (cooling flow)
- Through the gas side contact seal [38 L (10 gal) – 76 L (20 gal) per day per seal]
- Through the atmospheric seal

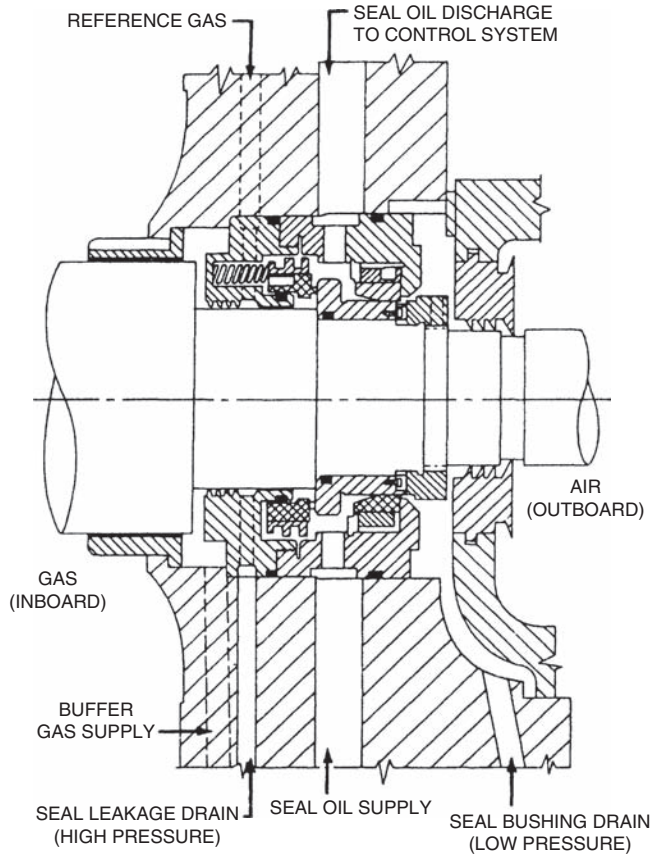


FIGURE 28.9 Compressor shaft seal. (Courtesy of IMO Industries.)

28.11.1 Flow Through the Gas Side Contact Seal

The flow through the gas side contact seal is constant. This is because the pressure differential across this seal is maintained constant. The flow through this seal is normally less than 38 L (10 gal) per day. This flow is negligible when compared with the remaining flows through the seal. Thus, it is assumed to be 0 L/min (gal/min).

28.11.2 Flow Through the Atmospheric Side Bushing Seal

The flow through the atmospheric side bushing seal varies depending on the referenced gas pressure. This flow will be low when the pressure of the gas being sealed is low. (the reference pressure is almost the same as the pressure of the gas on the inner side of the seal). This is because the differential pressure between the seal oil supply and the reference pressure is maintained constant by a control system. The flow through the atmospheric side bushing will increase when the reference pressure will increase. The design of the atmospheric side bushing seal should take into consideration the variations in reference pressure. The seal oil flow through the atmospheric side bushing seal should be adequate under all conditions.

28.11.3 Flow Through the Seal Chamber

The flow through the seal chamber removes the frictional heat from the seal. This flow is regulated by a downstream control valve. The following is an example of a specific seal application:

1. The gas side seal flow is 0 L/min (0 GPM)
2. Atmospheric side seal flow
 - a. When the reference pressure is 0 kPag (0 psig), the atmospheric side flow is 11.37 L/min (3 GPM)
 - b. When the reference pressure is 1.37 MPag (200 psig), the atmospheric side seal flow is 45.5 L/min (12 GPM)
3. Flow through the seal chamber
 - a. When the reference pressure is 0 kPag (0 psig), the flow through the seal chamber is 45.5 L/min (12 GPM)
 - b. When the reference pressure is 1.37 MPag (200 psig), the flow through the seal chamber is 11.37 L/min (3 GPM)
4. Seal oil supply flow

The seal oil supply flow is 56.85 L/min (15 GPM) in both cases.

28.12 Seal Liquid Leakage System

The function of the seal liquid leakage system is to collect all the leakage from the gas side seal. This system returns the collected liquid to the seal reservoir. The gas being compressed can, in some applications, alter the specifications of the seal oil. A clean buffer gas is introduced into the seal (Fig. 28.9) in applications where the seal oil becomes unacceptable for re-usage. The gas and the oil are present in the seal oil drainer. A vent is installed in the drainer pot of some seal systems to remove some of the gas. A degassing tank is used in some applications to remove the gas from the oil.

28.13 Bibliography

Forsthoffer, W. E., *Forsthoffer's Rotating Equipment Handbooks, Volume 3: Compressors*, Elsevier Ltd., Oxford, United Kingdom, 2006.

CHAPTER 29

Dry Seals, Advanced Sealing Mechanisms, and Magnetic Bearings

29.1 Introduction

During the 1970s, several programs were initiated to identify, research, and eliminate the numerous problems associated with oil systems on compressors. The following are the purposes of these programs:

- Improvement of the safety of compressor systems
- Reduction in the maintenance and operating costs

Fully functional dry seals were operating successfully in 1978. The first oil-free compressor using both dry seals and magnetic bearings operated successfully in 1985. Recent advancements in thrust-reducing seals and magnetic bearing control systems have increased the reliability and reduced the operating and maintenance costs of compressor systems.

29.2 Background

The studies performed in the 1970s indicate that a large portion of the downtime on compressor systems was related to problems with the seal or lube oils systems. In 1978, the dry seal had reliably replaced the conventional seal oil systems in centrifugal compressors. In 1985, an active magnetic bearing system was installed in a centrifugal compressor that has been retrofitted earlier with dry seals. This compressor operated successfully for more than 9 years since the bearing retrofit. In 1988, magnetic bearings were retrofitted on the power turbine* driving this compressor. The operating history indicates that magnetic bearings operate successfully in this higher-temperature

*The power turbine is the component of the gas turbine that drives the compressor.

environment. This allowed the elimination of the lube oil systems that serves the compressor and the power turbine. Many compressor systems have operated successfully using one or both of these technologies since the late 1980s.

29.3 Dry Seals

29.3.1 Operating Principles

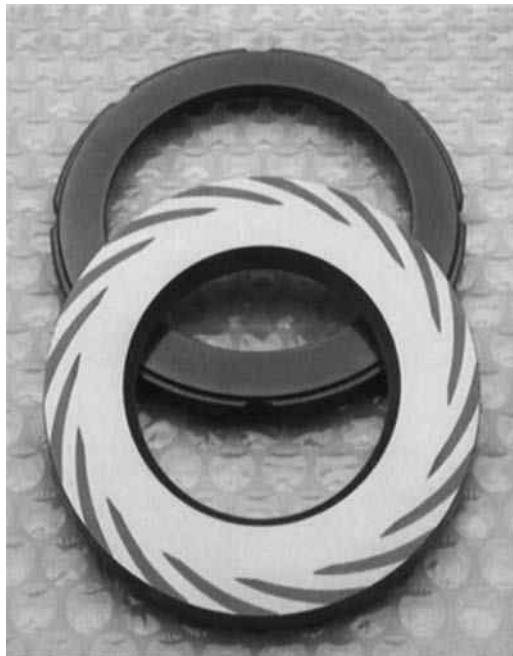
The design of dry seals is based on the gas film technology. This technology has been used successfully in air bearings in high-precision machining and measurement equipment. This seal consists of two seal rings (Fig. 29.1). The mating ring has grooves etched into a hard face. This ring rotates with the shaft. The primary ring is restrained from movement except along the axis of the shaft. This ring has a softer face.

Springs are used to force the faces of the rings against each other. The spring forces result in contact of the faces when the compressor is shut down and depressurized. The gas static pressure acting on the seal mechanism allows a minimal volume of gas to leak through the faces when the compressor becomes pressurized.

During normal operation, the combination of the following forces generates a noncontact seal face equilibrium:

- Process gas pressure (hydrostatic forces)
- Pumping pressure generated by the spiral grooves (hydrodynamic forces)

FIGURE 29.1 Dry seal rings. (Source: Revolve Technologies, Calgary, Alberta, Canada.)



The gas film pressure drops due to an increase in the clearance. The hydrostatic gas pressure acting on the faces tends to decrease the clearance. This seal design does not experience mechanical wear. This is due to the noncontacting nature of the faces.

Figure 29.2 illustrates a single dry seal. Double and tandem seals (Fig. 29.3) are also used in industry. Tandem seals are used commonly in natural gas pipeline applications. Each stage of this seal is capable of sealing against the full process gas pressure. This pressure varies up to 10.35 MPa.

During normal operation, the stages of the seal perform the following functions:

- The primary stage seals against full process gas pressure.
- The second stage is unloaded.

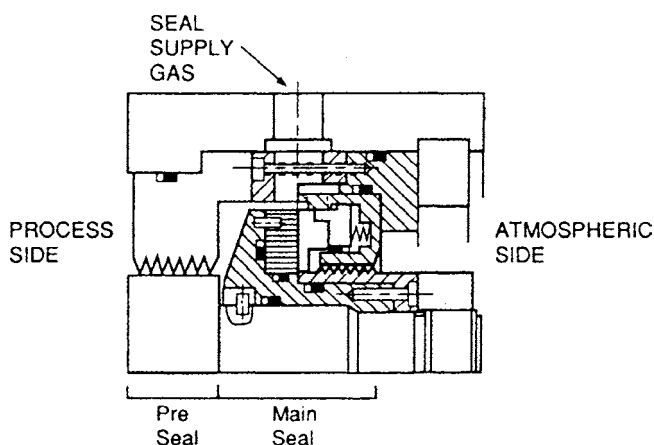


FIGURE 29.2 Single dry seal configuration. (Source: Revolve Technologies, Calgary, Alberta, Canada.)

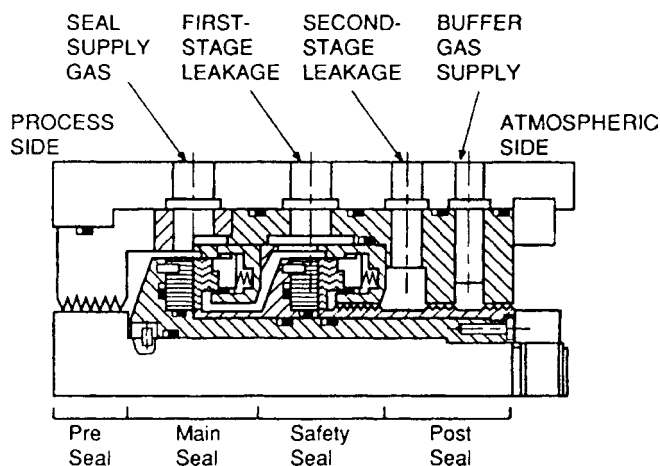


FIGURE 29.3 Tandem dry seal configuration. (Source: Revolve Technologies, Calgary, Alberta, Canada.)

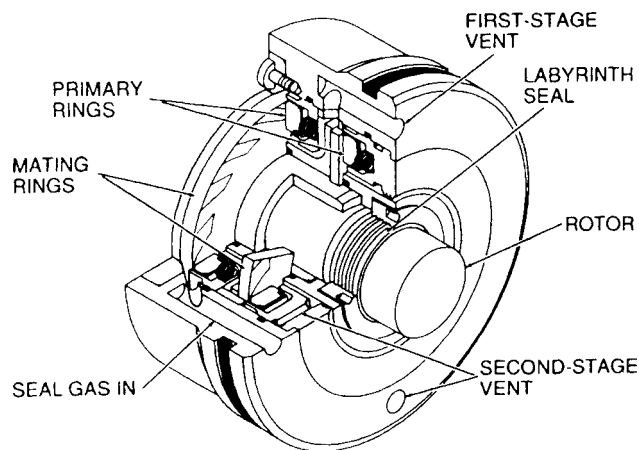


FIGURE 29.4 Dry seal cartridge. (Source: Revolve Technologies, Calgary, Alberta, Canada.)

The secondary stage seals the process gas when the primary stage seal fails. This design provides a backup seal for safe shutdown. The entire seal assembly is encapsulated (Fig. 29.4). This assembly can be installed and removed as a complete unit. This feature facilitates the installation of the seal.

Figure 29.5 illustrates the monitoring and control system of a dry seal. There is a clean gas supply to each seal. This feature prevents potentially dirty process gas from

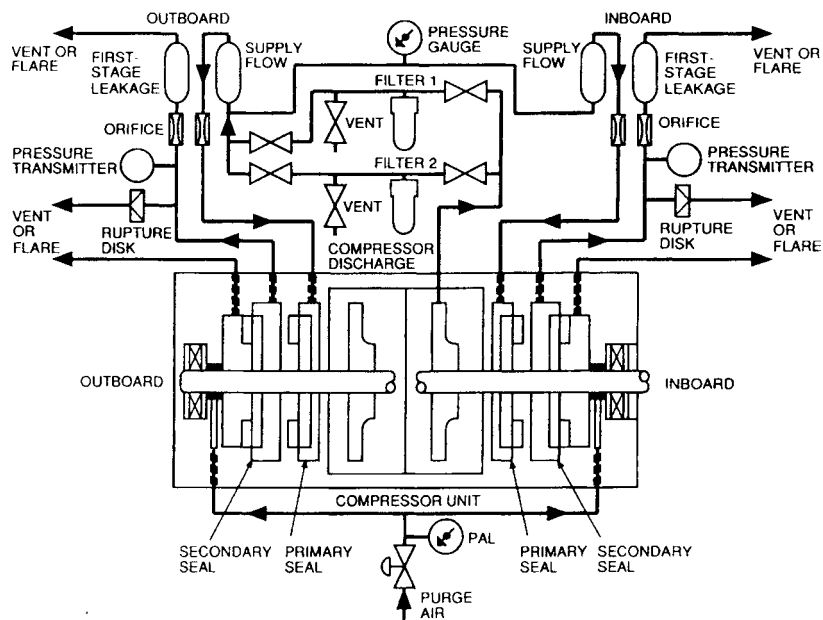


FIGURE 29.5 Dry seal monitor and control system. (Source: Revolve Technologies, Calgary, Alberta, Canada.)

entering the seal. The seal gas supply is normally taken from the compressor discharge piping. However, other sources have been used in some applications. The seal gas supply is filtered before entering the seal. A coalescing 0.1- μm filter is used in applications involving liquid entrainment in the gas.

The seal gas flow is monitored by a small flow meter. This flow is sent to the seal gas supply. The majority of this flow reenters the process cavity across a labyrinth seal. The grooves pump the leakage flow across the seal faces. The flow passes across a smooth dam area of the mating ring. It then reaches the pressure in the first-stage leakage port. The leakage flow is measured at this point. This is because it provides an important indication about the health of the seal. The leakage flow should remain within predetermined limits. The increase or decrease in this flow beyond these limits will result in an alarm or shutdown of the compressor. A second flow meter provides a visual indication of this leakage flow. This allows the operators to determine the cleanliness of the gas leaving the seal. The leakage flow depends on the size and speed of the compressor. However, it is typically below 80 L/min (3 scfm). Oil from the adjacent bearing cavity can contaminate the seal. Oil contamination is prevented by using a buffer gas such as nitrogen or instrument air.

29.3.2 Operating Experience

Dry seals were retrofitted on more than 30 compressors. The shaft size of these units varied from 45 to 255 mm (1.75–10 in). The pressure varied up to 10.88 MPa (1600 psig) and the speed up to 27,060 rpm. Dry seals were also installed on new compressors. As of 2005, more than 100 compressors were operating with dry seals. These seals have improved the safety of these compressors. They have also decreased the operating and maintenance costs of the units. The reliability of these compressors is higher than that of the compressors that use oil systems. Dry seals should be considered for most centrifugal compressors. These seals are now an accepted design standard for most industries.

29.3.3 Problems and Solutions

In the past, problems occurred due to contamination of the seal gas. This is because the seal faces must be extremely flat. The soft primary ring can be damaged by even minute particulates. This will disrupt the stability of the seal film. These problems were resolved by improving the filtration system. This includes the installation of a coalescing filter upstream of the original. A separate source of seal gas should be provided during the commissioning phase of a new compressor. Nitrogen is normally used in most applications. This protects the seals from contamination. Debris can be present in new compressors or intake piping. This debris normally causes problems during the first few hours of operation.

The second problem encountered with these seals is the explosive decompression of o-rings. The gas becomes entrapped in the elastomeric o-ring materials when the pressure is 4.5 to 11 MPa (650–1600 psi). This gas could blister or burst the o-rings as it attempts to escape when the compressor is depressurized. This problem has been resolved as follows:

- Using higher-density elastomers
- Changing the compressor control logic to depressurize during an emergency only

29.3.4 Upgrade Developments of Dry Seals

The design of a dual hard-face dry seal started in 1986. Hard faces allowed complete static separation of the faces. The use of hard silicon carbide in the primary ring provides this capability. It also allowed these seals to be used in applications having a pressure as high as 20 MPa (2900 psi). Dry seals have provided great results since 1988.

Recently, additional groove patterns were used successfully. Figure 29.6 illustrates bidirectional T-groove seals. This design was used first in 1991. The main advantage of this design over spiral groove and other one-directional mating ring designs is that it can be installed on either end of the compressor. This reduces the spares inventory requirements. This design will also operate successfully if the compressor rotates in reverse. Bidirectional T-groove seals have proven to be as effective as spiral groove seals. They also provide substantially lower leakage than spiral groove seals (Fig. 29.7).



FIGURE 29.6 Bidirectional dry seal faces. (Source: Revolve Technologies, Calgary, Alberta, Canada.)

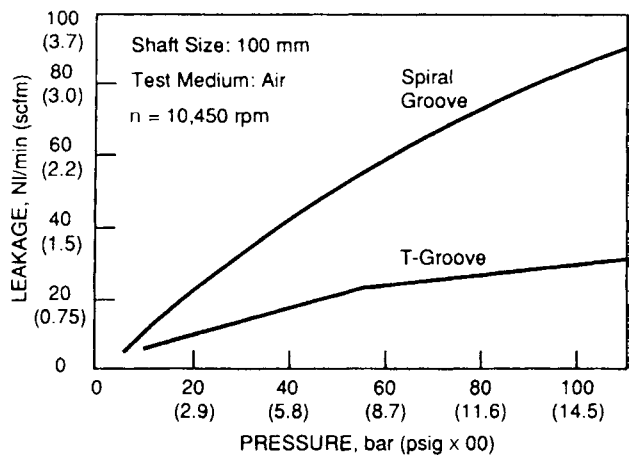


FIGURE 29.7 Groove shape and seal leakage rates. (Source: Revolve Technologies, Calgary, Alberta, Canada.)

This feature is important in natural gas pipeline applications. This is because the compressors used in these applications vent their seal leakage to the atmosphere. Since there is increasing requirements to reduce environmental emissions, bidirectional T-groove seals provide the best option for these applications. These seals have been used successfully in several industries since 1994.

29.3.5 Prevention of Dry Gas Seal Failures by Gas Conditioning

Diagnostics of modern dry gas compressor seals that failed during operation indicate that 90% of all seal failures are caused by a lack of clean and dry buffer gas. The most important requirement for trouble-free operation and high seal longevity is providing a continuous supply of clean and dry buffer gas. The failure to provide this requirement particularly during the commissioning period will lead to multiple seal failures. This results in downtime and delays in plant start-up.

The control system for gas seals consists of the following:

- Filtration
- Regulation
- Monitoring

The filtration is often overlooked. A standard filtration system is installed on almost every application. The gas composition and/or the presence of liquid or condensation in the gas is often ignored. This problem is the cause of most seal failures. The solution of this problem involves a review of the following:

- Gas composition
- Commissioning procedures
- Control system design
- Various interfaces between the seal, compressor, and seal control system

A common problem is determining when condensation will occur from a drop in pressure or temperatures. Many seal failures are caused by condensate contamination. Condensate are formed in some applications due to depressurization of the buffer gas across a pressure regulator valve which is located downstream of the filters. This problem can be avoided by relocating the pressure regulator upstream of the filter. This problem can be eliminated permanently by adding a heater and insulating the entire line.

Many seal manufacturers developed gas conditioning units (GCUs). These units deliver clean, dry, properly pressurized gas to the seals. Well-engineered GCUs have greatly enhanced the reliability of dry gas seals. This is because they resolved the critical gas supply problems. The conventional gas panels include only coalescing filters. A modern GCU includes the following features:

- *Knockout filter/coalescer vessel.* This equipment removes solid particulates and free liquids and aerosols.
- *A heater controller.* This device monitors and maintains the gas temperature.

The gas temperature is maintained above its dew point. This prevents condensation of aerosols in the process gas stream. Thus, GCUs manage liquids effectively to ensure the cleanest possible gas supply is always available.

Need for Training

The lack of training will increase the failure rate of seals. Many maintenance technicians in plants having dry gas seals lack essential training in the operation and maintenance of these seals. Most maintenance technicians are familiar with conventional or “wet” seals. Thus, they are used to seeing flooded seal cavities with no consequences. In contrast, dry seals do not require any lubrication. In fact, their support systems must prevent liquids, including lube oil, from reaching the seal faces.

Maintenance technicians should receive training in the operation and maintenance of dry gas seals. The seal manufacturer should provide this training before the installation of the seals. Also, it is essential to choose a seal manufacturer that provides guidance to increase equipment reliability, and the mean time between replacement or repair.

Many industries have improved their performance significantly by training their personnel in dry gas seals. This is because wet seals are notorious for allowing the seal oil to enter the compressor. This reduces the effectiveness of heat exchangers and other equipment located downstream of the compressor. The conversion to dry gas seals eliminates these problems. Most gas seals operate for more than a decade without problems. These seals are normally found with little wear and tear after operating in the compressor for more than 10 years. Extensive training in these seals contributes significantly to this success.

Minimizing the Risk of Sealing Problems

The minimization of seal failures is accomplished by performing a thorough review of plant components and their impact on the total system. The following factors should be considered in order of importance when examining dry seal support systems for centrifugal compressors:

1. *Gas composition.* Successful operation requires a thorough understanding of the actual gas composition and true operating condition. For example, it is essential to determine when and where condensation will occur in the sealing fluid.
2. *Commissioning procedures.* The following requirements should be confirmed during the commissioning phase:
 - a. Providing clean and dry buffer gas.
 - b. The seal is protected from bearing oil.
 - c. Detailed procedures for pressurizing and depressurizing the compressor.
 - d. Detailed procedures for taking the machine to full speed.
 - e. All personnel are familiar with the compressor operating and maintenance manual.
 - f. Detailed manuals that describe the control system.
3. *Control system design.* A clean and dry buffer gas should always be provided to the compressor. The main requirements of the buffer gas system design should include the following:
 - a. Gas conditioning
 - b. Filtration
 - c. Regulation (flow vs. pressure)
 - d. Monitoring.

The control system should be reviewed thoroughly to ensure that problems do not occur. The design and logic of the control system should be understood.

The buffer gas conditions should also be reviewed. Ensure that a heater is installed if required. Determine the temperature setting of the heater.

4. *Interface between the compressor, seals, and control system.* The following should be reviewed thoroughly:
 - a. The start-up and shutdown procedures
 - b. Liquid removal system (if installed)
 - c. Alarm settings
 - d. Trip settings
 - e. Flow instruments
5. *Plant specifications, including tubing vs. piping, pipe sizing, logic system, and wiring diagrams.* The plant specifications may differ from the supplier's recommendations in some applications. For example, the plant specifications may require tubing instead of piping, or different welding procedures. The supplier may recommend a trip on a specific setting. However, the plant may decide to have a coordinated trip to avoid process upset.

29.4 Magnetic Bearings

29.4.1 Operating Principles

Magnetic bearings have substituted radial and axial bearings in many applications. Each magnetic bearing (Fig. 29.8) includes the following:

- A rotor and stator
- Position sensors
- An electronic control system

The rotor of a radial magnetic bearing is made from a stack of circular laminations. These laminations are pressed onto a sleeve. This sleeve is fitted on a shaft.

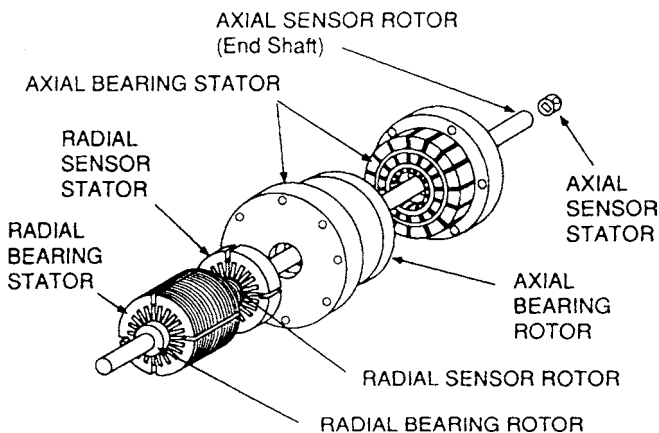


FIGURE 29.8 Magnetic bearing construction. (Source: Revolve Technologies, Calgary, Alberta, Canada.)

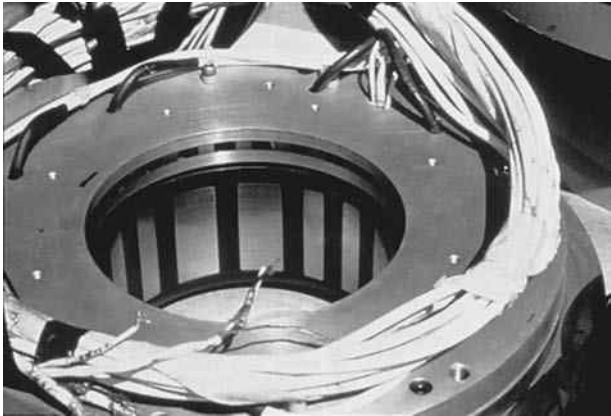


FIGURE 29.9 Radial magnetic bearing stator. (Source: Revolve Technologies, Calgary, Alberta, Canada.)

The laminations reduce the eddy-current losses. They are made from a material that has a high magnetic permeability. This is required to achieve higher flux densities. The stator of a radial magnetic bearing (Fig. 29.9) consists of the following two:

1. A stack of slotted laminations
2. Coils wound about the laminations

This stator is divided into four separate electromagnetic quadrants. Each quadrant has pairs of north and south poles. The quadrant centerlines are oriented 45° to the vertical in horizontal rotors.

The rotor of an axial magnetic bearing consists of a solid ferromagnetic disk. This disk is attached to the shaft. The stator of an axial bearing (Fig. 29.10) is made from solid



FIGURE 29.10 Axial magnetic bearing stator. (Source: Revolve Technologies, Calgary, Alberta, Canada.)

steel wedges. Coils are wound in annular grooves within these wedges. This is done to form electromagnetic windings. Laminations are placed between the wedges. These laminations are made from highly permeable material. The laminations reduce eddy-current losses. The double-acting thrust bearing is created by positioning a stator on both sides of the rotor disk.

The position sensors measure the exact position of the rotor. They provide a feedback about this position to the control system. The most reliable sensors form an inductive bridge. (Refer to *Electrical Equipment Handbook* by Philip Kiamah for the definition of inductance.) The inductance varies when the air gap varies. The control system position error signal is zero when the bearing rotor is centered. The inductance varies due to changes in the shaft position. This varies the control system position error signal.

The control system compares the position signal from the sensors and the reference signal (desired position). The error is the difference between these two signals. This error is processed by the control system. The output signal from the control system varies the current supplied to the appropriate electromagnet (Fig. 29.11). The control system has also a monitoring and security system. This system provides the following functions:

- Initiates alarms
- Trips the unit to protect the bearing from damage

A battery backup system is also provided. This system maintains operation when the normal power supply fails. The surfaces of the bearing rotor and stator are ground smooth. This is done to minimize mechanical run-out and variation in forces. For this reason, the surface of the bearing rotor should not contact that of the stator. The separation between these surfaces should be maintained during operation and any other time. An auxiliary landing system is provided to prevent contact between these surfaces. This system consists of rolling element bearings. These units are located in a removable

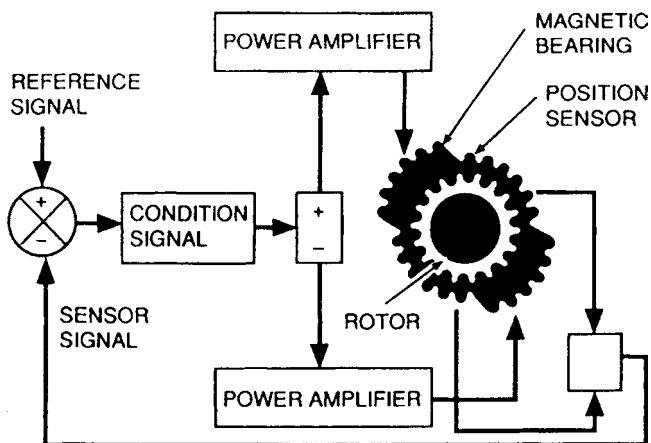


FIGURE 29.11 Control loop for magnetic bearings. (Source: Revolve Technologies, Calgary, Alberta, Canada.)

bearing holder. The clearance between the shaft and auxiliary bearings is around half the clearance between the surface of the magnet rotor and that of the stator. The auxiliary bearings support the shaft when the system is de-energized. These bearings are normally rated for three emergency coast downs from full load and speed. Purge air is also supplied to the bearing cavities. This air is normally provided from the instrument air system at the plant. The start-up procedure involves the following five steps:

1. Supplying air to the bearing cavities
2. Static levitation (rise and float in air) of the shaft
3. Pressurization of the casing
4. Opening the unit valves that isolate the compressor from the piping system
5. Beginning the driver start sequence

The shutdown procedure involves the following three steps:

1. The unit valves are closed when the shaft stops rotating.
2. The casing is left pressurized.
3. The shaft is delevitated after a predetermined period of time.

29.4.2 Operating Experience and Benefits

Magnetic bearings have been used in a wide range of applications. The benefits of these bearing systems include increased efficiency. This is due to the elimination of the parasitic shear losses associated with the oil. The power consumption of a magnetic bearing system is around 3.5 kW (5 hp). This is the power consumed by the bearing coil windings, amplifiers, and control system. On a power turbine or compressor with two magnetic bearing systems, the power saving is around 3% of the output power.

The second benefit is the enhancement in the safety of the installations. This is due to the elimination of the oil system (an oil leak can cause a fire). Insurance company statistics indicate that 80% of all rotating equipment fires are oil-related.

Magnetic bearing systems also improve the operating safety and reliability of the machine. This is due to the establishment of alarms and trips based on the signals (current and position) received from these bearing systems. Magnetic bearing systems have the ability to detect the following conditions:

- Improperly installed or damaged inlet and exit guide vanes (these vanes are used in axial-flow compressors)
- Compressor balance line blockage
- Unbalance, etc.

These conditions remain hidden with hydrodynamic bearings. This results in higher loads and shorter bearing life. The weight and space requirements of magnetic bearing systems are significantly lower than those of hydrodynamic bearings. This feature is of great importance in offshore applications.

29.4.3 Problems and Solutions

Most of the problems encountered with magnetic bearings were more related to the manufacturing procedures than the technology itself. The majority of these problems were identified and corrected.

29.4.4 Development Efforts

Development in the following areas is required to improve magnetic bearing technology:

- Auxiliary landing systems
- Higher permeability of materials
- Sensor developments
- Standardized electronic capabilities
- Software tuning capabilities

The original landing systems for magnetic bearings included only rolling element bearings. However, these bearings were not specifically designed for this application. Their life span is limited in such demanding service. Alternative systems were developed for this application. These systems do not rely on rolling elements. They rely on passive friction between the rotor and the stator.

Encouraging results were obtained when materials were sized to allow dissipation of the heat generated by friction from the shaft. However, these materials were not used with heavy rotors yet. Nevertheless, with further developments, such a landing system could be implemented successfully.

The size of magnetic bearings can be reduced by making the laminations from a material that has higher magnetic permeability. This feature provides an advantage in areas where there are space limitations.

29.5 Thrust-Reducing Seals

Thrust-reducing seals have been used for both overhung and beam-type machines. Overhung units experience normally a much higher thrust loading during the start-up than beam-type machines. This is because their pressure forces are not balanced. The size of the axial magnetic bearing required for an overhung unit would be prohibitive if thrust-reducing seals were not used. These seals allowed the reduction of the bearing to a size that can easily be incorporated into the housing.

Figure 29.12 illustrates a compressor that was selected for this application. This unit had originally a radial inlet design. It was retrofitted to an axial inlet configuration. Thrust-reducing seal design involves the usage of a tube. This tube encloses a volume projecting from the eye of the impeller. It is almost equivalent to the cross section of the shaft. A dry seal is placed at the eye of the impeller. It is used to isolate the volume enclosed by the tube from the compressor cavity (Fig. 29.13).

The tube is at atmospheric pressure at the beginning of the start-up procedure. A net thrust is developed across the impeller as the head increases. This thrust is in the direction of the compressor suction. This generates a load in the Z2 axial magnetic stator

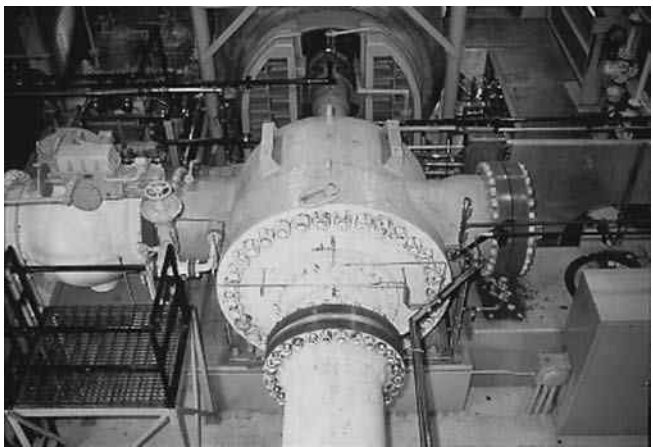


FIGURE 29.12 Axial inlet compressor being retrofitted with magnetic bearings. (Source: Revolve Technologies, Calgary, Alberta, Canada.)

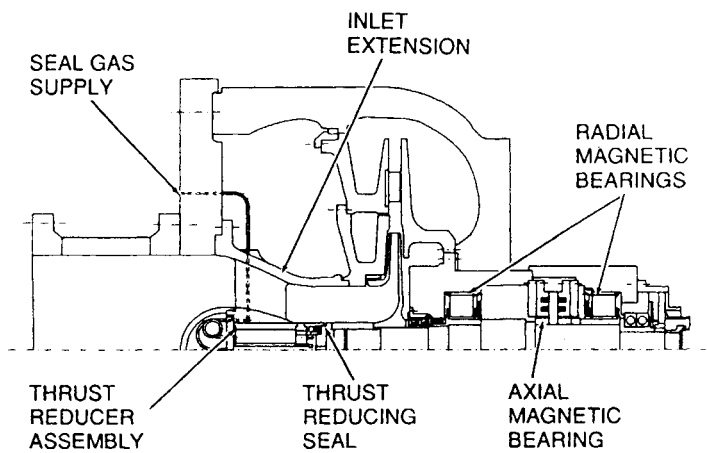


FIGURE 29.13 Axial inlet thrust reducer. (Source: Revolve Technologies, Calgary, Alberta, Canada.)

(Fig. 29.14). The current sent from the controller increases as a result of the increase in the load. This current is processed through the electro-pneumatic (I/P) transducer. This allows gas to be discharged from the seal supply gas into the tube. The pressure at the suction of the impeller will increase due to the flow of gas. This counteracts the thrust load on the shaft, and reduces the axial bearing current. The increase in pressure at the suction of the impeller increases until equilibrium is reached.

Figure 29.15 illustrates a thrust-reducing seal installed on a beam-type compressor. The operation of this seal is similar to one used with overhung compressors. This design is capable of handling a range of axial loads.

Axial loads are normally counteracted by a balance piston or a balance drum. This design consists of a cylinder fitted onto the shaft on the discharge side of the impeller.

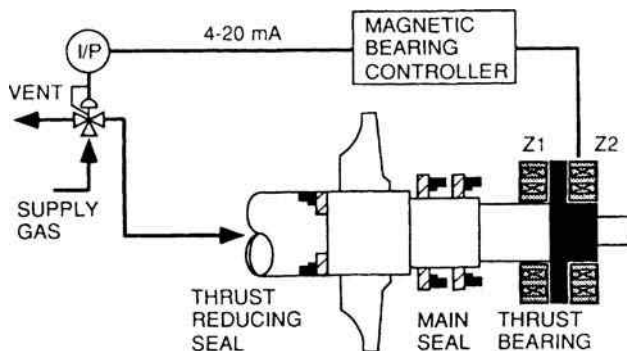


FIGURE 29.14 Control for an overhung thrust reducer. (Source: Revolve Technologies, Calgary, Alberta, Canada.)

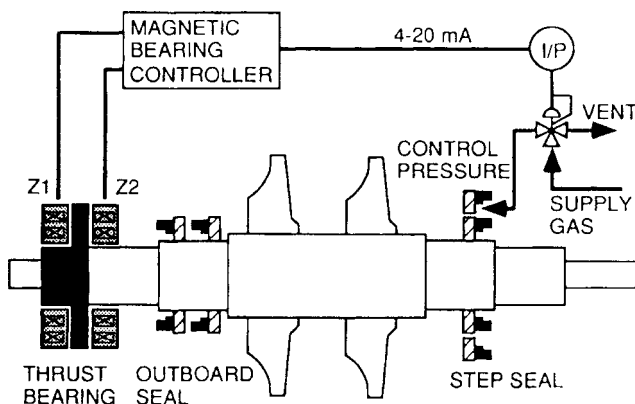


FIGURE 29.15 Control for a beam-type thrust reducer. (Source: Revolve Technologies, Calgary, Alberta, Canada.)

The net thrust is created by the differential pressure across the impeller. This thrust is balanced by a thrust in the opposite direction created across the cylinder. This is done by installing a labyrinth seal on the outside diameter of the cylinder. A gas stream flows from the discharge side of the impeller to the suction side of the compressor through a *balance line*. This flow is known as balance piston leakage. It can reach several percent of the flow through the compressor. This decreases the overall efficiency of the compressor. Thrust-unloading seals eliminate this leakage. Thus, they increase the overall efficiency of the unit.

29.6 Integrated Design

Magnetic bearing technology has provided information that was previously unknown. This includes the effect of internal aerodynamic design on bearing loads. A predictive model for bearing loads could be established by combining the operational knowledge obtained to date with modern methods of numerical analysis.

The retrofit of magnetic bearings has proved the practicality and robustness of this technology. However, further improvements in operational efficiency and serviceability can be obtained from this technology. For example, the advantage of magnetic bearings has been demonstrated in applications involving thrust-reducing seals.

Dry seal and magnetic bearing technologies provide significant advantages over conventional systems. Most of the problems encountered with the implementation of these technologies have been solved. These problems were not related to the technologies themselves in most cases. The incorporation of these technologies in the design stage of turbo-machinery has led to advantageous designs. These designs include the following:

- Sulzer motor pipeline compressor (Mopico) shown in Figs. 29.16 and 29.17
- Sulzer-Acec high-speed oil-free intelligent motor (Hofim) compressor shown in Fig. 29.18

The Mopico gas pipeline compressor includes a high-speed, two-pole squirrel-cage induction motor. The motor and compressor of this machine are housed in a hermetically sealed vertically split casing. This casing is made from forged steel. The motor and bearings are located in the center of the casing. Each of the end casing sections houses the following:

- A compressor wheel
- A fixed-vane diffuser
- Inlet and discharge flanges

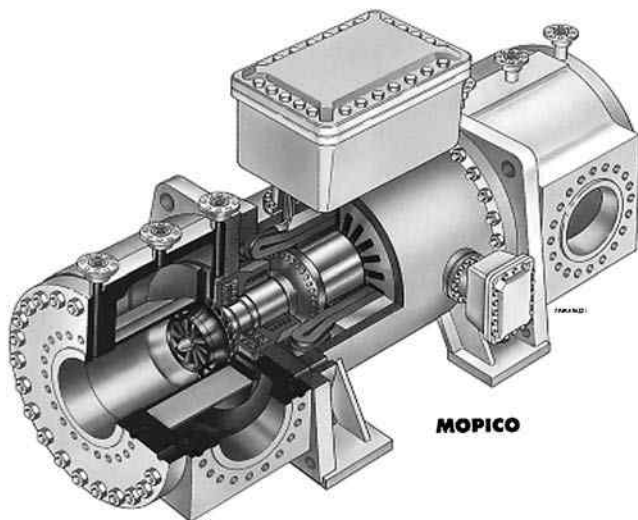
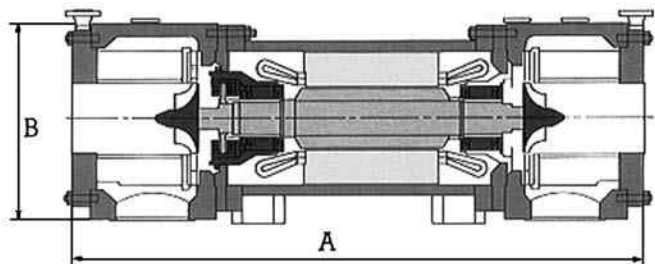


FIGURE 29.16 Sulzer Mopico motor pipeline compressor, incorporating magnetic bearings.
(Source: MAN Diesel & Turbo SE.)



Dimensions and weights

Frame size	Power kW	A mm	B mm	Weight kg
RM 28	2000	2500	900	9 000
RM 35	3500	2900	1000	11 000
RM 40	6000	3300	1200	13 000

FIGURE 29.17 Dimensions, weight, and simplified cross section of a Mopico compressor. (Source: MAN Diesel & Turbo SE.)

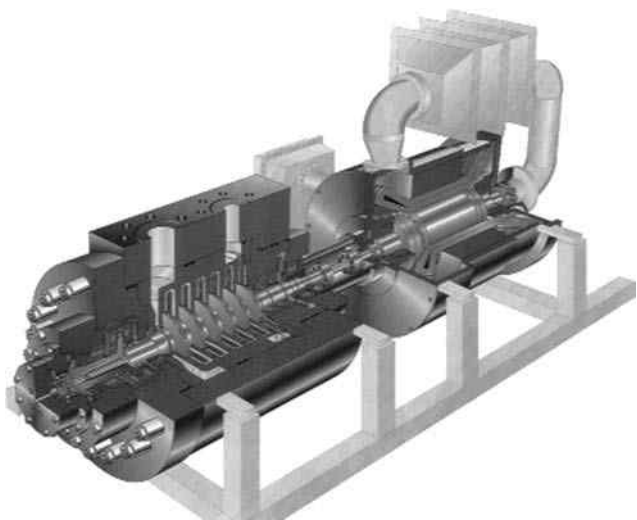


FIGURE 29.18 Hofim high-speed oil-free intelligent motor compressor. (Source: MAN Diesel & Turbo SE.)

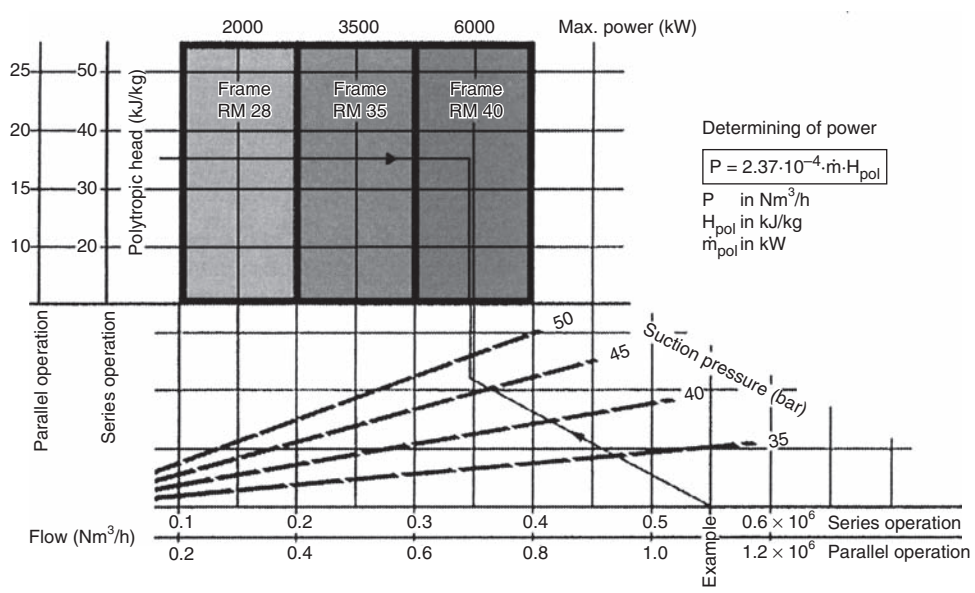


FIGURE 29.19 Application ranges for Mopico compressors. (Source: MAN Diesel & Turbo SE.)

These units can be operated in series or parallel (Fig. 29.19). Magnetic radical bearings and a double-acting magnetic thrust bearing are used in this design. The thrust bearing maintains the rotor position. The motor is cooled by gas. The flow of this gas is metered from the high-pressure plenum of one of the compressor housings. Thus, the Mopico operates oil-free.

A thyristored variable-frequency drive controls the speed of the rotor. This controls the discharge rate of the Mopico unit. The variable frequency provides the following:

- Variation of the rotor speed between 70% and 105% of the speed range.
- Allows the unit to start without requiring inrush current. (Peak current equal to 6 to 8 times the normal operating current. This current is required when the motor is started. Refer to the *Electrical Equipment Handbook* by P. Kiameh for details.)

Figure 29.20 illustrates the overall installation schematic for a Mopico compressor. This design provides the following advantages:

- Low installation, maintenance, and power costs
- Operation over a broad range of flow and pressure at high economic performance
- Compatibility with various compressors
- Unattended operation using remote control
- Emissionless and oil-free operation
- Possibility of outdoor installation of the equipment

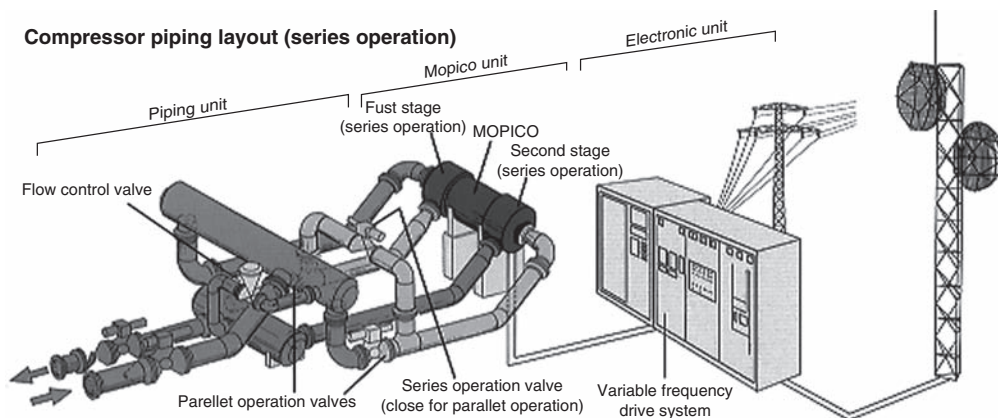


FIGURE 29.20 Installation schematic for a Mopico compressor. (Source: MAN Diesel & Turbo SE.)

The cost per installed horsepower of a Mopico compressor is as follows:

- About two-thirds that of a gas turbine unit
- Less than half that of a low-speed reciprocating compressor
- About 10% less than that of a conventional centrifugal compressor with dry seals, magnetic bearings, and a direct-drive high-speed induction motor with variable-frequency drive

The high-speed oil-free intelligent motor compressor shown in Fig. 29.18 includes a separate motor couple directly to a compressor. Both machines operate with magnetic bearings. This prototype application is in a natural gas storage facility. The following are the key parameters of this installation:

Motor nominal speed	20,000 rpm
Motor nominal power	2000 kW
Compressor inlet pressure	5000 kPa
Compressor discharge pressure	15,500 kPa
Compressor speed range	70–102%
Compressor flow	38,000 N · m ³ /h

The motor is driven by a solid-state variable-frequency drive. It is an asynchronous squirrel-cage induction machine. The compressor has six stages. It is a barrel machine of fully modular design. Dry gas seals are used in this application. They minimize internal and external leakage. The residual thrust of the rotor is controlled by an active balance system. This system limits the axial thrust to a level compatible with the rating of the axial magnetic bearing.

29.7 Bibliography

- Bloch, H. P., *A Practical Guide to Compressor Technology*, 2nd ed., John Wiley & Sons Inc., Hoboken, New Jersey, 2006.
- Kiameh, P., *Electrical Equipment Handbook: Troubleshooting and Maintenance*, McGraw-Hill, New York, 2003.

This page intentionally left blank

CHAPTER 30

Compressor System Calculations

30.1 Calculations of Air Leaks from Compressed-Air Systems

Figures 30.1 and 30.2 illustrate a typical industrial compressed-air system. Calculate the leakage rate from the system if the piping has a hole of 0.1 in (0.25 cm) diameter.

Assume the following:

The air pressure in the pipe is 8 psi (55.12 kPa); the atmosphere is at sea level, the plant operates 7800 h/yr, the air temperature is 19.4°C (67°F). Calculate the annual cost of air leakage if the cost of compressed air is \$1.85 per 1000 ft³ (28.m³).

Solution

Air reaches a critical pressure of $0.53 \times$ (inlet pressure) when it flows through an orifice or nozzle. This pressure is reached at the vena contracta which is the minimum area of the flow. It occurs slightly downstream of the orifice or nozzle. If the critical pressure calculated by the above equation is lower than the outlet or back pressure, then the pressure at the vena contracta will be equal to the back pressure. The characteristics of a flow through an orifice or nozzle are identical to the ones of a flow through a hole in a pipe.

The absolute pressure of the air in the pipe is

$$P_{\text{abs}} = 14.7 + 8 = 22.7 \text{ psi (abs)} [155.97 \text{ kPa (abs)}]$$

The calculated critical pressure in this case is given by

$$P_{\text{critical}} = 0.53 \times 22.7 = 12.03 \text{ psi (abs)} [82.66 \text{ kPa (abs)}]$$

Since this pressure is lower than the atmospheric back pressure [14.7 psi (abs), 101.3 kPa (abs)], the pressure at the vena contracta would be equal to the back pressure [14.7 psi (abs), 101.3 kPa (abs)]. The mass flow of air leakage is given by:

$$W = 1.06 A [P_1(P - P_1)/T]^{0.5}$$

where W = the leakage flow rate, lb/s (kg/s)

A = hole area, in² (cm²)

P = air pressure in the pipeline, psi (abs) [kPa (abs)]

P_1 = outlet or back pressure, psi (abs) [kPa (abs)]

T = absolute temperature of air in the pipe = °F + 460 (°R) or °C + 273 (K)

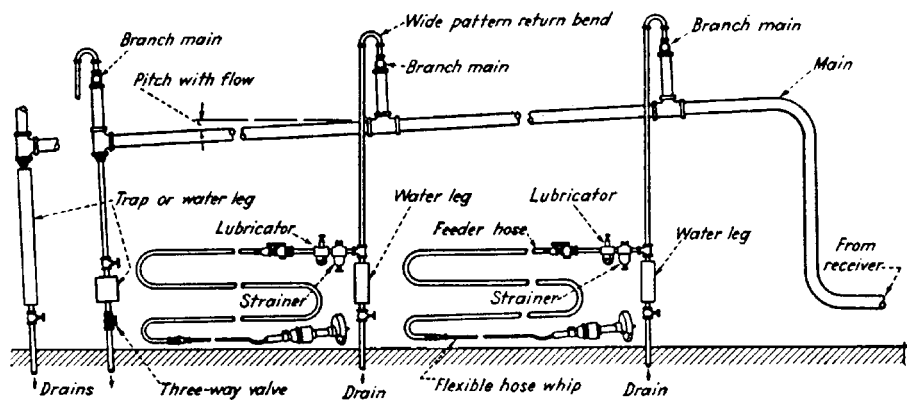


FIGURE 30.1 Typical compressed-air system main and branch pipes (factory management and maintenance).

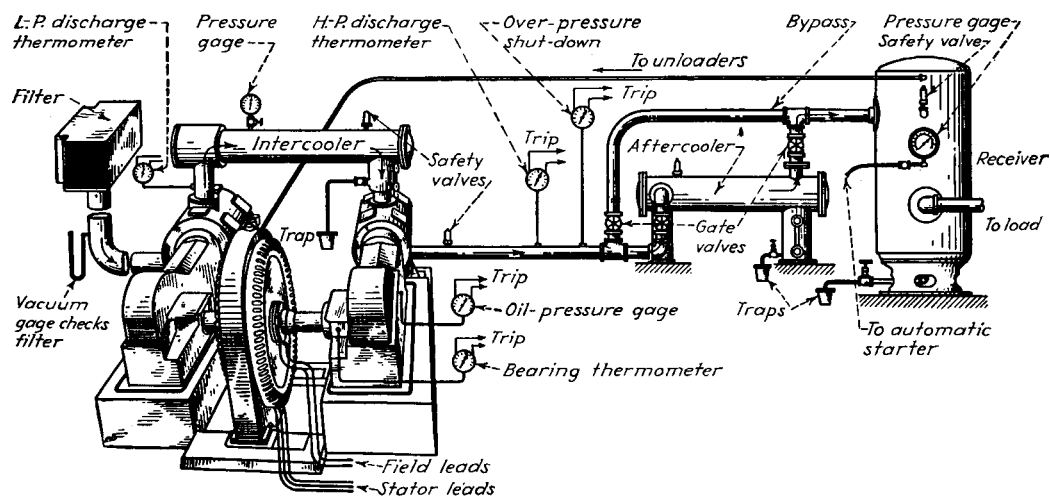


FIGURE 30.2 Typical compressed-air plant showing compressor and its associated piping and accessories. (Source: Reprinted with permission from *Power magazine*.)

Thus,

$$W = 1.06 \times 0.00785 [14.7(22.7 - 14.7)/527]^{0.5} = 0.00393 \text{ lb/s (0.00179 kg/s)}$$

The leakage flow rate per hour is $0.00393 \times 3600 = 14.15 \text{ lb/h (6.43 kg/h)}$.

30.1.1 Annual Cost of Air Leakage

Since the weight of air at 14.7 psi (abs) [101.3 kPa (abs)] is 0.075 lb/ft³ (1.2 kg/m³), the volumetric leakage rate is

$$14.15 / 0.075 = 188.67 \text{ ft}^3/\text{h (5.36 m}^3/\text{h)}$$

The annual cost of air leakage is

$$\begin{aligned} & (\text{volumetric leakage rate}) \times (\text{operating hours per year}) \times (\text{cost of air leakage}) \\ & = 188.67 \text{ ft}^3/\text{h} \times 7800 \text{ h} \times 1.85/1000 (\$/\text{ft}^3) = \$2722.5 \end{aligned}$$

The cost of air leaks would be very significant if the system has many leaks.

Calculate the cost of air leakage if the pressure in the pipe is 60 psi (412.24 kPa) and all the remaining variables are the same as the example above. The critical pressure in this case is

$$0.53 (60 \text{ psi}) + 14.7 = 46.5 \text{ psi (abs)} [319.49 \text{ kPa (abs)}]$$

Since the critical pressure in this case is greater than the atmospheric back pressure of 14.7 psi (abs) [101.3 kPa (abs)], the mass flow of air leakage is given by

$$W = 0.5303 (ACP)/(T)^{0.5}$$

where, C = flow coefficient = 1.0. The remaining variables are the same as the above equation. Thus,

$$W = 0.5303 (0.00785)(1.0)(60 + 14.7)/(527)^{0.5} = 0.0255 \text{ lb/s (0.0116 kg/s)}$$

The hourly mass flow rate is

$$0.0255 \times 3600 = 91.8 \text{ lb/h (41.73 kg/h)}$$

The annual leakage cost is

$$(91.8 \text{ lb/h})(7800 \text{ h/yr})(1.85/1000 \text{ \$/ft}^3)(0.075 \text{ lb/ft}^3) = \$17,662.3$$

This result proves that the losses are more significant at higher pressures. The emissions from the power generating plant that supplied the energy to compress the air in the system would also increase due to this leak.

30.2 Centrifugal Compressor Power Requirement

Calculate the horsepower requirement of a centrifugal compressor having the following parameters:

Air flow = 24,000 ft³/min (679.2 m³/min)

Inlet pressure = 14 psi (abs) [96.46 kPa (abs)]

Inlet temperature = 15.6°C (60°F)

Outlet pressure = 100 psi (abs) [687.07 kPa (abs)]

Outlet temperature = 107.2°C (225°F)

Area of suction pipe = 3.5 ft² (0.325 m²)

Area of discharge pipe = 0.75 ft² (0.0696 m²)

The discharge pipe is located 30 ft (9.14 m) above the suction pipe.

Inlet temperature of Jacket water = 18.33°C (65°F)

Outlet temperature of jacket water = 48.89°C (120°F)

Mass flow of jacket water = 1009 lb/min (458.64 kg/min)

The horsepower requirement of any centrifugal compressor is given by

$$\text{hp} = \frac{w}{0.707} \left[c_p(t_2 - t_1) + \frac{V_2^2 - V_1^2}{50,000} + \frac{Z_2 - Z_1}{778} \right] + \left[\frac{w_j(t_0 - t_i) + R_c}{0.707} \right]$$

where W = weight in lb (kg) of unit flow rate, ft³/s (m³/s) through the compressor

$$W = (P_1)(V_1)/R(T_1)$$

$$P_1 = \text{inlet pressure, psi (abs) [kPa (abs)]}$$

$$V_1 = \text{inlet volume flow rate, ft}^3/\text{s (m}^3/\text{s)}$$

$$R = \text{gas constant for air} = 53.3 \text{ (British units psi, ft}^3/\text{s, }^\circ\text{R)}$$

$$T_1 = \text{inlet air temperature (}^\circ\text{R) or (K)}$$

The inlet flow rate is

$$24,000 \text{ ft}^3/\text{min} = 400 \text{ ft}^3/\text{s (11.33 m}^3/\text{s)}$$

$$P_1 = 14 \text{ psi (abs) [96.46 kPa (abs)]}$$

$$T_1 = 60 + 460 = 520^\circ\text{R}$$

Thus,

$$W = (14 \text{ lb/in}^2)(144 \text{ in}^2/\text{ft}^2)(400 \text{ ft}^3/\text{s})/53.3(520) = 29.1 \text{ lb (13.22 kg)}$$

$$c_p = \text{specific heat of air at inlet temperature} = 0.24 \text{ Btu/lb}^\circ\text{F (1004.2 J/kg K)}$$

$$t_2 = \text{outlet temperature} = 225 + 460 = 685^\circ\text{R}$$

$$t_1 = \text{inlet temperature} = 60 + 460 = 520^\circ\text{R}$$

$$V_1 = \text{air velocity at compressor entrance ft/min (m/min)}$$

$$V_2 = \text{air velocity at compressor discharge ft/min (m/min)}$$

$$Z_1 = \text{elevation of inlet pipe to compressor above a selected datum ft (m)}$$

$$Z_2 = \text{elevation of compressor outlet pipe above the same datum ft (m)}$$

$$W_j = \text{weight of water flowing in the compressor cooling jacket, lb/s (kg/s)}$$

$$t_i = \text{water inlet temperature to the cooling jacket}$$

$$t_o = \text{water outlet temperature from the cooling jacket}$$

$$V_1 = (\text{flow rate, ft}^3/\text{s})/(\text{inlet area, ft}^2)$$

$$V_1 = (400 \text{ ft}^3/\text{s})/(3.5 \text{ ft}^2) = 114.29 \text{ ft/s (34.83 m/s)}$$

$$V_2 = (400 \text{ ft}^3/\text{s})/(0.75 \text{ ft}^2) = 533.33 \text{ ft/s (162.56 m/s)}$$

$$R_c = 0 \text{ based on the assumption that there are no losses from radiation heat transfer}$$

$$\begin{aligned} \text{hp} &= \frac{29.1}{0.707} \left[0.24(685 - 520) + \frac{533.33^2 - 114.29^2}{50,000} + \frac{30}{778} \right] \\ &\quad + [10,009/60 \times (120 - 65)]/0.707 \\ &= 41.16(39.6 + 5.43 + 0.038) + 1308.23 \\ &= 1854.99 + 1308.23 = 3163.23 \text{ hp} \\ \text{hp} &= 3163.23 \text{ hp (2359.77 kW)} \end{aligned}$$

30.2.1 Compressor Selection

Select a compressor for an air system in an industrial plant having the tools listed in Table 30.1. The plant is located at 2000 ft above the sea. It operates 18 h/day.

Tool	(1) Air Consumption ft ³ /min	(1) Air Consumption ft ³ /min	(2) Number of Tools	(3) Air Required (1) × (2) ft ³ /min	(3) Air Required (1) × (2) m ³ /min	(4) Load Factor	(5) Air Demand (3) × (4) ft ³ /min	(5) Air Demand (3) × (4) m ³ /min
Steel drills, ½ in (1.27 cm), 91b (4. 1 kg)	70	2.27	4	280	9.08	0.3	84	2.72
Grinders 6 in (15.2 cm) diameter wheels	50	1.4	3	150	4.2	0.4	60	1.68
Plug drills	50	1.4	4	200	5.6	0.3	60	1.68
Rotary sanders 9 in (22.9 cm) diameter pads	55	1.56	2	110	3.12	0.4	44	1.25
Paint spray	20	0.57	3	60	1.71	0.3	18	0.51
Total							266	7.84

TABLE 30.1 Calculations of Air System Requirements

The selection process of a compressor for a specific application should start by completing the compressed-gas requirements table (similar to Table 30.3) for the application. Table 30.2 can be used as a guide to determine the air requirements of pneumatic tools. Columns (3) of Table 30.1 is equal to the product of column (1) and column (2).

The load factor of the equipment is given by:

Load factor =
$$\frac{\text{actual air consumption of tool, ft}^3/\text{min}}{\text{full-load continuous air consumption of tools, ft}^3/\text{min}}$$

The air demand [column (5)] is the product of column (3) and column (4). A leakage factor should be applied to the system. A 10% leakage is normally assumed for most systems.

	ft ³ /min	m ³ /min
Grinders:		
6- and 8-in (15.2- and 20.3-cm) diameter wheels	50	1.4
2- and 2½-in (15.2- and 6.4-cm) diameter wheels	14–20	0.40–0.57
File and burr machines	18	0.51
Rotary sanders, 9-in (22.9-cm) diameter pads	55	1.56
Sand rammers and tempers:		
1 × 4 in (2.5 × 10.2 cm) cylinder	25	0.71
1¼ × 5 in (3.2 × 12.7 cm) cylinder	28	0.79
1½ × 6 in (3.8 × 15.2 cm) cylinder	39	1.1
Chipping hammers:		
10 to 13 lb (4.5 to 5.9 kg)	28–30	0.79–0.85
2 to 4 lb (0.9 to 1.8 kg)	12	0.34
Nut setters:		
To 5/16 in, 8 lb (0.79 cm, 3.6 kg)	20	0.57
½ to ¾ in, 18 lb (1.3 to 1.9 cm, 8.1 kg)	30	0.85
Paint spray	2–20	0.06–0.57
Plug drills	40–50	1.1–1.4
Riveters:		
3/32 to 1/8-in (0.24- to 0.32-cm) rivets	12	0.34
Larger, weighing 18 to 22 lb (8.1 to 9.9 kg)	35	0.99
Rivet busters	35–39	0.51–0.75
Steel drills, rotary motors:		
To ¼ in (0.64 cm) weighing 1¼ to 4 lb (0.56 to 1.8 kg)	18–20	0.57–1.1
¼ to 3/8 in (0.69 to 0.95 cm) weighing 6 to 8 lb (2.7 to 3.6 kg)	20–40	1.98
½ to ¾ in (1.27 to 1.91 cm) weighing 9 to 14 lb (4.1 to 6.3 kg)	70	2.27
7/8 to 1 in (2.2 to 2.5 cm) weighing 25 lb (11.25 kg)	80	1.1
Wood borers to 1-in (2.5 cm) diameter, weighing 14 lb (6.3 kg)	40	

TABLE 30.2 Approximate Air Needs of Pneumatic Tools

Thus,

$$\text{the required air capacity} = 1.1 \times 266 = 292.6 \text{ ft}^3/\text{min} (8.62 \text{ m}^3/\text{min})$$

The future requirements of the system should also be estimated. A table similar to Table 30.1 should be prepared listing the predicted equipment and their air requirements. In this case, it is assumed that the future air requirements are $120 \text{ ft}^3/\text{min}$ ($3.4 \text{ m}^3/\text{min}$).

Therefore, the total air capacity is

$$292.6 + 120 = 412.6 \text{ ft}^3/\text{min} (12.02 \text{ m}^3/\text{min})$$

Most air equipment requires 90 psi (620 kPa) at their inlet. Thus, industrial compressors are normally rated for 100 psi (689 kPa) to allow for the pressure drop between the compressor and the equipment. The compressor capacity should include the future air requirements if the equipment will be added within the next few years.

Table 30.3 lists the power required by air compressor operating at different discharge pressures and altitudes above the sea. Clearly, the two-stage compressor consume less power than single-stage compressors when the discharge pressure is 100 psi (689 kPa) and altitude 2000 ft (610 m). The difference between the power required by a two-stage compressor and a single-stage compressor in this case is: $21.3 - 18.4 = 2.9 \text{ hp}$ (2.2 kW).

Thus, a two-stage compressor will be selected, its power requirement is:

$$(412.6/100) (18.4) = 75.92 \text{ hp} (56.64 \text{ kW})$$

Reciprocating compressors are most suitable for plant air systems. Table 30.4 shows the characteristics of the various types of reciprocating compressors.

A two-stage double-acting water-cooled compressor is selected based on Table 30.4 due to the requirement of 18 h/day of operation.

Rotary compressors are not normally selected for industrial air systems because their discharge pressure are usually lower than 100 psi (68.9 kPa), except when they are multistage units. Centrifugal compressors are normally used for flows exceeding several thousand ft^3/min . Thus, they are normally used in applications such as steel-mill blowing*, copper conversion, and small gas turbines (<3 MW). Axial flow compressors are normally used for applications having higher flows than centrifugal compressors (exceeding $10,000 \text{ ft}^3/\text{min}$) such as intermediate (>3 MW) and large gas turbines.

30.2.2 Selection of Compressor Drive

Compressor drives include ac and dc motors, gas turbines, steam turbines, gasoline engines, and diesel engines. However, the most common drive is ac squirrel-cage induction motors. Synchronous motors are used for applications requiring power factor correction. Cylinder unloaders are used to change the gas flow delivered by the compressor to the receiver.

**Blowers* are machines having a discharge pressure lower than 35 psi (241 kPa). *Compressors* are machines having a discharge pressure exceeding 35 psi (241 kPa).

Altitude, ft (m)	Single- Stage Discharge Pressure, lb/in ² gauge (kPa)			Two-Stage Discharge Pressure, lb/in ² gauge (kPa)		
	60 (414)	80 (552)	100 (689)	60 (414)	80 (552)	100 (689)
0 (0)	16.3 (12.2)	19.5 (14.6)	22.1 (16.5)	14.7 (10.9)	17.1 (12.8)	19.1 (14.3)
2000 (610)	15.9 (11.9)	18.9 (14.1)	21.3 (15.9)	14.3 (10.7)	16.5 (12.3)	18.4 (13.7)
4000 (1212)	15.4 (11.5)	18.2 (13.6)	20.6 (15.4)	13.8 (10.3)	15.8 (11.8)	17.7 (13.2)
6000 (1820)	15.0 (11.2)	17.6 (13.1)	20.0 (14.9)	13.3 (9.9)	15.2 (11.3)	17.0 (12.7)

*Courtesy of Ingersoll-Rand. Values shown are the approximate bhp input required per 100 ft³/min (2.8 m³/min) of free air actually delivered. The bhp input can vary considerably with the type and size of compressor.

TABLE 30.3 Air Compressor Brake Horsepower (kW) Input*

Compressor Type	Up to hp (kW)	Up to Pressure psi (kPa)	Type of Service
Single-stage air-cooled	3 (2.2 kW)	150 (1034 kPa)	Light and intermittent operation up to 1 h/day
Two-stage air-cooled	3 (2.2 kW)	150 (1034 kPa)	4 to 8 h/day of operation
Single-stage air-cooled	15 (11.2 kW)	80 (552 kPa)*	up to 24 h/day
Single-stage horizontal double-acting water-cooled	10–100 (7.5–75 kW)	100 (689 kPa)	up to 24 h/day
Two-stage single-acting air-cooled	10–100 (7.5–75 kW)	Higher than 80 (552 kPa)	5–10 h/day
Two-stage double-acting water-cooled	Higher than 100 (75 kW)	Higher than 100 (689 kPa)	up to 24 h/day

*Two-stage air-cooled compressors should be used if pressure higher than 80 psi (552 kPa) is required.

TABLE 30.4 Characteristics of Reciprocating Compressors

The power requirement of a large reciprocating compressor is 22 hp (16.4 kW) per 100 ft³/min (2.8 m³/min) of income *free air*. It should be noted that the term “free air” refers to air at the inlet conditions to the compressor. It does not refer to air at standard conditions. Since these conditions vary with altitude, barometric pressure, air temperature, and air compressors are rated by their free air capacity. The compressor “displacement” is the volumetric flow through the compressor based on inlet conditions in ft³/min.

30.2.3 Selection of Air Distribution System

Figure 30.3 illustrates a *central* air distribution system used in industrial plants. It consists of one or more centrally located compressors and supply piping to all the areas within the plant requiring air. This design is used in application having relatively short distribution system to minimize the pressure drop.

Figure 30.4 illustrates a *unit* air distribution system. It consists of several smaller compressors and a small distribution system associated with each compressor. Emergency connections between the various stations are made in some applications. This design is selected for plants where the equipment requiring air is far apart.

30.2.4 Water Cooling Requirements for Compressors

Air-cooled compressors are normally less expensive than water-cooled compressors. However, the power requirement of a water-cooled compressor is lower than an air-cooled compressor having the same rating. In general, two-stage compressors with intercoolers are more economical for applications requiring more than 4 hours of operation per day. Table 30.5 shows the cooling water recommended for various compressor systems.

30.2.5 Variation of Compressor Delivery with Inlet Air Temperature

The compressor delivery decreases when the inlet air temperature increases. Table 30.6 shows the relative delivery of air compressors based on nominal intake temperature of 15.6°C (60°F).

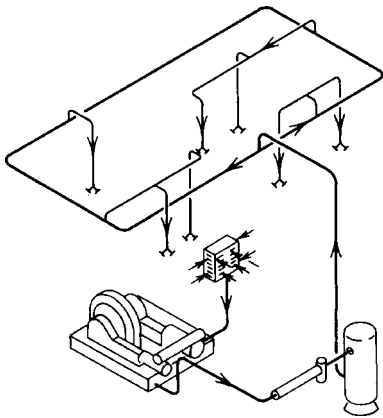


FIGURE 30.3 Central system for compressed air supply. (Source: Reprinted from Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006 with permission from McGraw-Hill.)

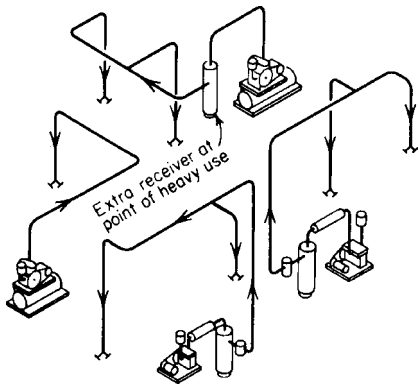


FIGURE 30.4 Unit system for compressed-air supply. (Source: Reprinted from Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006 with permission from McGraw-Hill.)

Cooling Equipment	Actual Free Air, gal/min per 100 ft ³ /min (L/s per 100 m ³ /s)
Intercooler separate	2.5–2.8 (334.2–374.3)
Intercooler and jackets in series	2.5–2.8 (334.2–374.3)
Aftercoolers:	
80 to 100 lb/in ² (551.6–689.5 kPa), two-stage	1.25 (167.1)
80 to 100 lb/in ² (551.6–689.5 kPa), two-stage	1.8 (240.6)
Two-stage jackets alone (both)	0.8 (106.9)
Single-stage jackets:	
40 lb/in ² (275.8 kPa)	0.6 (80.2)
60 lb/in ² (413.7 kPa)	0.8 (106.9)
80 lb/in ² (551.6 kPa)	1.1 (147.0)
100 lb/in ² (689.5 kPa)	1.3 (173.8)

TABLE 30.5 Cooling Water Recommended for Intercoolers, Cylinder Jackets, Afercoolers

30.2.6 Sizing of Compressor System Components

Sizing of Air Receiver

Calculate the volume of the air receiver required for a compressed air system having a compressor displacement of 1100 ft³/min (0.52 m³/s) when the suction pressure is 14.7 psi (abs) [101.4 kPa (abs)] and the discharge pressure is 180 psi (abs) [1241.1 kPa (abs)].

Initial Temperature				Relative Delivery
°F	°R	°C	K	
40	500	4.4	277.4	1.040
50	510	10.0	283.0	1.020
60	520	15.6	288.6	1.000
70	530	21.1	294.1	0.980
80	540	26.7	299.7	0.961

TABLE 30.6 Effect of Initial Temperature on Delivery of Air Compressors [Based on a nominal intake temperature of 15.6°C (60°F)]

Solution

The following equation should be used to calculate the volume of the air receiver:

$$V_m = \frac{d P_1}{P_2}$$

where V_m = the minimum volume of the receiver, ft³

d = compressor displacement, ft³/min (the first-stage displacement should be used in a multistage compressor)

P_1 = compressor inlet pressure, psi (abs)

P_2 = compressor discharge pressure, psi (abs)

Thus,

$$V_m = \frac{1100(14.7)}{180} = 89.83 \text{ ft}^3 (2.5 \text{ m}^3)$$

However, the receiver should have a good reserve capacity. Therefore, a receiver having a volume of 150 or 180 ft³ (4.2 or 5.04 m³) should be selected.

30.2.7 Calculation of Receiver Pump-Up Time

Calculate the time required for the above compressor to pump-up a 250 ft³ (7.08 m³) receiver from 100 to 180 psi (689.5–1241.1 kPa) if the compressor average volumetric efficiency is 70%.

Solution

The following equation should be used to calculate the pump-up time of the receiver:

$$t = \frac{V(P_f - P_i)}{14.7 d e}$$

where t = pump-up time of receiver, min

P_f = final pressure, psi (abs)

P_i = initial receiver pressure, psi (abs)

d = compressor piston displacement, ft³/min

e = compressor volumetric efficiency, %

Thus,

$$t = \frac{250(180 - 100)}{14.7(1100)(0.7)} = 1.77 \text{ min}$$

The compressor volumetric efficiency can be taken out of the above equation if the compressor discharge flow is given in ft³/min of free air instead of the piston displacement.

30.3 Bibliography

Hicks, G. T., *Handbook of Mechanical Engineering Calculations*, McGraw-Hill, New York, 1997.

CHAPTER 31

Pumps

31.1 Introduction

Pump is an equipment used to increase the pressure of liquids. Pumps are divided into two categories:

1. Dynamic (reciprocating), in which energy is continuously added to increase the liquid velocities within the machine, thus reducing this increase (in liquid velocities) to produce a pressure increase
2. Displacement (rotary), in which energy is periodically added by application of force to one or more movable boundaries of any desired number of enclosed, liquid-containing volumes, resulting in a direct increase in pressure up to the value required to move the liquid through valves or ports into the discharge line.

Figure 31.1 indicates the ranges of pressure and capacity of the different types of pumps.

The following points should be noted:

- The very high pressures can only be reached with a reciprocating pump.
- If the liquid can be handled by any of the three basic types of pumps, the most economical type would be centrifugal, rotary, and reciprocating, in that order.
- Most pumps found in industry are of the centrifugal type.

31.2 Centrifugal Pumps

A typical centrifugal pump is shown in Fig. 31.2. To be able to determine similarities between pumps, the specific speed has been established. It is used to lump together a large number of designs into a single expression.

$$N_{sm} = \frac{N\sqrt{Q}}{H^{0.15}}$$
$$N_s = 51.65 N_{sm}$$

where N = the number of revolutions per second, rev/s
 Q = capacity, m^3/s ; H : head, m

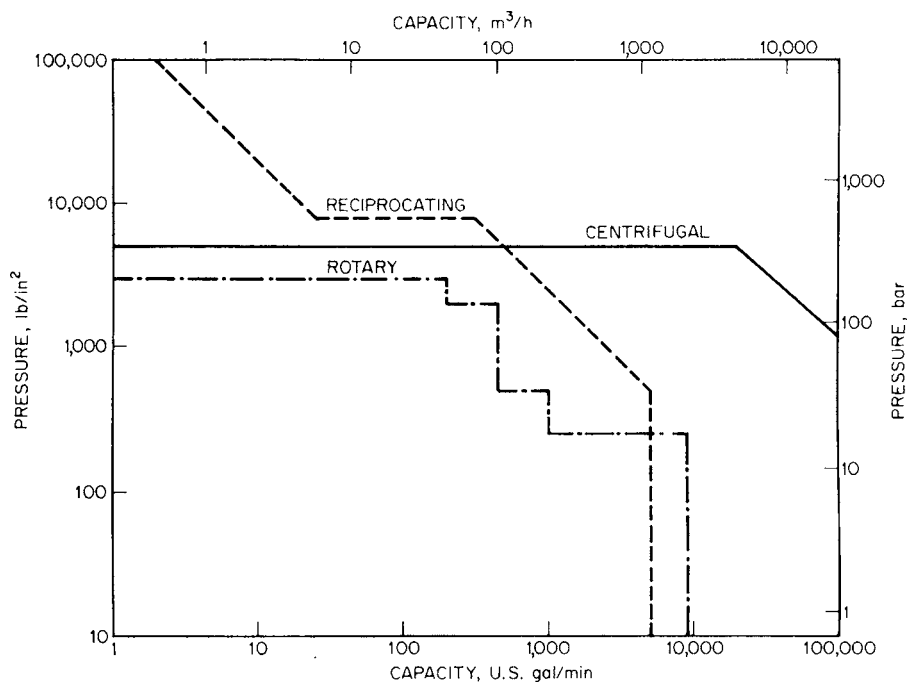


FIGURE 31.1 Approximate upper limit of pressure and capacity by pump class. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

Accepted dimensions are also (in U.S.A.)

- N = revolution per minute
- Q = gallons per minute
- H = feet

$$N_s = \frac{N\sqrt{Q}}{H^{0.15}}$$

A pump efficiency* chart is presented in Fig. 31.3. It illustrates the efficiencies of a large number of commercial centrifugal pumps versus specific speed. (The efficiency can be predicted knowing the head and capacity.)

A centrifugal pump has the two main parts:

1. A rotating element, including an impeller and a shaft
2. A stationary element made up of a casing, stuffing box, and bearings

*Efficiency = power output in the form of pressure increase/mechanical power input in the form of shaft rotation.
Pump/motor overall efficiency = power output in the form of pressure increase/electric energy supplied to the motor.

Typical Centrifugal Pump

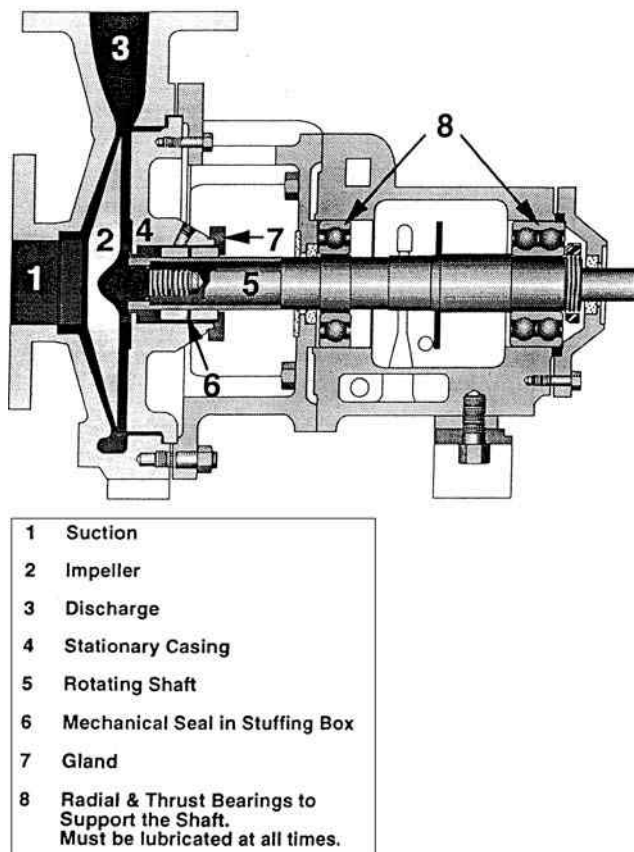


FIGURE 31.2 The purpose of the seal is to stop the liquid in the pump from leaking between the rotating shaft (5) and the stationary casing (4).

31.2.1 Theory of Operation of a Centrifugal Pump

The liquid entering the pump suction is forced by the upstream pressure to enter the impeller that discharges the liquid at its periphery at a higher velocity. This velocity is converted to pressure energy by means of a volute (Fig. 31.4) or by a set of stationary diffusion vanes (Fig. 31.5) surrounding the impeller periphery.

Bernoulli's equation in SI units:

$$H = \frac{0.102p}{r} + \frac{c^2}{2g} + y = \text{constant}$$

where p = pressure, pa

r = density, kg/m³

c = absolute velocity, m/s

g = gravity constant, 9.81 m/s²

y = elevation, m

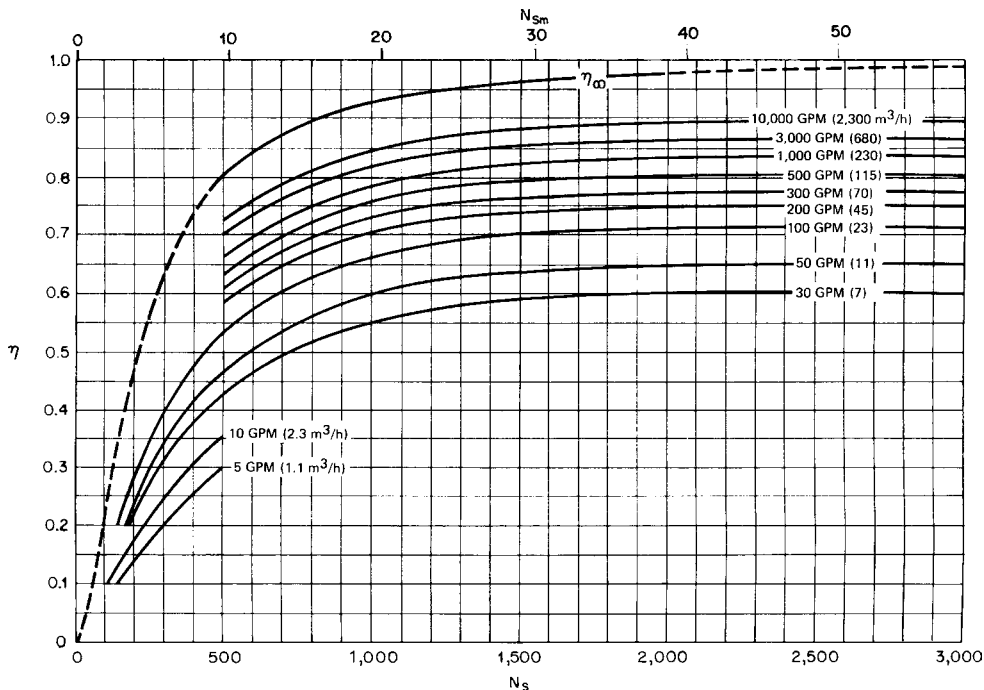


FIGURE 31.3 Efficiency as a function of specific speed and capacity. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

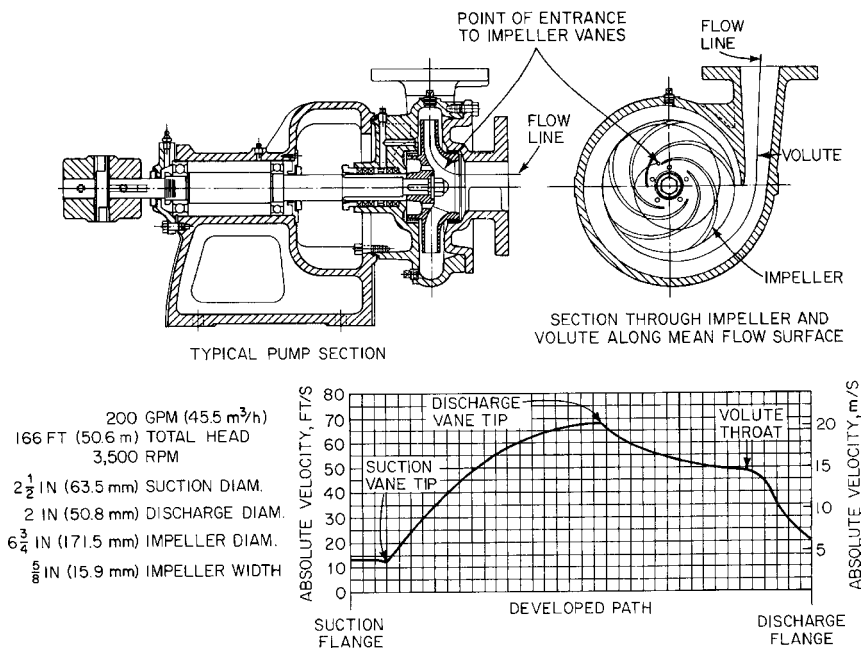
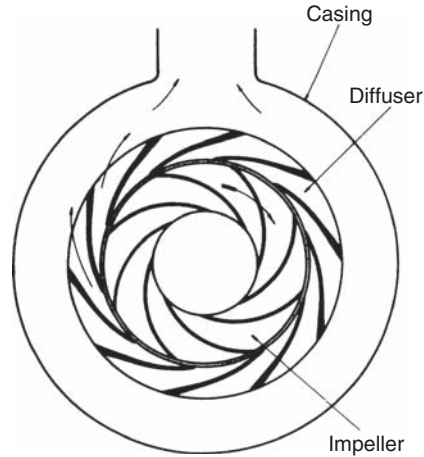


FIGURE 31.4 Typical single-stage end-suction volute pump. (Source: Reprinted with permission from Flowserve Corporation.)

FIGURE 31.5 Typical diffuser-type pump.



Writing this equation between intake and discharge (conservation of energy) confirms that a drop in velocity (in the volute) will result in pressure increase at the discharge of the pump.

Figure 31.4 illustrates the variation of velocity in a centrifugal pump. The pump head is the pressure increase across the pump ($P_{\text{discharge}} - P_{\text{intake}}$) converted to meters or feet.

A centrifugal pump can be (1) single stage (one impeller), or (2) multistage (two or more impellers operating in series, each taking its suction from the discharge of the preceding impeller).

The pump can be classified as axially split (the casing splits by a plane passing through the pump shaft), or radially split (the casing splits by a plane perpendicular to the pump shaft). The axis of rotation determines whether the pump is a horizontal or vertical unit. Centrifugal pumps are classified still further according to the location of the suction nozzle:

1. End suction (Figs. 31.6 through 31.9)
2. Side suction (Figs. 31.10 and 31.11)
3. Bottom suction (Fig. 31.12)
4. Top suction (Fig. 31.13)

Some pumps have the liquid coming to and being conducted away from the pumps by piping. Other pumps, most often vertical types, are submerged in their suction supply.

Figures 31.6 and 31.7 show typical constructions of the bowl section of a single-stage axial-flow propeller pump, a vertical dry-pit single-suction volute pump, and a horizontal double-suction volute pump, respectively. Names recommended by the Hydraulic Institute for various parts are given in Table 31.1.

31.2.2 Casings and Diffusers

The volute casing pump (refer to Fig. 31.4) are spiral-shaped casing surrounding the impeller collects the liquid discharged by the impeller and converts velocity energy to pressure energy. The volute increases in area to convert velocity energy to pressure energy.

FIGURE 31.6 Vertical wet-pit diffuser pump bowl (numbers refer to parts listed in Table 31.1). (Source: Reprinted with permission from Flowserve Corporation.)

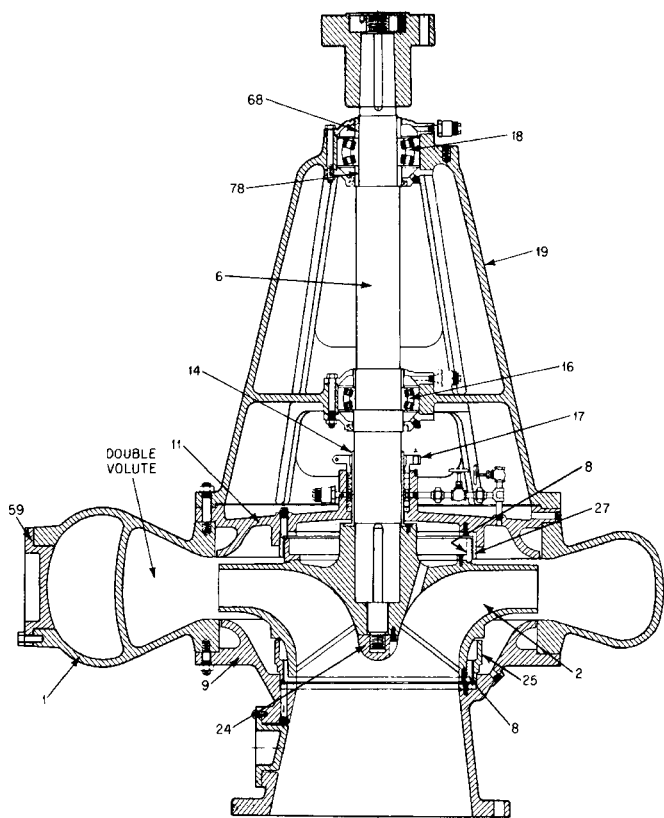
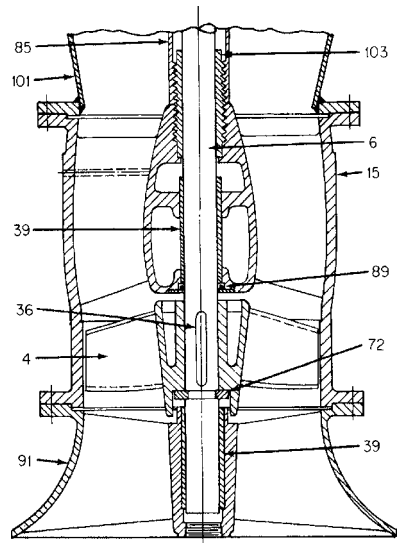


FIGURE 31.7 Sectional view of a vertical-shaft end-suction pump with a double-volute casing (numbers refer to parts listed in Table 31.1). (Source: Reprinted with permission from Flowserve Corporation.)

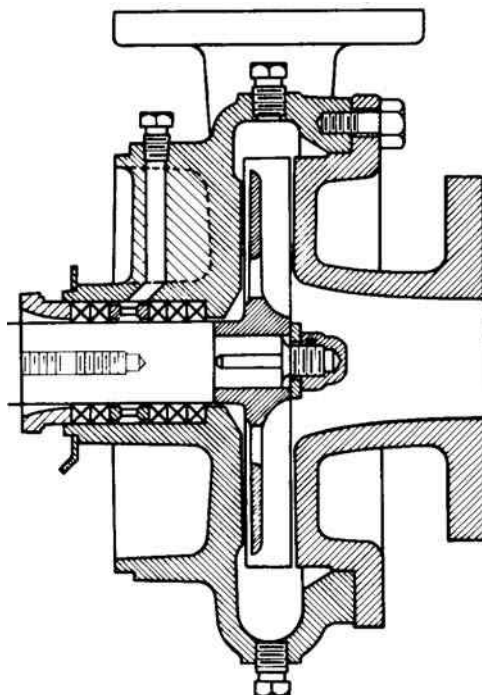


FIGURE 31.8 End-suction pump with open impeller and removable suction cover. (Source: Reprinted with permission from Flowserve Corporation.)

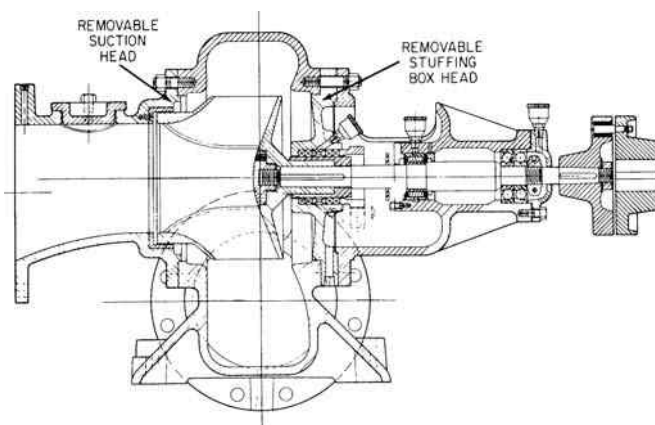


FIGURE 31.9 End-suction pump with removable suction and stuffing box heads. (Source: Reprinted with permission from Flowserve Corporation.)

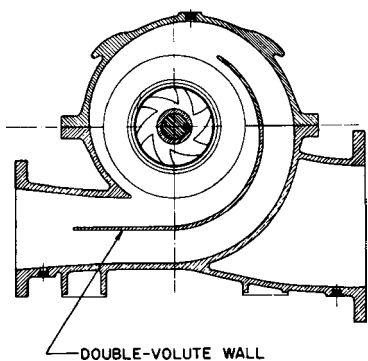


FIGURE 31.10 Transverse view of a double-volute casing pump.



FIGURE 31.11 Axially split casing horizontal-shaft double-suction volute pump. (Source: Reprinted with permission from Flowserve Corporation.)

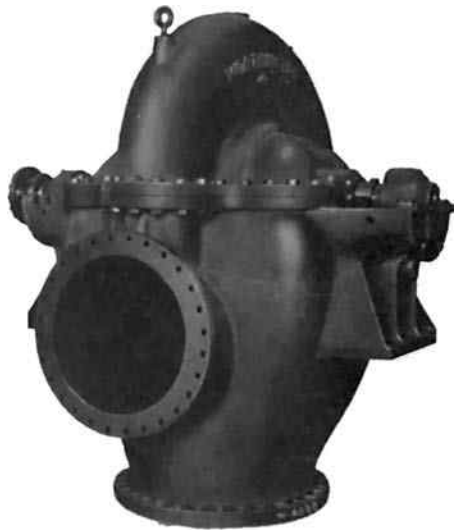


FIGURE 31.12 Bottom-suction axially split casing single stage pump. (Source: Reprinted with permission from Flowserve Corporation.)

The diffusion vanes and concentric casing of a diffuser pump fulfill the same function as the volute casing in energy conversion.

31.2.3 Radial Thrust

In a single-volute casing (Fig. 31.15), uniform or near-uniform pressures act on the impeller when the pump is operated at design capacity (which coincides with the

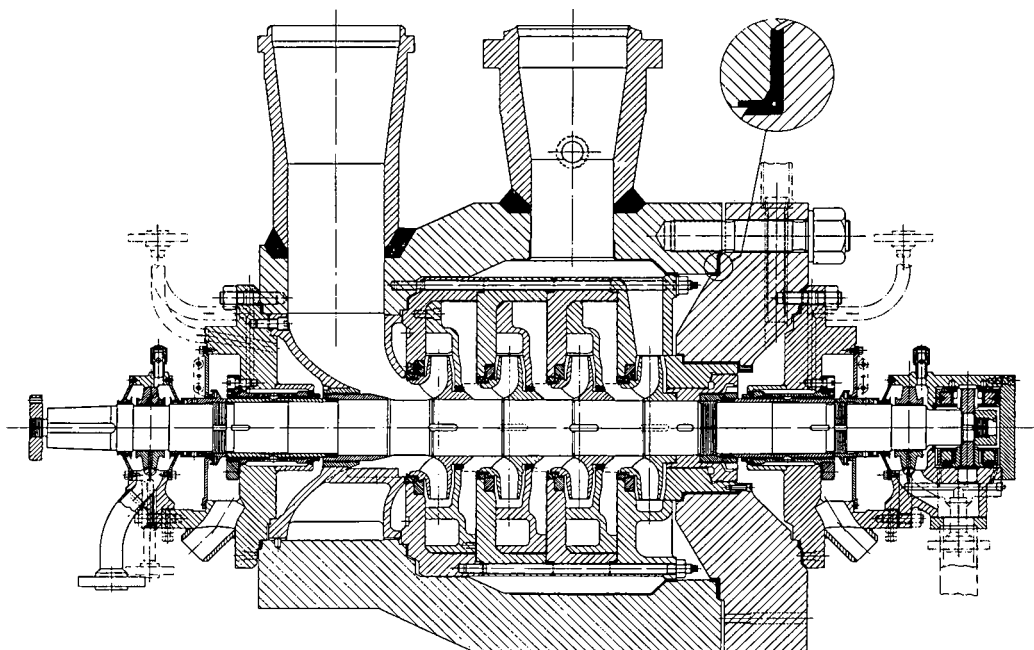


FIGURE 31.13 Double-casing multistage pump with radially split inner casing. (Source: Reprinted with permission from Flowserve Corporation.)

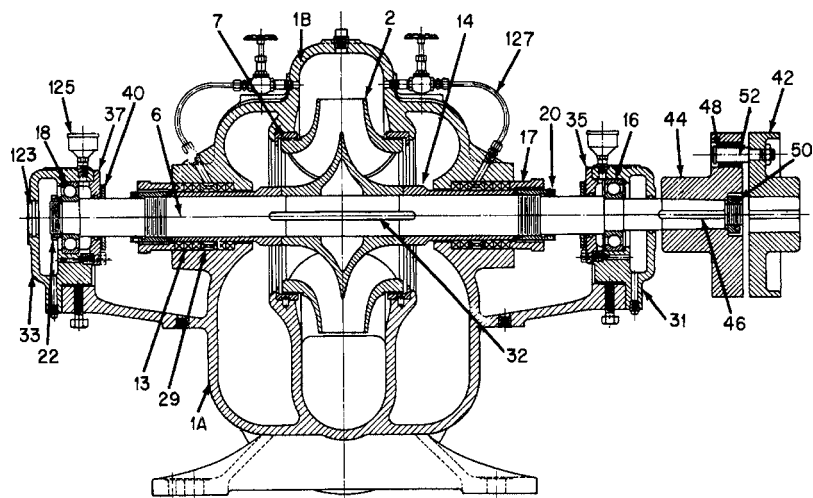


FIGURE 31.14 Horizontal single-stage double-suction volute pump (numbers refer to parts listed in Table 31.1). (Source: Reprinted with permission from Flowserve Corporation.)

Item No.	Name of Part*	Item No.	Name of Part*
1	Casing	33	Bearing housing (outboard)
1A	Casing (lower half)	35	Bearing cover (inboard)
1B	Casing (upper half)	36	Propeller key
2	Impeller	37	Bearing cover (outboard)
4	Propeller	39	Bearing bushing
6	Pump shaft	40	Deflector
7	Casing ring	42	Coupling (driver half)
8	Impeller ring	44	Coupling (pump half)
9	Suction cover	46	Coupling key
11	Stuffing box cover	48	Coupling bushing
13	Packing	50	Coupling lock nut
14	Shaft sleeve	52	Coupling pin
15	Discharge bowl	59	Handhole cover
16	Bearing (inboard)	68	Shaft collar
17	Gland	72	Thrust collar
18	Bearing (outboard)	78	Bearing spacer
19	Frame	85	Shaft enclosing tube
20	Shaft sleeve nut	89	Seal
22	Bearing lock nut	91	Suction bowl
24	Impeller nut	101	Column pipe
25	Suction head ring	103	Connector bearing
27	Stuffing box cover ring	123	Bearing end cover
29	Lantern ring (seal cage)	125	Grease (oil) cup
31	Bearing housing (inboard)	127	Seal pipe (tubing)
32	Impeller key		

*These parts are called out in Figs. 31.6, 31.7, and 31.14.

TABLE 31.1 Recommended Names for Centrifugal Pump Parts

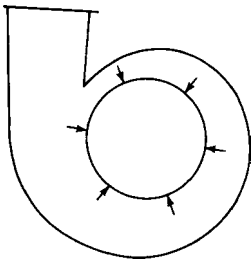
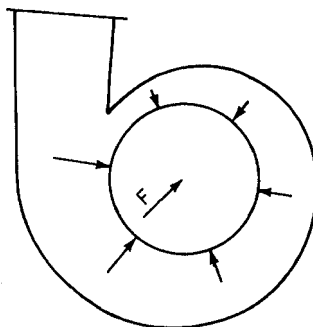


FIGURE 31.15 Uniform casing pressures exist at design capacity, resulting in zero radial reaction. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

FIGURE 31.16 At reduced capacities uniform pressure do not exist in a single-volute casing, resulting in a radial reaction F .



best efficiency). At other capacities, the pressures around the impeller are not uniform (Fig. 31.16) and there is a resultant radial reaction (F).

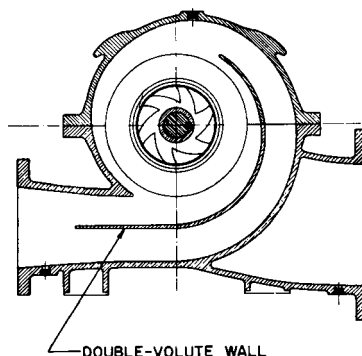
Therefore, the shaft diameter as well as bearing size must be increased to cope with the radial thrust, which occurs at reduced capacities (i.e., lower than rated capacity), or a casing design which develops a much smaller radial reaction force at partial capacities (double-volute design).

The application of the double-volute design principal to neutralize radial reaction forces at reduced capacity is illustrated in Fig. 31.17. Basically, the design consists of two 180° volutes. The forces around the shaft are approximately equal and opposite. Therefore, the radial force acting on the shaft and bearings has been reduced significantly. Also, the rib forming the second volute strengthens the casing (Fig. 31.7). An individual stage of a multistage pump can be made as a double volute as illustrated in Fig. 31.18.

31.2.4 Hydrostatic Pressure Tests

It is a standard practice for the manufacturer to conduct hydrostatic tests for the parts of a pump that contain liquid under pressure. The duration of the test is about 5 minutes. Such a test demonstrates that the casing containment is sound and that there is no leakage of liquid to the exterior.

FIGURE 31.17 Transverse view of a double-volute casing pump.



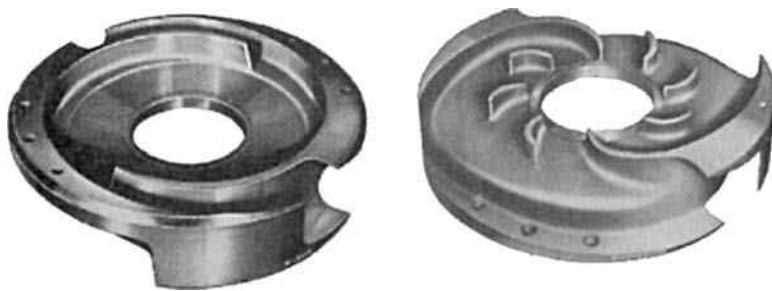


FIGURE 31.18 Double volute of a multistage pump: front view (left) and back view (right).
(Source: Reprinted with permission from Flowserve Corporation.)

Each part of the pump that contains liquid under pressure should be capable of withstanding a hydrostatic test at not less than the greater of the following:

1. 150% of the pressure that will occur in that part when the pump is operated at rated conditions for the given application of the pump
2. 125% of the pressure that would occur in that part when the pump is operated at rated pump rpm for the given application, but with the pump discharge valve closed.

31.2.5 Impeller

In a single-suction impeller, the liquid enters the suction eye on one side only. A double-suction impeller is, in effect, two single-suction impellers arranged back to back in a single casing, in which the liquid enters the impeller simultaneously from both sides, while the two casing suction passageways are connected to a common suction passage and a single-suction nozzle.

For the general service, single-stage axially split casing design (ease of maintenance), a double-suction impeller is favored because it is theoretically in axial hydraulic balance and because the greater suction area of a double-suction impeller permits the pump to operate with less net absolute suction head.

For small units, the single-suction impeller is more practical for manufacturing reasons. End-suction pumps with single-suction impellers have both first-cost and maintenance advantages not obtainable with double-suction impellers.

In multistage pumps, single-suction impellers are almost universally used because of the design and first-cost complexity that double-suction staging introduces.

Impellers can be classified by the shape and form of their vanes:

1. The straight-vane impeller (Figs. 31.19 through 31.23)
2. The Francis-vane or screw-vane impeller (Fig. 31.23b)
3. The mixed-flow impeller (Fig. 31.24)
4. The propeller or axial-flow impeller (Fig. 31.25)

Figure 31.26 shows the variations in impeller profiles with specific speed for the various types of impellers.



FIGURE 31.19 Straight-vane radial single-suction closed impeller. (Source: Reprinted with permission from Flowserve Corporation.)



FIGURE 31.20 Open impellers. Notice that the impellers at left and right are strengthened by a partial shroud. (Source: Reprinted with permission from Flowserve Corporation.)



FIGURE 31.21 Open impeller with partial shroud. (Source: Reprinted with permission from Flowserve Corporation.)



FIGURE 31.22 Semiopen impeller. (Source: Reprinted with permission from Flowserve Corporation.)



FIGURE 31.23 (a) Front and back views of an open impeller with partial shroud and pump-out vanes on back side. (b) Francis-vane radial double-suction closed impeller. (Source: Part (b) has been reprinted with permission from Flowserve Corporation.)



FIGURE 31.24 Open mixed-flow impeller. (Source: Reprinted with permission from Flowserve Corporation.)

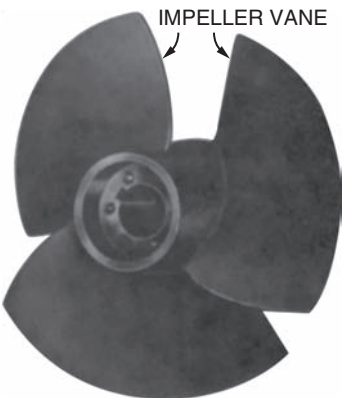


FIGURE 31.25 Axial-flow impeller. (Source: Reprinted with permission from Flowserve Corporation.)

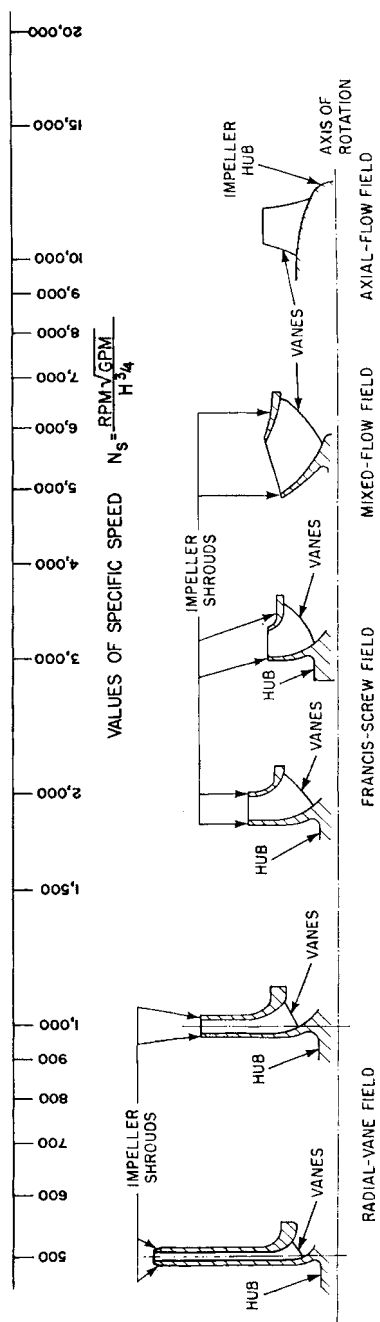


FIGURE 31.26 Variations in impeller profiles with specific speed and approximate range of specific speed for the various types. (To convert specific speed in USCS units to SL, multiply by 0.6123.)

31.2.6 Axial Thrust

The axial hydraulic thrust on an impeller is the sum of the unbalanced forces acting in the axial direction. Thrust bearings are required to handle the axial thrust.

Theoretically, a double-suction impeller is in hydraulic axial balance (Fig. 31.27). In reality, the balance of forces may not exist for the following reasons:

- 1. The suction passages to the two suction eyes may not provide equal or uniform flows to the two sides.
- 2. External conditions, such as an elbow located too close to the pump suction nozzle, may cause unequal flow to the two suction eyes, etc.

All the above factors will create an axial unbalance. Therefore, all the centrifugal pumps having double-suction impellers require thrust bearings.

The single-suction radial-flow impeller is subjected to axial thrust (Fig. 31.27). Thrust bearings are required.

31.2.7 Axial Thrust in Multistage Pumps

Most multistage pumps are built with single-suction impellers. Two possible arrangements exist:

- 1. Several single-suction impellers are mounted on the shaft, each having its suction inlet from the discharge of the preceding impeller. The pressure is ascending gradually across the pump (Fig. 31.28). The axial thrust is then balanced by a hydraulic balancing device.
- 2. An even number of single-suction impellers may be used, one-half facing in one direction and the other half facing in the opposite direction. Therefore, the axial thrust is balanced out (Fig. 31.29).

This mounting is called opposed impellers (the pump design is more complex and its cost is higher).

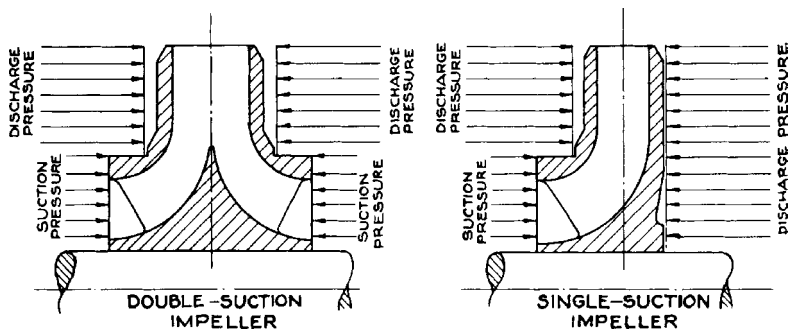


FIGURE 31.27 Origin of pressures acting on impeller shrouds to produce axial thrust. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

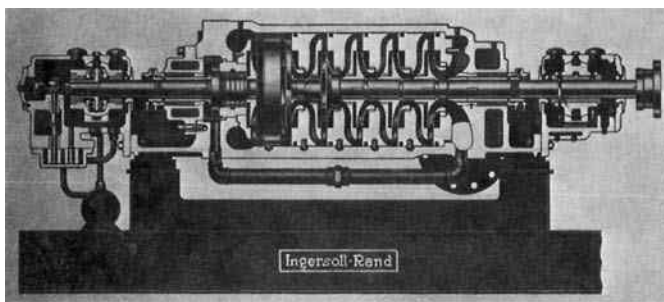


FIGURE 31.28 Multistage pump with single-suction impellers facing in one direction and hydraulic balancing device. (Source: Ingersoll-Rand.)

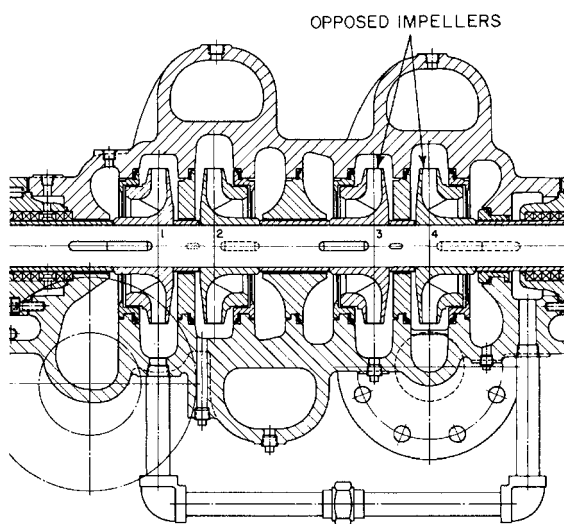


FIGURE 31.29 Four-stage pump with opposed impellers. (Source: Reprinted with permission from Flowserve Corporation.)

31.2.8 Hydraulic Balancing Devices

The total axial thrust is the sum of all the individual impeller thrusts. A hydraulic balancing device is needed to balance the axial thrust.

Balancing Drums

The balancing drum is illustrated in Fig. 31.30. The balancing chamber at the back of the last-stage impeller is separated from the pump interior by a drum that is either keyed or screwed to the shaft and rotates with it. The drum is separated by a small radial clearance from the stationary portion of the balancing device, called the balancing-drum head, which is fixed to the pump casing.

The balancing chamber is connected either to the pump suction or to the vessel from which the pump takes its suction.

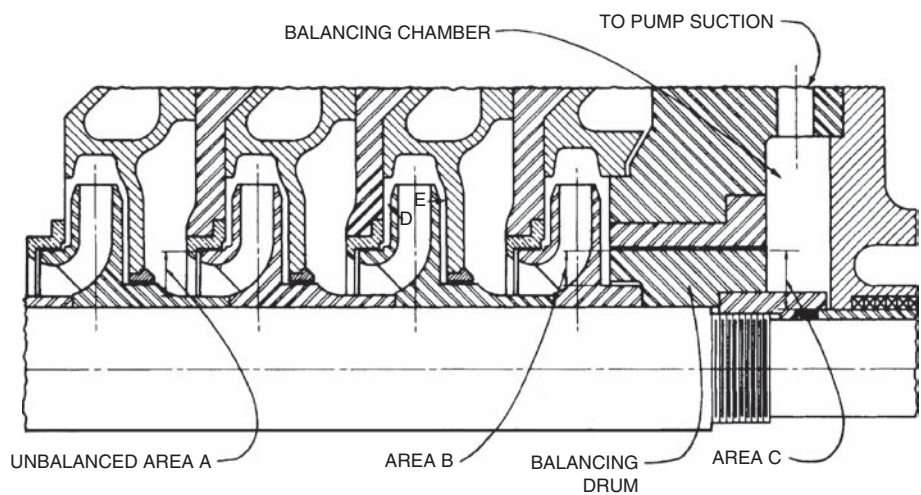


FIGURE 31.30 Balancing drum. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

The forces acting on the balancing drum are

1. Toward the discharge end: the discharge pressure multiplied by the front balancing area (area B) of the drum
2. Toward the suction end: the back pressure in the balancing chamber multiplied by the back balancing area (area C) of the drum

The first force is greater than the second, thereby counterbalancing the axial thrust exerted upon the single-suction impellers. The drum diameter can be selected to balance the axial thrust completely or within 95%.

Balancing Disks

The operation of the simple balancing disk is illustrated in Fig. 31.31. The disk is fixed to and rotates with the shaft. It is separated by a small axial clearance from the balancing-disk head, which is fixed to the casing. The leakage through this clearance flows into the balancing chamber and from there either to the pump suction or to the vessel from which the pump takes its suction. The back of the balancing disk is subjected to the balancing chamber back pressure. Whereas the disk face experiences a range to back pressure. These vary from discharge at its smallest diameter to back pressure at its periphery. The inner and outer disk diameters are chosen so that the difference between the total force acting on the disk face and that acting on its back will balance the impeller axial thrust.

If the axial thrust of the impellers exceeds* the thrust acting on the disk during operation, the disk-shaft assembly moves toward the disk head. This reduces the axial clearance between the disk and the disk head, which results in a reduction in the amount of leakage through the clearance. Therefore, the back pressure in the balancing chamber

*How would the axial thrust of the impellers exceed the thrust acting on the disk during operation? (Changes in flow or liquid temperature.)

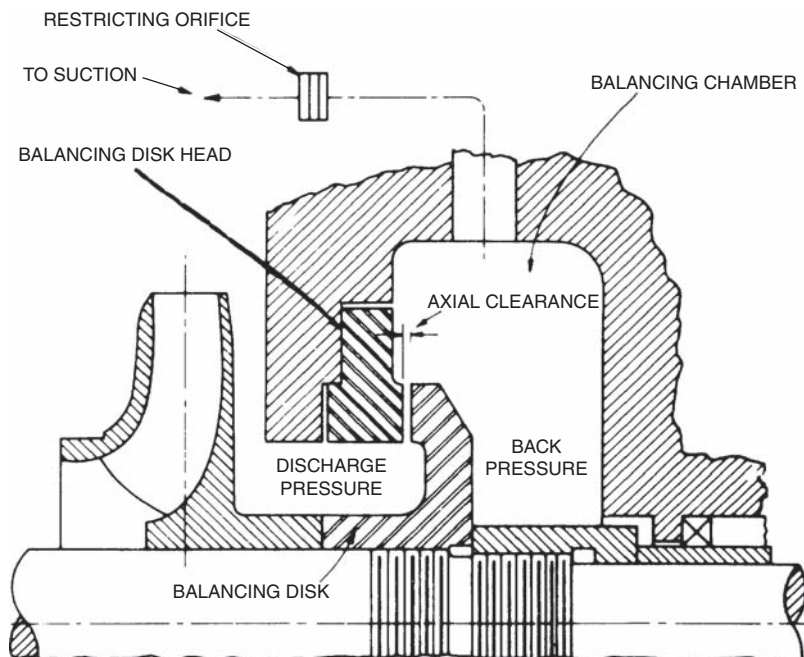


FIGURE 31.31 Simple balancing disk. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

will be reduced, which increases the differential pressure acting on the disk and moves the disk-shaft assembly away from the disk head, increasing the clearance. Now the pressure builds up in the balancing chamber, and the disk is again moved toward the disk head until an equilibrium is reached.

A combination of a balancing disk and a drum can also be used (Fig. 31.32).

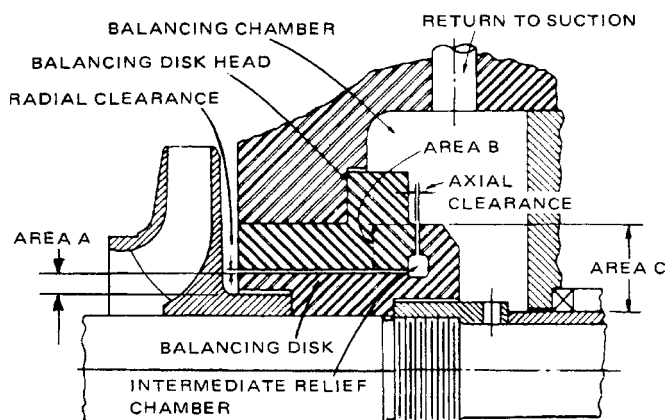


FIGURE 31.32 Combination balancing disk and drum. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

NOTE The water equality is very important in maintaining smooth operation of the pump. If the water has small, even microscopic impurities (solids), clearances may become plugged, leading to pump impairment. (Filters are needed to clean the water—demineralized water is usually used.)

31.3 Mechanical Seals

In the ordinary stuffing box, the sealing between the moving shaft sleeve and the stationary portion of the box is accomplished by means of rings of packing forced between the two surfaces and held tightly in place by stuffing box gland (Fig. 31.33).

The leakage around the shaft is controlled by tightening up or loosening the gland studs. The sealing area consists of the surface of the rotating shaft (or shaft sleeve) and the stationary packing. Leakages can be reduced or eliminated by tightening the gland nuts; however, excessive tightening of the gland will result in accelerated wear of the shaft or shaft sleeve and packing.

Under severe conditions (high temperature and pressure) or if the reliability of the pump cannot be compromised (i.e., failure of the pump will lead to a complete or partial plant shutdown), mechanical seals are used. This type of seals are studied in detail later.

31.4 Bearings

The function of bearings in centrifugal pumps is to support the shaft or rotor and keep it in correct alignment with the stationary parts under the action of radial and transverse loads. Cooling of the heat generated (by friction) and lubrication are essential for bearing operation. Usually, cooling is accomplished by circulating the oil (used for lubrication) through a separate water-to-oil cooler. Otherwise, a jacket through which a cooling liquid is circulated is usually incorporated as part of the housing. Pump bearings may be rigid or self-aligning, that is, the bearing will automatically adjust itself to a change in the angular position of the shaft. Refer to Fig. 31.34.

31.5 Couplings

Couplings are used to connect centrifugal pumps to their drivers (motors).

31.6 Bedplates

The prime function of a pump bedplate is to provide a mounting surface for the pump and driver feet that can be rigidly attached to the foundation (Figs. 31.35 through 31.37).

FIGURE 31.33 Split stuffing box gland.



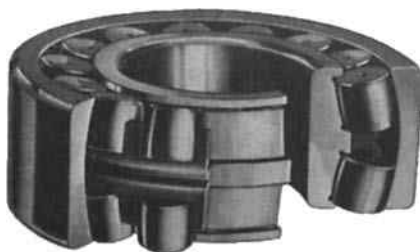


FIGURE 31.34 Self-aligning spherical roller bearing. (Source: Reprinted with permission from SKF.)

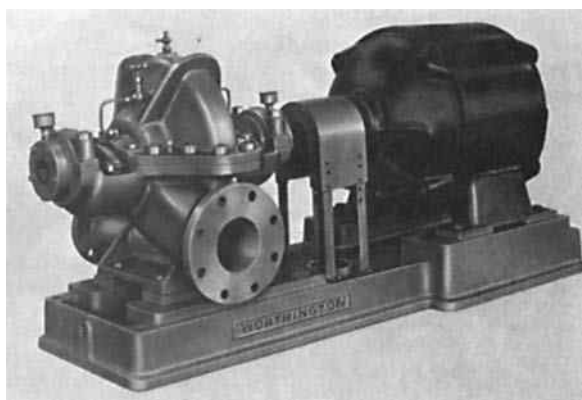


FIGURE 31.35 Horizontal-shaft centrifugal pump and driver on cast iron bedplate. (Source: Reprinted with permission from Flowserve Corporation.)

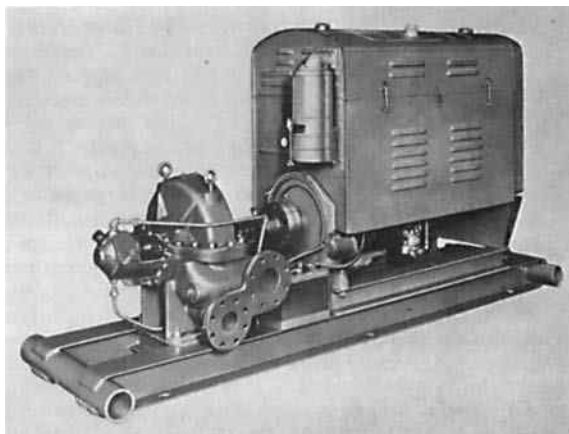


FIGURE 31.36 Pump and internal combustion engine mounted on portable steel skid base. (Source: Reprinted with permission from Flowserve Corporation.)

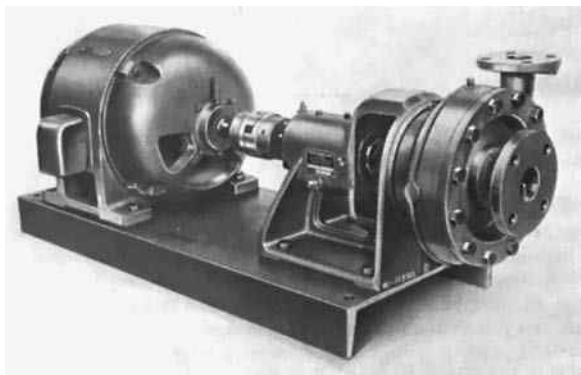


FIGURE 31.37 Small centrifugal pump on structural steel bedplate made of an unple channel.
(Source: Reprinted with permission from Flowserve Corporation.)

31.7 Minimum Flow Requirement

Each centrifugal pump has a minimum flow requirement to operate the pump safely. This is the minimum acceptable flow required to remove the heat generated by the pumping process. If it is not met, the pump will overheat and destroy itself. The minimum flow required is usually specified by the manufacturer. It depends on the flow, temperature, and pressure of the liquid being pumped.

31.8 Centrifugal Pumps: General Performance Characteristics

Following are the characteristics of a centrifugal pumps:

- *Head.* The pump head H represents the net work done on a unit weight of liquid in passing from the inlet or suction flange (s) to the discharge flange (d).

$$H = \left(\frac{P}{\gamma} + \frac{V^2}{2g} + Z \right)_d - \left(\frac{P}{\gamma} + \frac{V^2}{2g} + Z \right)_s$$

where P/γ = the pressure head
 $V^2/2g$ = velocity head (represents the kinetic energy)
 Z = elevation head or potential head

- *Power.* The pump output is customarily given as liquid horsepower.

$$1\text{hp} = \frac{QH(\text{sp. gr.})}{8.82} \quad (Q \text{ in ft}^3/\text{s})$$

$$P = 9797 QH(\text{sp. gr.}) \quad (P \text{ in watts})$$

- *Efficiency.* The pump efficiency is the liquid horsepower divided by the power input to the pump shaft. The latter is called brake horsepower (bhp). The overall efficiency (motor-pump) is called the wire-to-liquid efficiency. It is the liquid horsepower divided by the power input to the motor.

FIGURE 31.38 Pump characteristics, radial vanes ($\text{ft} \times 0.3048 = \text{m}$; $\text{hp} \times 745.7 = \text{W}$; $\text{gpm} \times 0.06309 = 1/\text{s}$).

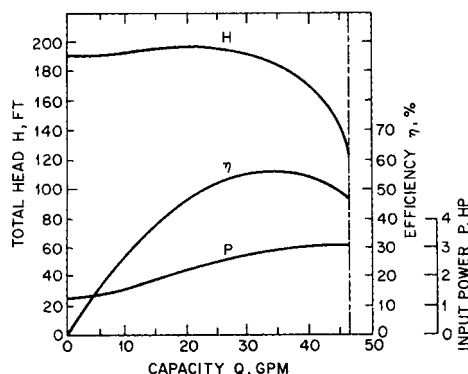


Figure 31.38 shows a typical characteristics of a fully shrouded pump. The best efficiency, $\eta = 55\%$ occurred at a specific speed, $n_s = 475$.

At low specific speed, the efficiency is usually low because the head is very high (i.e., there is a significant pressure increase across the pump and a relatively low flow. Refer to Fig. 31.39).

The head-capacity curve is a very useful tool in predicting flows downstream of a centrifugal pump. It is usually developed at the commissioning stage of a pump and remains relatively unchanged for the lifetime of the pump. The curve is accurate within a few percentages.

- Problem. If the pump flow varies between 0 and 800 L/s, set up the appropriate instrumentations to measure the flow accurately.

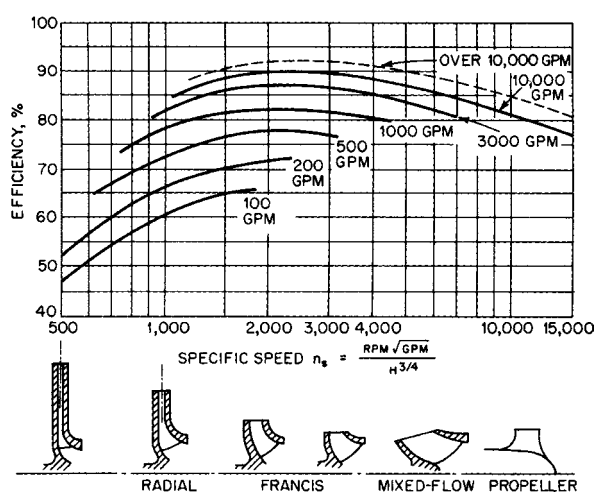


FIGURE 31.39 Pump efficiency versus specific speed and size ($\text{gpm} \times 0.06309 = 1/\text{s}$). (Source: Reprinted with permission from Flowserve Corporation.)

31.9 Cavitation

The information and subsequent collapse of vapor-filled cavities in a liquid due to dynamic action are called *cavitation*. The cavities may be bubbles, vapor-filled pockets, or a combination of both. The local pressure must be at or below the vapor pressure of the liquid for cavitation to begin, and the cavities must encounter a region of pressure higher than the vapor pressure in order to collapse.

Bubbles that collapse on a solid boundary may cause servere mechanical damage, resulting in pitting of the boundary. Pressures in the order of 10^4 atm have been estimated during the collapse of bubble. All known materials can be damaged by exposure to bubble collapse for sufficiently long time. This is properly called “cavation erosion” or pitting.

Figure 31.40 shows extensive damage to the suction side of pump impeller vanes after about 3 months’ operation with cavitation. At two locations, the pitting has penetrated deeply into the 9.5 mm thickness of SS. The unfavourable inlet flow conditions, believed to have been the cause of the cavitation, were at least partly due to elbows in the approach piping. Modifications in the approach piping and the pump inlet passages reduced the cavitation to the point that impeller life was extended to several years.

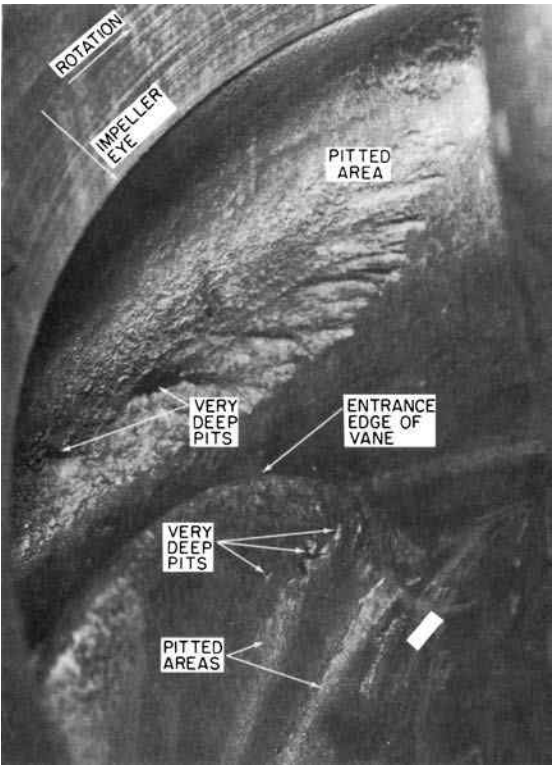


FIGURE 31.40 Impeller damaged by cavitation. (Source: Transamerica Delaval.)

Cavitation pitting, as measured by weight of the boundary material removed per unit time, frequently increases with time (especially if the velocity of the liquid is very high). Cast iron and steel are particularly vulnerable.

Centrifugal pumps begin to cavitate when the suction head is insufficient to maintain pressures above the vapor pressure throughout the flow passages. Apart from the noise and vibration, cavitation damage may render an impeller useless in as little as a few weeks of continuous operation.

31.10 Net Positive Suction Head

The net positive suction head (NPSH), h_{sv} , is a statement of the minimum suction conditions required to prevent cavitation in a pump. The required, or minimum, NPSH must be determined by test and is usually stated by the manufacturer. The available NPSH at installation must be at least equal to the required NPSH if cavitation is to be prevented. Increasing the available NPSH provides a margin of safety against the onset of cavitation. Figure 31.40 and the following symbols are used to compute the NPSH:

p_a = absolute pressure in atmosphere surrounding gauge Fig. 31.41

p_s = gage pressure indicated by gauge or manometer connected to pump suction at section s-s; may be positive or negative

p_t = absolute pressure on free surface of liquid being pumped corresponding to the temperature at section s-s

p_{vp} = vapor pressure of liquid being pumped corresponding to the temperature at section s-s

h_f = lost head due to friction in suction line between tank and section s-s

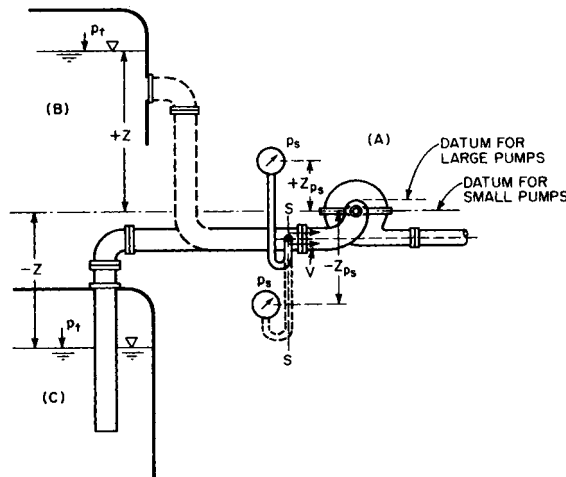


FIGURE 31.41 Definition sketch for computing NPSH. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

V = average velocity at section s-s

Z, Z_{ps} = vertical distances defined by Fig. 31.41; may be positive or negative

γ = specific weight of liquid at pumping temperature

It is satisfactory to choose the datum for small pump as shown in Fig. 31.41, but with large pumps the datum should be raised to the elevation where cavitation is most likely to start. For example, the datum for a large horizontal-shaft propeller pump should be taken at the highest elevation of the impeller-vane tips. The available NPSH is given by

$$h_{sv} = \frac{p_a - p_{vp}}{\gamma} + \frac{p_s}{\gamma} + Z_{ps} + \frac{V^2}{2g} \quad (31.1)$$

$$h_{sv} = \frac{p_t - p_{vp}}{\gamma} + Z - h_f \quad (31.2)$$

Consistent units must be chosen so that each term in Eqs. (31.1) and (31.2) represents feet (or meters) of the liquid pumped. In Eq. (31.1), the first term represents the height of a liquid barometer, h_b , containing the liquid being pumped, and the sum of the remaining terms represents the suction head h_s (refer to the head equation). Therefore,

$$h_{sv} = h_b + h_s$$

Usually, a positive value of h_s is called a suction head and a negative value of h_s is called a suction lift.

Figures 31.42 and 31.43, taken from the Hydraulic Institute Standards, are useful in determining suction conditions so that cavitation may be avoided. They relate specific speed, total head of first stage, and total suction head or total suction lift at sea level for pumps handling clear cold water, that is, temperature not exceeding 29.4°C (85°F).

There are separate capacity scales for single- and double-suction impellers.

31.11 Maintenance Recommended on Centrifugal Pumps

Maintenance must be performed on centrifugal pumps to ensure smooth operation for extended periods of time. The major contributors to poor mechanical performance are

1. Unbalance of mechanical components.
2. Incorrect alignment between the pump and the driver.
3. Pump shaft end play, radial deflection, run-out, and pump squareness exceed acceptable tolerances.

Incorrect alignment between the pump and the driver can cause

1. Extreme heat in couplings
2. Cracked shafts and totally failed shafts due to fatigue failure
3. Bearing failure

Also, pump mechanical problems prevent the best seals from operating for a few hours.

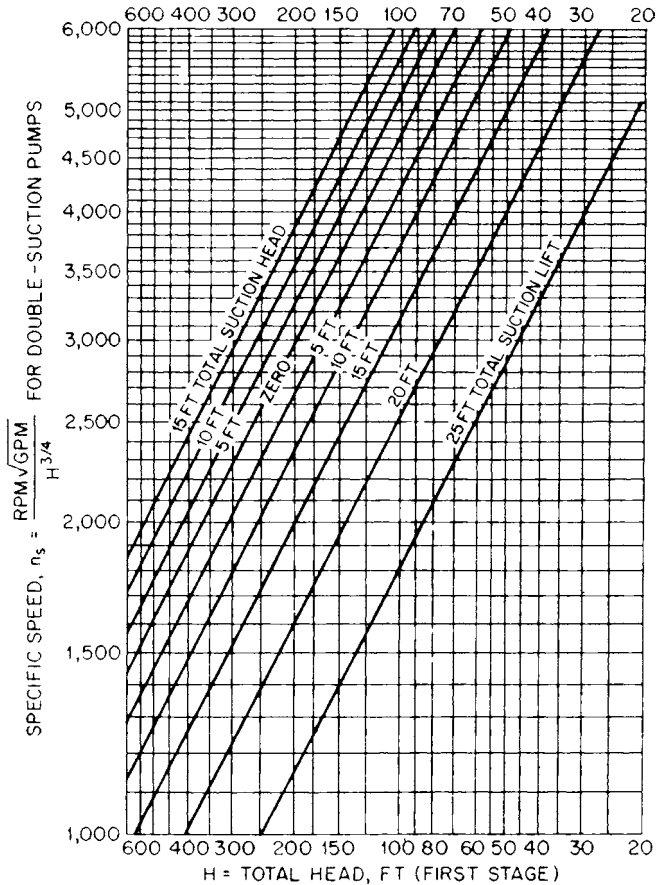


FIGURE 31.42 Upper limits of specific speeds for double-suction pumps handling clear water at 29.4°C (85°F) at sea level (ft $\times$ 0.3048 = m). (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

31.12 Recommended Pump Maintenance

1. Hot and cold alignment of pump and driver shafts. Several procedures have been suggested by various people to estimate or actually measure alignment while a pump is running at operating temperature. Some techniques are
 - a. Shutdown after temperatures have stabilized for “hot check” by dial indicators
 - b. Optical measurements cold and hot (A.J. Campbell, Compressor Engineering Corp., Houston)
 - c. Dodd bars (DynAlign) technique (R. Dodd, Chevron), etc.

After measurements have been made and actual misalignment is determined for the hot running condition of the pump, the next step is to calculate the alignment correction required between machines to bring them into alignment during operation.

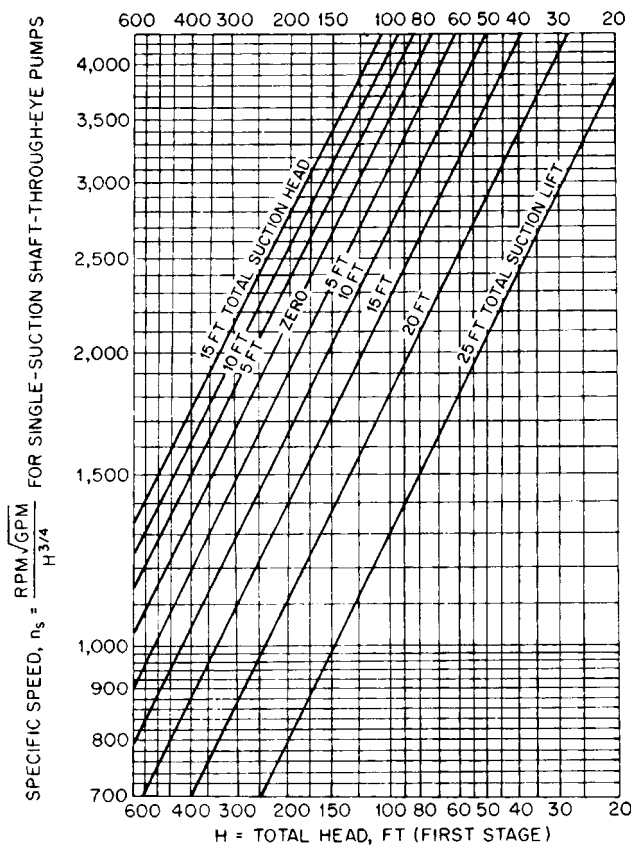
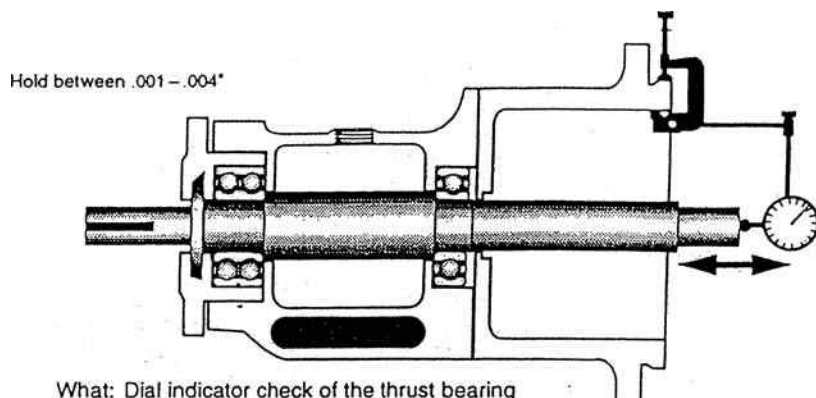


FIGURE 31.43 Upper limits of specific speeds for single-suction shaft-through-eye pumps handling clear water at 29.4°C (85°F) at sea level (ft $\times$ 0.3048 = m). (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

The following should be noted:

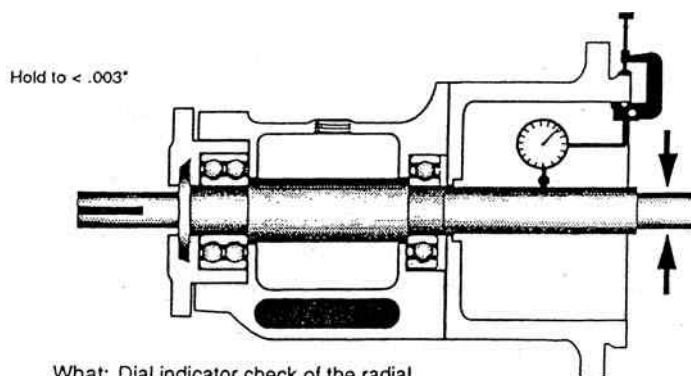
1. The driver and pump are purposely misaligned during cold conditions so that actual operating temperature put the two shafts within acceptable limits.
2. Some pumps have gauges mounted on the pump and motor monitoring the alignment of the pump and motor. These gauges will send a signal to an alarm when the misalignment exceeds the acceptable level and a signal to trip the motor if the misalignment reaches a dangerous level.
3. Checks to improve the pump and seal performance include end play, radial deflection (Fig. 31.44), shaft run-out, stuffing box to shaft perpendicularity (Fig. 31.45), and gland register or stuffing box concentricity (Fig. 31.46).

The part that usually fails in a pump is seldom the root cause failure. The failure will reoccur if the root cause of the failure has not been corrected.



What: Dial indicator check of the thrust bearing and its fit in the housing by moving the shaft back and forth.

Why: Axial movement of the shaft will over or under compress the seal. Also, it can cause pin wear and fretting damage to the sleeve.



What: Dial indicator check of the radial deflection (up and down) of the shaft. This is a check of the radial bearing and its fit in the housing.

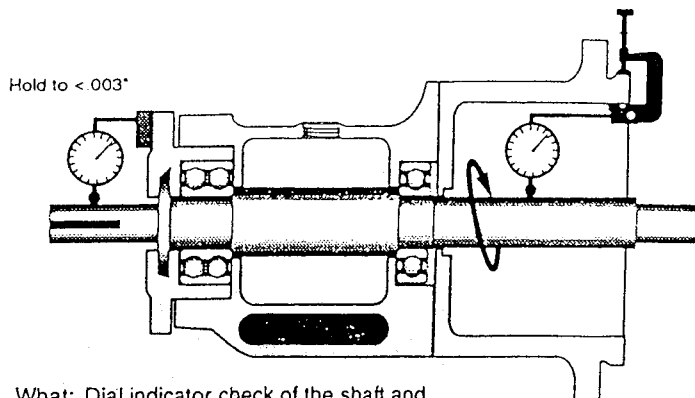
Why: Deflection causes misalignment of the rotor and stator of the seal. This shortens seal life by overworking the drive and spring members.

FIGURE 31.44 Equipment condition (end play and radial deflection).

31.13 Vibration Analysis

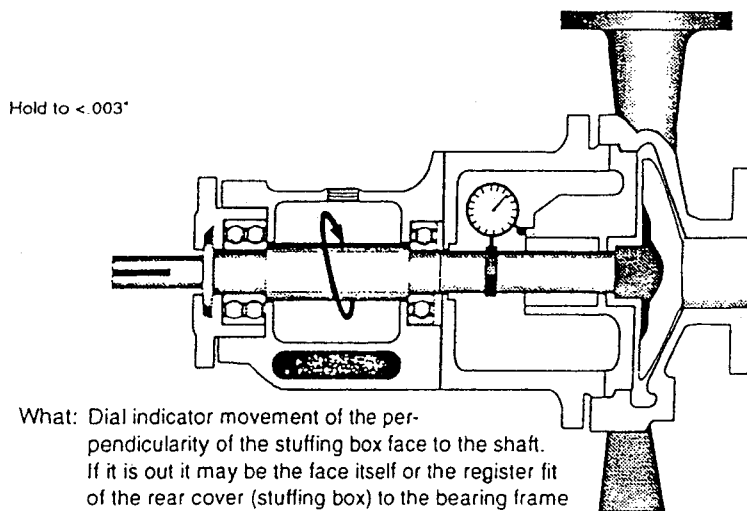
Most mechanical problems with pumps cannot be detected visually. The three gauges of mechanical problems are temperature, vibration, and sound.

It is normal for a machine to vibrate at some level. However, when the velocity vibration level starts to increase to 0.1 in/s zero to peak (0-P) above the "as new installed



What: Dial indicator check of the shaft and sleeve for roundness and straightness. Also drive end of the shaft for a bend.

Why: Run-out or whip causes face wiper. Excessive wiper, affected by seal and shaft speed, causes leakage which in an emission sensitive liquid acceptable. Should the drive end be bent, you can expect pump and motor bearing problems as well as coupling wear.

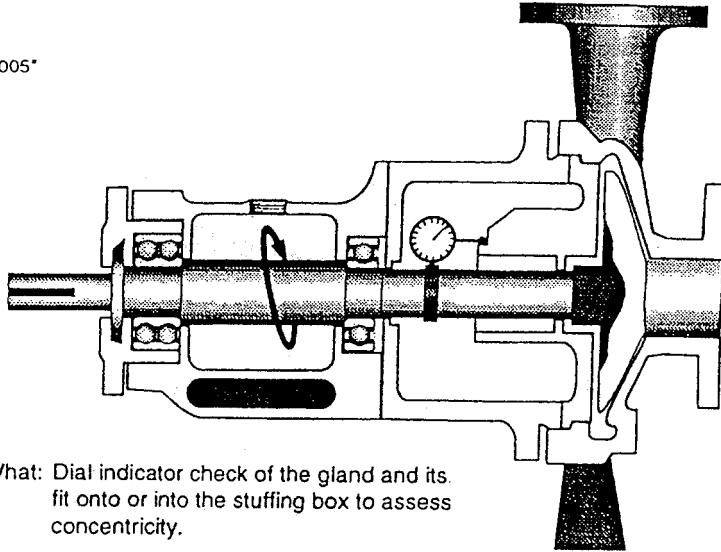


What: Dial indicator movement of the perpendicularity of the stuffing box face to the shaft. If it is out it may be the face itself or the register fit of the rear cover (stuffing box) to the bearing frame or adapter. It must be corrected.

Why: A lack of perpendicularity will overwork a seal to the point of premature failure. The drive, secondary seal (fretting) and spring mechanism can all be points of wear.

FIGURE 31.45 Shaft run-out and stuffing box shaft perpendicularity.

Hold to $< .005^{\circ}$



What: Dial indicator check of the gland and its fit onto or into the stuffing box to assess concentricity.

Why: In many cases a lack of concentricity is not a major problem but it can be. Particularly if a pumping device is incorporated into the seal. Because it sometimes is a critical area, it is good practice to consider it always critical.

FIGURE 31.46 Gland register or stuffing box concentricity.

level," the vibration should be analyzed to determine the possible sources of the mechanical and/or hydraulic problem. Several mechanical and/or hydraulic problems may be producing, for instance, the $1\times$ running speed frequency vibration. The key in using vibration to define the mechanical and/or hydraulic problems is to determine the frequency at which the vibration occurs. Vibration amplitude is also an important factor because it indicates the severity of the vibration.

Many pumps have vibration monitoring equipment built into the pump. They indicate the velocity and amplitude of the vibration continuously so that the appropriate action can be taken to repair any deficiency.

31.14 Bibliography

Karassik, I. J., Krutzsch, W. C., Fraser, W. H., and Messina, J. P., *Pump Handbook*, 2nd ed. McGraw-Hill, New York, 1986.

This page intentionally left blank

CHAPTER 32

Centrifugal Pump Mechanical Seal

32.1 Introduction

A mechanical seal is installed in the pump stuffing box to seal the liquid at various speeds, pressures, and temperatures. Packed stuffing boxes have proven to be inadequate under severe service conditions (high temperature and pressure).

Some mechanical seals have a proven record of good performance under severe conditions for extended periods of time. They should be used where the reliability of the pump cannot be compromised [to reduce the probability of a system or plant failure as well as the maintenance manpower and exposure to contaminated liquids (high radiation dose, hydrazine, etc.)].

32.2 Basic Components

The sealing surfaces are perpendicular to the shaft, with contact between the rotating (primary) and stationary (mating) faces to achieve a dynamic seal.

The primary ring rotates with the pump shaft and the mating ring is fixed to the pump gland plate. Each of the sealing planes is lapped flat (within 3 lightband; 1 lightband = 11.6 millionth of an inch). Wear occurs at the seal faces from sliding contact between the primary and mating rings. The amount of wear is small, as a film of the liquid sealed is maintained between the sealing faces. Normally, the mating surfaces of the seal are of dissimilar materials and held in contact with a spring.

The seal installation is made of two assemblies (Fig. 32.1). The seal head assembly includes the primary ring and its associated component parts. The mating ring assembly includes those parts required for the mating ring to function. During operation, the seal head assembly rotates with the shaft while the mating ring assembly is stationary. There are three points of sealing common to all mechanical seal installations:

1. At the mating surfaces of the primary and mating rings
2. Between the rotating component and the shaft or sleeve
3. Between the stationary component and the gland plate

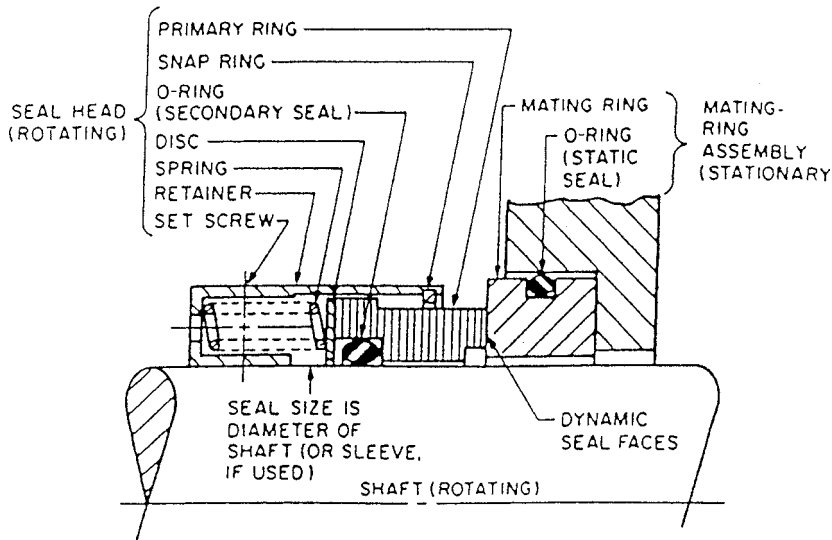


FIGURE 32.1 Mechanical seal.

The secondary seal (located between the rotating seal component and the shaft or sleeve) is partially dynamic. The primary ring moves forward when the seal faces wear. This seal moves along the shaft due to the following:

- Vibration
- Shaft runout
- Thermal expansion

The flexibility in sealing is achieved on this seal by using any of the following:

- Bellows
- O-ring
- Wedge
- V-ring

The mating ring is normally replaceable. Leakage between the mating ring and the gland plate is prevented by a static seal. The mating ring assembly consists of the static seal and the mating ring.

32.2.1 Seal Balance

The performance of the dynamic contact between the mating seal faces determines the effectiveness of the seal. The liquid film between the seal faces could vaporize if the seal load at the faces is high. This will result in high wear rate of the sealing faces. The seal face materials have a bearing limit. This limit should not be exceeded. This condition can be avoided by proper seal balancing.

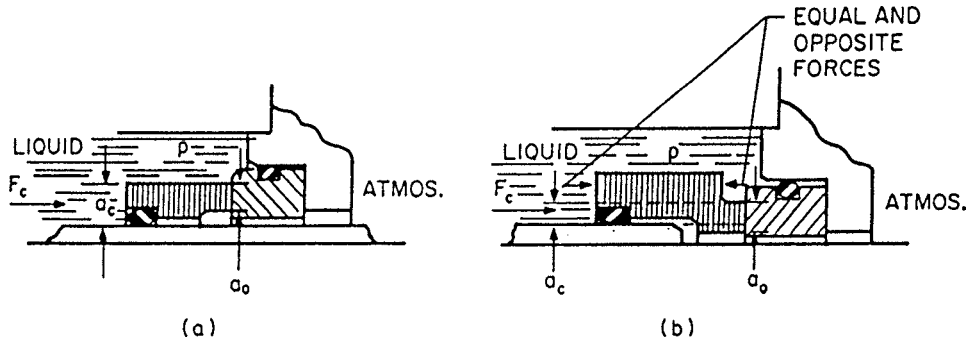


FIGURE 32.2 Hydraulic pressure on the primary ring: (a) unbalanced, (b) balanced.

The liquid pressure forces the primary ring against the mating ring. Figure 32.2a illustrates the pressure acting on the annular area a_c . The closing force on the seal face is given by

$$F_c = pa_c$$

where p = stuffing box pressure, lb/in² (N/m²)
 a_c = hydraulic closing area, in² (m²)

The pressure between the primary and mating rings is

$$p'_f = \frac{F_c}{a_o} = \frac{pa_c}{a_o}$$

where a_o = hydraulic opening area (seal face area), in² (m²).

The pressure can be relieved by reducing a_c (Fig. 32.2b). This is normally done by introducing a shoulder on a sleeve. This technique is called seal balancing. A seal without a shoulder is an unbalanced seal. The balanced seal has a shoulder.

The seal balance is given by

$$b = \frac{a_c}{a_o}$$

The seal balance varies from 0.65 to 1.35.

32.2.2 Face Pressure

The liquid film between the seal faces is generated by the waviness of the sealing planes. This results in a hydraulic pressure between the seal faces. This pressure tends to separate the sealing faces. The pressure distribution between the faces is known as a pressure wedge (Fig. 32.3). This distribution can be as follows:

- Linear
- Concave
- Convex

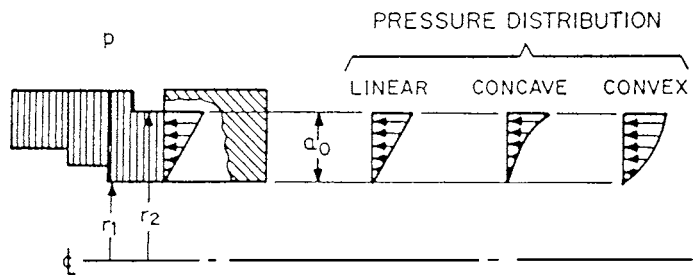


FIGURE 32.3 Pressure wedge.

The actual face pressure is given by

$$P_f = P_h + P_{sp}$$

where $P_h = \Delta p(b - k)$, lb/in² (N/m²)
 Δp = pressure differential across seal face, lb/in² (N/m²)
 b = seal balance
 k = pressure gradient factor (Table 32.1)

The mechanical pressure is given by

$$P_{sp} = \frac{F_{sp}}{a_o} \text{ lb/in}^2 \text{ (N/m}^2\text{)}$$

where F_{sp} = seal spring load, lb (N)
 a_o = seal face area, in² (m²)

Thus, the actual face pressure can be given as

$$P_f = \Delta p(b - k) + p_{sp}$$

This pressure is used to estimate the operating pressure and velocity of a given seal installation.

Liquid Sealed	<i>k</i>
Light-specific gravity liquids	0.3
Water-base solutions	0.5
Oil-base solutions	0.7

TABLE 32.1 Pressure Gradient Factors for Various Services

32.2.3 Pressure-Velocity

The seal is affected during operation by the following:

- Actual face pressure
- Rotational speed

The power per unit area is defined by

$$PV = \frac{N_f}{a_o}$$

The coefficient of friction is assumed to be unity in this case.

The equation for PV (pressure $\times$ velocity) is given by the following for seals:

$$PV = P_f V_m = [\Delta p(b - k) + P_{sp}] V_m$$

where V_m = velocity at the mean face diameter d_m , ft/min (m/s).

This PV is considered a measure of adhesive wear for a seal installation.

32.2.4 Power Consumption

The power loss at the seal is given by

$$N_f = (PV) f a_o \text{ ft} \cdot \text{lb/min (N} \cdot \text{m/s)}$$

where f is the coefficient of friction.

The power required to start the seal is normally around five times the running value. Table 32.2 provides the coefficients of friction for the most common seal face materials. These coefficients were developed using water at an operating PV of 35.03 bar $\cdot$ m/s

Sliding Materials		Coefficient of Friction
Rotating	Stationary	
Carbon-graphite (resin filled)	Cast iron	0.07
	Ceramic	0.07
	Tungsten carbide	0.07
	Silicon carbide	0.02
	Silicon carbide converted carbon	0.015
Silicon carbide	Tungsten carbide	0.02
Silicon carbide-converted carbon		0.05
Silicon carbide		0.02
Tungsten carbide		0.08

*Water lubrication, $PV = 100,000 \frac{\text{lb}}{\text{in}^2} \cdot \frac{\text{ft}}{\text{min}}$ (35.03 bar $\cdot$ m/s).

Source: John Crane-Houdaille, Inc.

TABLE 32.2 Coefficient of Friction for Various Seal Face Materials*

(100,000 lb/in² · ft/min). The coefficient values are slightly higher for oil. This is due to the viscous shear of the fluid film at the seal faces.

$$1 \text{ bar} = 10^5 \text{ Pa}$$

Example 32.1 A pump having a 2-in (50.8-mm)-diameter sleeve at the stuffing box is fitted with a balanced seal of this size and mean diameter. The seal operates in water at 300 lb/in² (20.68 bar), 3600 rpm, and ambient temperatures. The materials of construction are carbon and tungsten carbide. Determine the PV value and power loss of the seal.

Solution Given,

$$\Delta p = 300 \text{ lb/in}^2 \text{ (20.68 bar)}$$

$$b = 0.75$$

$$k = 0.5 \text{ (Table 32.1)}$$

$$d_m = 2 \text{ in (50.8 mm)}$$

$$P_{sp} = 25 \text{ lb/in}^2 \text{ (1.72 bar)}$$

$$V_m = \frac{\pi}{12} \times 2 \times 3600 = 1885 \text{ ft/min} \left(\frac{\pi \times 50.8 \times 3600}{1000 \times 60} = 9.57 \text{ m/s} \right)$$

$$a_o = 0.4 \text{ in}^2 \text{ (0.000258 m}^2\text{)}$$

$$f = 0.07 \text{ (Table 32.2)}$$

in USCS units

$$PV = [300(0.75 - 0.5) + 25](1885) = 188,400 \frac{\text{lb}}{\text{in}^2} \cdot \frac{\text{ft}}{\text{min}}$$

$$N_f = (188,400)(0.07)(0.4) = 5275 \frac{\text{ft} \cdot \text{lb}}{\text{min}} = 0.16 \text{ hp}$$

in SI units

$$PV = [20.68(0.75 - 0.5) + 1.72](9.57) = 66 \text{ bar} \cdot \text{m/s} = 66 \times 10^5 \text{ N/m}^2 \cdot \text{m/s}$$

$$N_f = (66 \times 10^5)(0.07)(2.58 \times 10^{-4}) = 119 \text{ N} \cdot \text{m/s} = 119 \text{ W}$$

32.2.5 Temperature Control

The wear of the seal faces is proportional to the temperature. Thus, the temperature control at the seal faces is desirable. Heat generated at the seal faces causes thermal distortion. This will increase the seal leakage. Therefore, most seal applications require cooling.

The temperature at the seal faces depends on the sum of the following terms:

- Heat generated by the seal
- Heat gained or lost to the pumpage

Mechanical power consumption of the seal is converted into heat. Thus,

$$Q_s = C_1 N_f = C_1 (PV f a_o)$$

where Q_s = heat input from the seal, Btu/h (W)
 $C_1 = 0.077$ for USCS units and 1 for SI units

The seal temperature will remain constant if the heat is generated at the same rate it is removed. However, the seal face temperature will increase if the amount of heat generated is more than that removed. Thus seal faces will be damaged if this temperature reaches a predetermined value. Figures 32.4 and 32.5 estimate the amount of heat generated by the unbalanced and balanced seal, respectively.

The seal flush removes the heat generated from the seal. This seal flush is normally provided from any of the following:

- A bypass from the discharge line of the pump
- An injection from an external source (e.g., a pressurizing pump)

The cooling flow rate is given by

$$\text{gpm (m}^3/\text{h)} = \frac{Q_s}{C_2(\text{sp. ht.})(\text{sp. gr.})\Delta T}$$

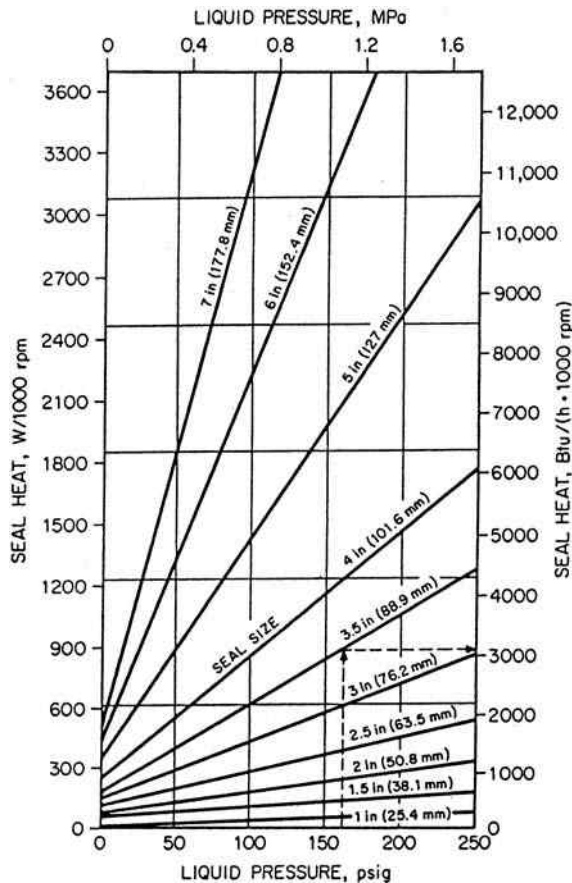


FIGURE 32.4 Unbalanced seal heat generation. (Source: John Crane-Houdaille, Inc.)

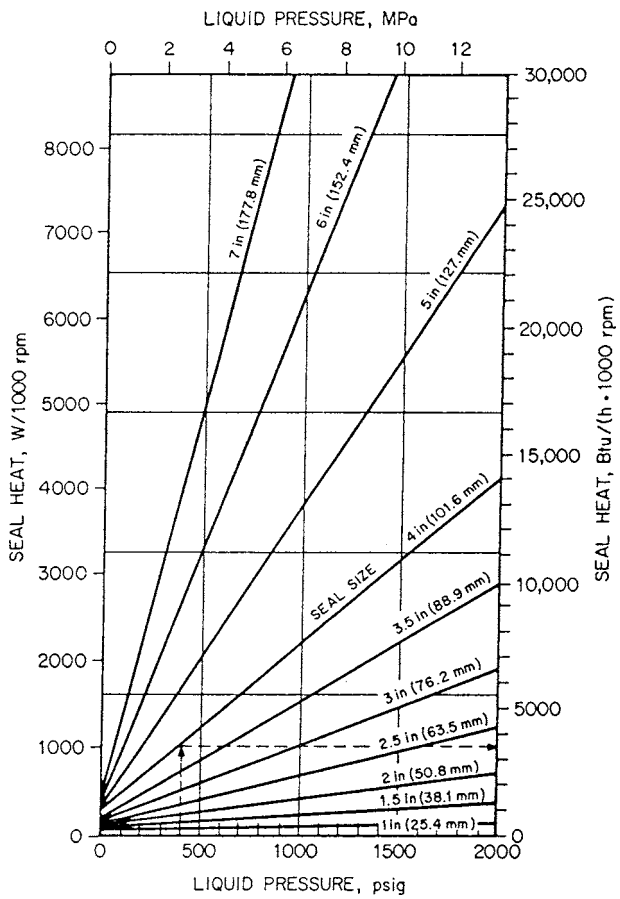


FIGURE 32.5 Balanced seal heat generation. (Source: John Crane-Houdaille, Inc.)

- where Q_s = seal heat, Btu/h (W)
- C_2 = 500 in USCS units and 1000 in SI units
- sp. ht. = specific heat of coolant, Btu/lb · °F (J/kg · K)
- sp. gr. = specific gravity of coolant
- ΔT = temperature rise, °F (K)

The heat added from the process must be considered if the liquid temperature is high. Thus,

$$Q_{\text{net}} = Q_p + Q_s$$

Figure 32.6 is used to determine heat load from the process (Q_p).

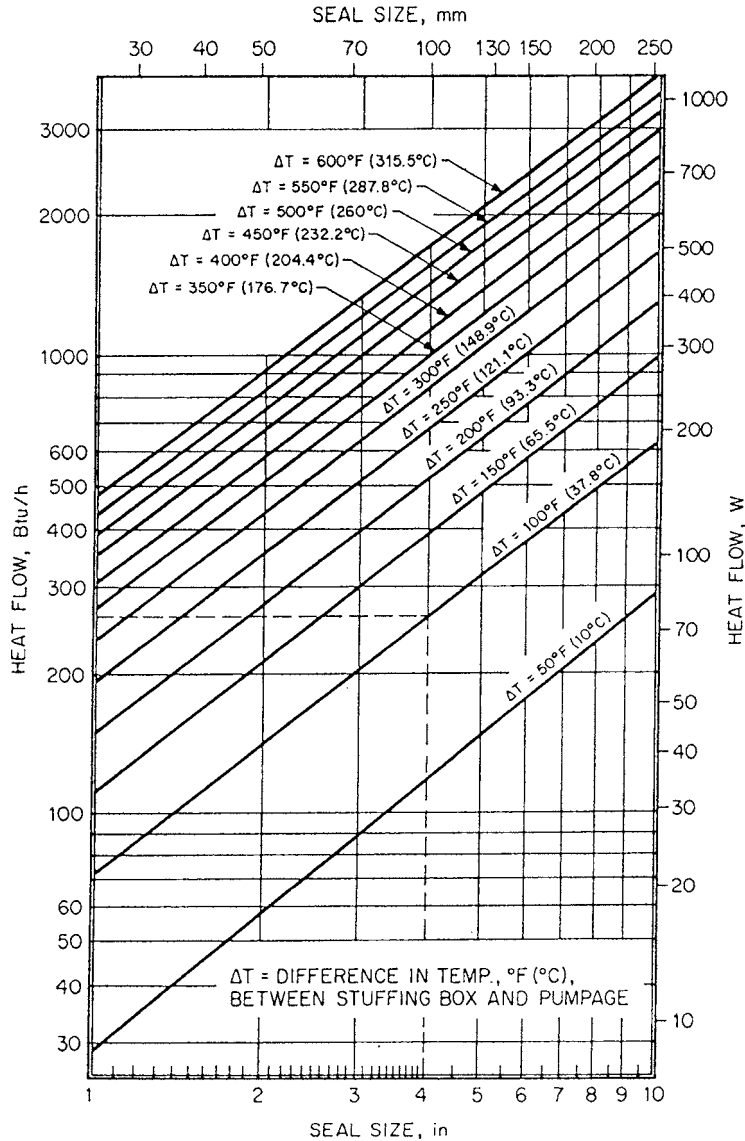


FIGURE 32.6 Heat soak from process when water is used for lubrication. (Source: John Crane-Houdaille, Inc.)

Example 32.2 Determine the net heat input for a 4-in (102-mm)-diameter balanced seal in water at 1800 rpm. Pressure and temperature are 400 lb/in² (2758 kPa) and 76.7 $^{\circ}\text{C}$ (170 $^{\circ}\text{F}$), respectively.

Solution

From Fig. 32.5, in USCS units

$$Q_s = (3500 \text{ Btu/h/1000 rpm}) \times 1800 = 6300 \text{ Btu/h}$$

$$Q_s = (1025 \text{ W/1000 rpm}) \times 1800 = 1845 \text{ W}$$

From Fig. 32.6, assuming that the stuffing box will be cooled to 21°C (70°F) and that the temperature difference between the stuffing box and pumpage is 37.8°C (100°F) in USCS units

$$Q_p = 255 \text{ Btu/h}$$

in SI units

$$Q_p = 75 \text{ W}$$

$$Q_{\text{net}} = 6555 \text{ Btu/h (1920 W)}$$

The total heat is used to determine the required cooling flow to the seal.

Figure 32.7 illustrates two methods used to supply cooling to the pump stuffing box. An internal flush connection at A is used to cool the seal when the liquid is clean. An external flush at B is used when the liquid is dirty. This method allows the flush (a bypass from the discharge line) to go through a filter or centrifugal separator. This flow flushes the seal faces with clean, cool liquid. Flashing can occur at the seal faces due to heat generation. This can be prevented by increasing the pressure of the flush.

The stuffing box requires additional cooling if the liquid is near the boiling point. Figure 32.8 illustrates this arrangement. This application is equipped with a pumping ring and heat exchanger. The pumping ring pumps the flow through the outlet piping. The liquid is cooled in the heat exchanger. The cooled flow cools the seal and the stuffing box. This method is very efficient. This is because the coolant flows only in the stuffing box. The temperature of the liquid in the pump is not affected in this case.

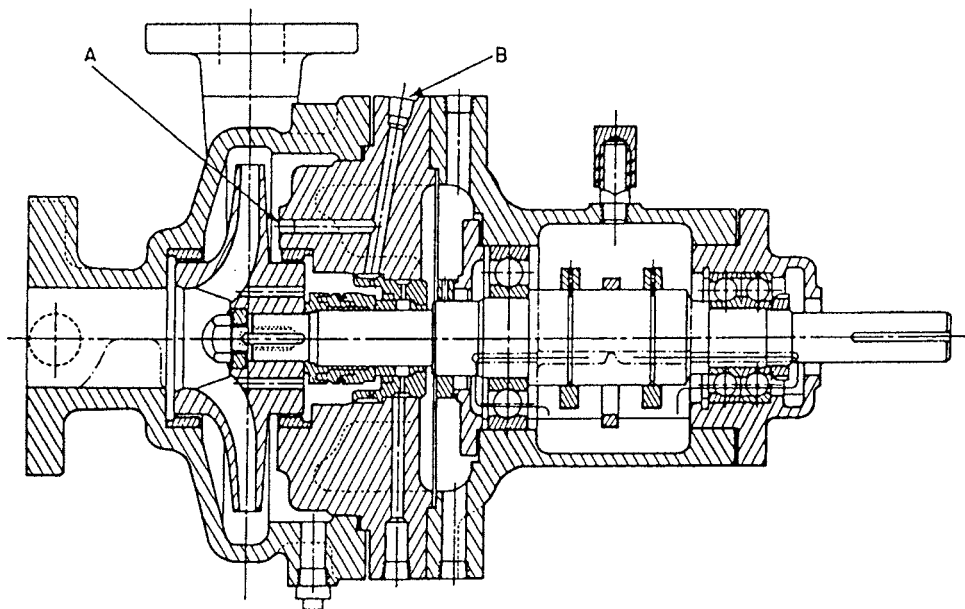


FIGURE 32.7 Cooling circulation to mechanical seal: (A) internal circulation plug port, (B) external circulation plug port.

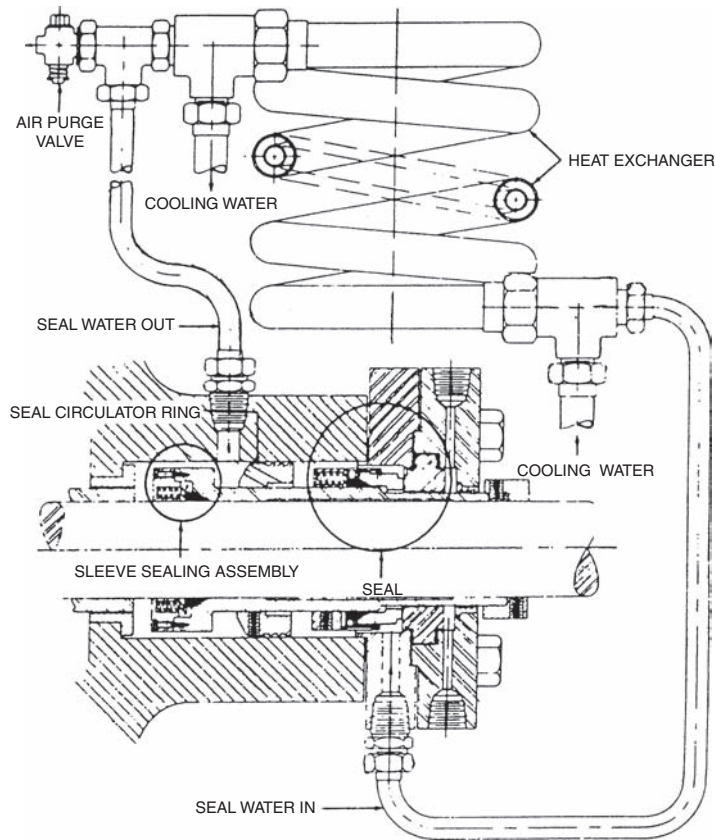


FIGURE 32.8 Mechanical seal with external cooling arrangement. (Source: John Carne-Houdaille, Inc.)

The pumping ring is installed on the outside diameter of the seal head (Fig. 32.9) in applications where the stuffing box has a small cross-sectional area. This ring is known as an axial-flow pumping ring. This is because it is a screw-type ring. This ring circulates the coolant in multiple-seal applications.

The following problems will occur during shutdown periods:

- The liquid may crystallize or solidify at ambient temperature. Damage to the seal faces may occur when pump is started in this case. The seal faces should be pre-heated before starting in these applications. This will prevent damage to the seal.
- Crevice corrosion may occur in the seal faces. This will impair the seal. The pump should be started at regular intervals to prevent damage to the seal.

32.2.6 Seal Lubrication/Leakage

The effective operation of the seal faces depends on the lubricating film between the seal faces. The seal leakage is the liquid that escapes from the film to the low pressure or atmosphere-side of the seal. The seal lubrication and leakage are affected by the following:

- Small distortions in the seal faces
- Vibrations transmitted to the seal faces

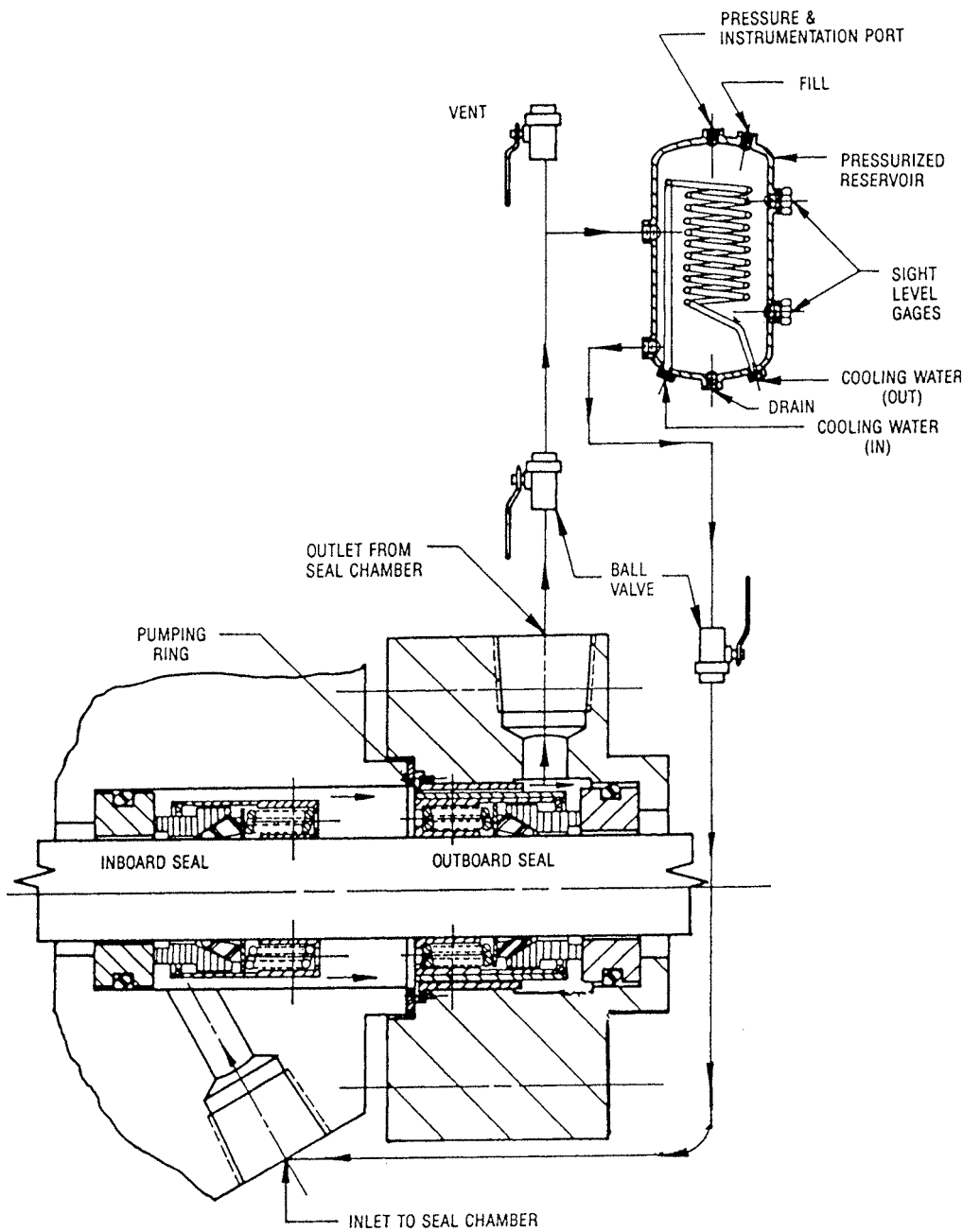
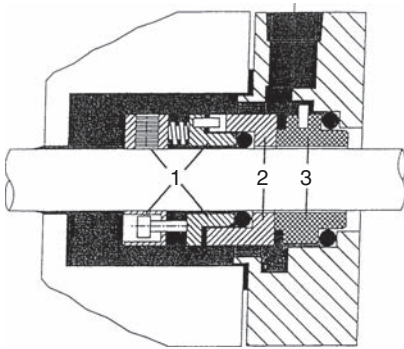
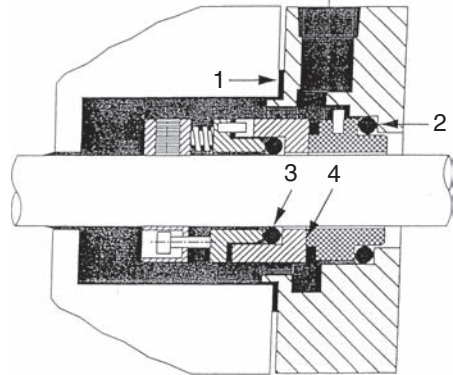


FIGURE 32.9 Double seal installation with axial-flow pumping ring. (Source: John Carne-Houdaille Inc.)



- Faces must be in contact.
- Faces must be flat to within 3 light bands (34.8 millionths of an inch).
Most are 1 light band (11.6 millionths).
- Faces must be lubricated by the liquid in the stuffing box.

FIGURE 32.10 Typical single inside pusher seal.



Leak Paths

- 1 Gland to pump face – sealed with a gasket
- 2 Over the top of the seal – o-ring
- 3 Under the seal – o-ring
- 4 Between the faces

FIGURE 32.11 Typical single inside pusher seal.

The seal faces have been lapped to precision flatness (Figs. 32.10 and 32.11). This flatness must be within 3 light bands (34.8 millionth of an inch). However, most seal faces are lapped to 1 light band (11.6 millionths of an inch). However, each seal develops natural surface waviness during fabrication (Fig. 32.12*a*). This surface waviness will grow during operation. The hydrodynamic film will develop between the sealing faces. Figure 32.12*b* illustrates the surface waviness during stable operation. This waviness is larger than the initial waviness during stable operation. This is due to the heat generated between the faces. The film minimizes the seal wear and power loss, which minimizes the leakage.

Patches of the sealing plane will break through the hydrodynamic film if the pressure on the seal faces increases. The seal will be operating in the thermoelastic region (Fig. 32.12*c*) at this stage. This will increase the seal leakage. This is because portions of the faces have broken through the film. The temperature of the faces will increase due to this change. These faces become bright red. This will carbonize, vaporize, or flash the liquid film. The increase in leakage will wear the seal parts. Flashing will also occur from the pump stuffing box. Flashing noise may be heard above the equipment as a sputtering sound. The spots will appear as depressions in the sealing planes (Fig. 32.12*d*) when the seal has cooled to room temperature. The incorporation of hydropads into the sealing planes will eliminate the seal difficulty from thermoelastic instability. This method is used by some seal manufacturers to cool the sealing surfaces (Fig. 32.13).

The seal leakage is affected by the following:

- Parallelism of the sealing planes
- Angular misalignment
- Coning (negative face rotation)
- Thermal distortion (positive face rotation)

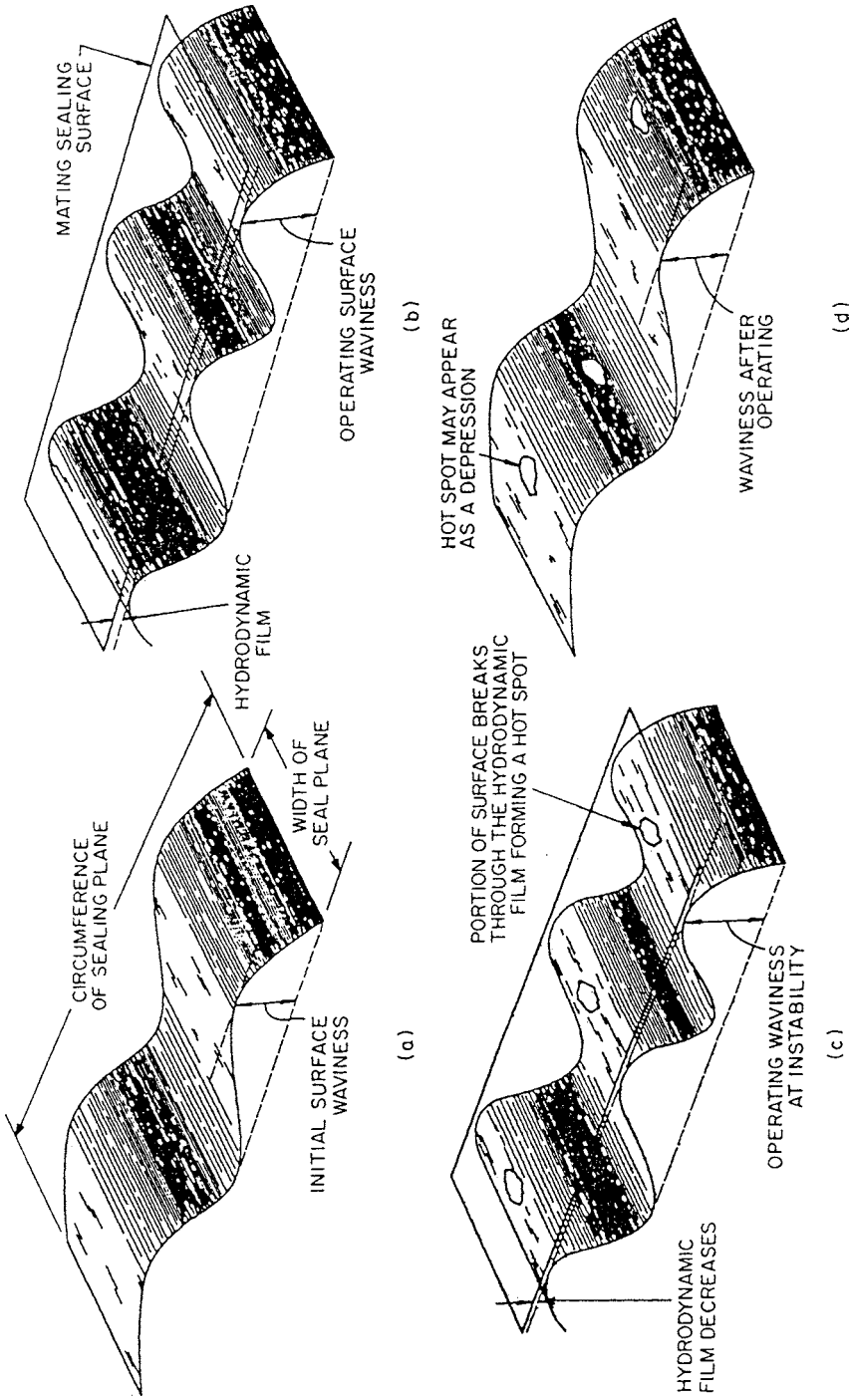


FIGURE 32.12 Mechanical seal face waviness for different conditions of operation. (a) New sealing plane, (b) during operation without surface instability, (c) during operation with instability, (d) after instability when sliding system comes to rest.

FIGURE 32.13 Primary ring with hydropads for face lubrication.
(Source: John Crane-Houdaille, Inc.)



- Shaft runout and whirl
- Axial vibration
- Fluctuation in pressure

The seal leakage in cubic centimeters per hour is estimated from

$$Q = -C_3 \frac{h^3(P_2 - P_1)}{\mu \ln(R_2/R_1)}$$

where $C_3 = 2.13 \times 10^{10}$ in USCS units and 1.88×10^9 in SI units

h = face gap, in (m)

P_2 = pressure at face ID, lb/in² (N/m²)

P_1 = pressure at face OD, lb/in² (N/m²)

μ = dynamic viscosity, cP (N · s/m²)

R_2 = outer face radius, in (m)

R_1 = inner face radius, in (m)

The flow from the face outer diameter (OD) to the inner diameter (ID) is indicated by negative leakage.

The gap between the seal faces depends on the following:

- Materials of the seal faces
- Flatness
- Liquid being sealed

This gap varies from 0.508×10^{-6} to 1.27×10^{-6} m (20×10^{-6} to 50×10^{-6} in).

Classification of Seals by Arrangement

The sealing arrangements are classified into the following:

1. Single seal installations
 - a. Internally mounted
 - b. Externally mounted
2. Multiple seal installations
 - a. Double seals
 - b. Tandem seals

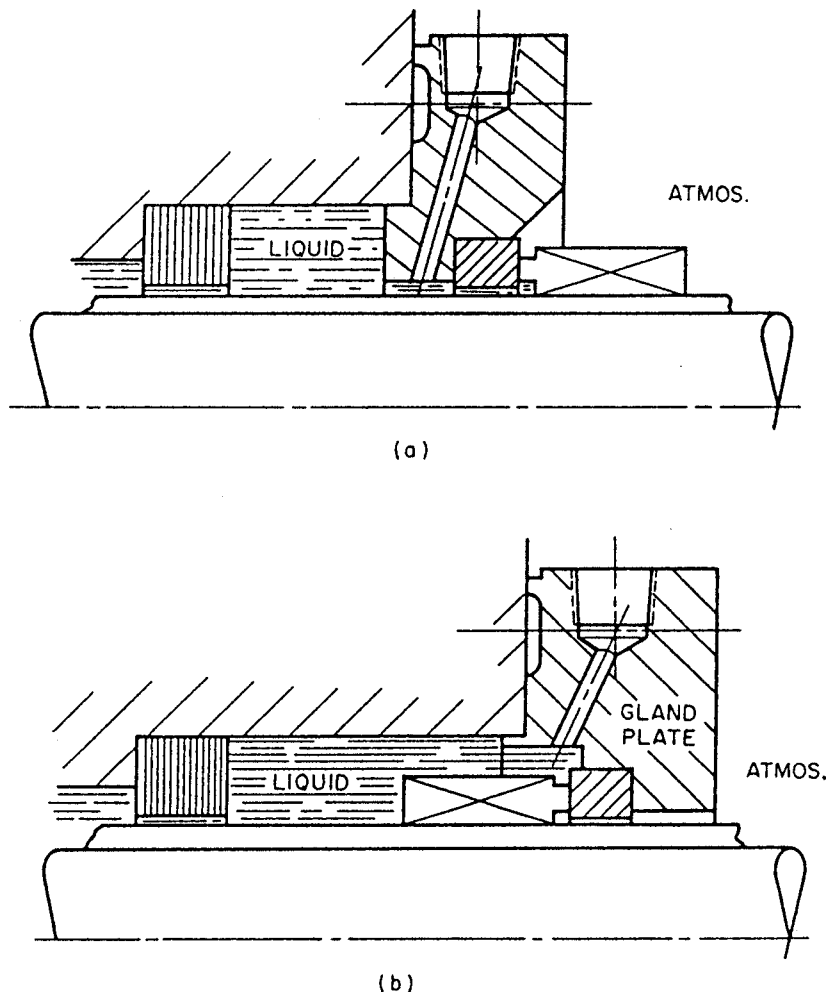


FIGURE 32.14 Single seal installations: (a) outside-mounted, (b) inside-mounted.

Most applications use *single seals*. This design has the least number of parts. Figure 32.14 illustrates the outside-mounted and the inside-mounted single seal installations.

These seals are installed outside and inside the pump stuffing box, respectively. The inside-mounted seal is the most common installation. The seal faces are kept in contact by the following:

- Liquid pressure (when the pump is pressurized)
- Spring force (when the pump is depressurized)

The spring force is much lower than the liquid pressure when the unit is pressurized.

The outside-mounted seals are normally used for low-pressure applications. This design is used normally to minimize corrosion of the seal faces in applications involving corrosive liquids.

Multiple seals are used in applications requiring the following:

- A neutral liquid for lubrication
- High corrosion resistance
- Higher reliability

Double seals consist of two back-to-back single seals (Fig. 32.15). The primary rings (rotating rings) face in opposite directions the mating rings (stationary ring). The neutral liquid is injected between the seals. This liquid lubricates the seal faces. The inboard seal (located on the left-hand side of Fig. 32.15) prevents the pumped liquid from entering the stuffing box. The lubricating liquid is maintained at a higher pressure than the pumped liquid. The inboard and outboard seals prevent the leakage of the neutral lubricating liquid.

Figure 32.16 illustrates double seals opposing each other. The primary rings rotate on a common mating ring. The neutral liquid circulates between the seals. The pressure

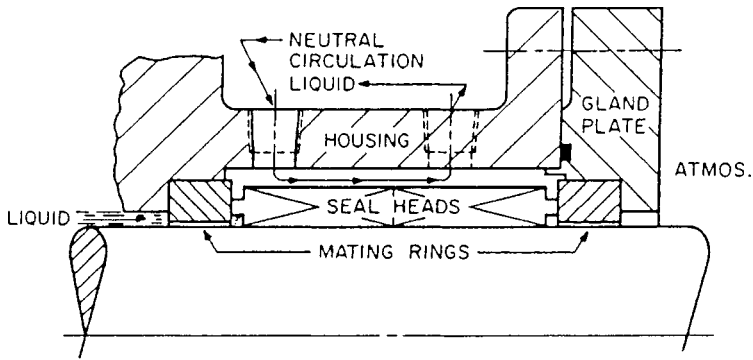


FIGURE 32.15 Double seals.

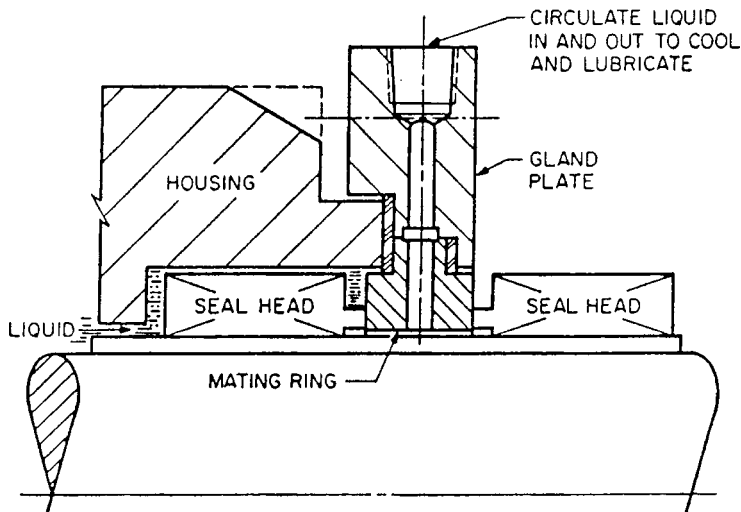


FIGURE 32.16 Opposed double seals.

of this liquid is lower than that of the process liquid. The inboard seal is similar to a single inside-mounted seal. This seal carries the differential pressure between the pump stuffing box and the neutral liquid. The outboard seal carries the differential pressure between the neutral liquid and atmosphere. The main advantage of this design is that its axial length is shorter than back-to-back double seals.

Tandem seals consist of two single seals mounted in the same direction (Fig. 32.17). The outboard seal creates a buffer zone between the inboard seal and atmosphere. The neutral lubricating liquid is normally maintained at atmospheric pressure. The inboard seal carries the differential pressure between the liquid being pumped and atmosphere. The neutral circulating liquid removes the heat from the seals. Tandem seals are used in applications involving toxic, flammable, and radioactive liquids. These applications require high reliability.

Package seals do not require any special measurements prior to installation (Fig. 32.18). These seals consist of the gland plate, sleeve, and drive collar. Most of these seals include a spacer. The purpose of this component is to set the seal faces properly. This spacer is removed after the following:

- The drive collar has been locked to the shaft.
- The gland plate has been bolted to the pump.

Classification of Seals by Design

The following are the seal categories:

- Unbalanced or balanced
- Rotating or stationary seal head
- Single- or multiple-spring design
- Pusher or nonpusher secondary seal

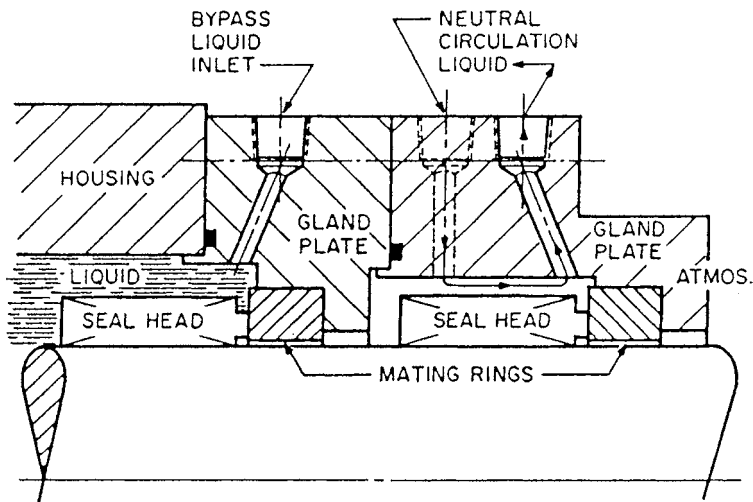


FIGURE 32.17 Tandem seals.

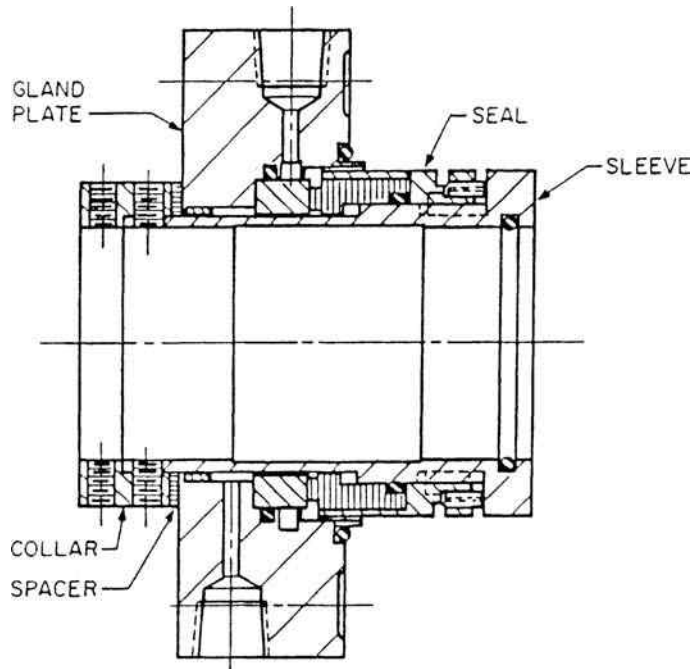


FIGURE 32.18 Single-package seal assembly. (Source: John Crane-Houdaille, Inc.)

The pressure in the pump stuffing box determines the selection of a balanced or unbalanced seal. Balanced seals control the following:

- The contact pressure between the seal faces
- Power generated by the seal

Unbalanced seals have a balance b (ratio of hydraulic closing area to seal face area) greater or equal to 100%. Balanced seals have a balance b less than 100%. Figure 32.19 illustrates the most common balanced and unbalanced seals.

The shaft speed determines the selection of a rotating or stationary seal. The primary ring rotates with the shaft in a rotating seal assembly (Figs. 32.17, 32.18, and 32.19). The mating ring is stationary in this design. However, the matching ring rotates with the shaft in a stationary seal assembly (Fig. 32.20). The rotating seals are used for normal shaft speeds. Stationary seals are used when the shaft speed exceeds 25.4 m/s (5000 ft/min or 5000 rpm for a 5-cm shaft diameter). This design is used on these applications to keep unbalanced forces. This is because they minimize seal vibrations. The space limitations and the liquid being sealed determine the selection of a single-spring or multi-spring seal head. Bellows seals use a single spring (Fig. 32.21a). This spring loads the seal faces. The advantage of this design is that clogging will not occur in the seal. The spring coils can withstand highly corrosive environments. This is because they are made of a large diameter wire.

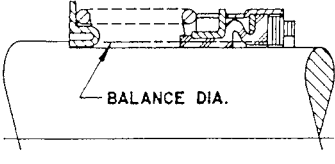
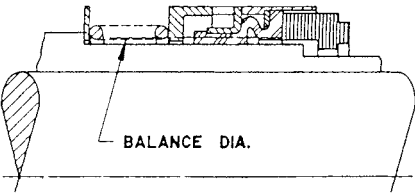
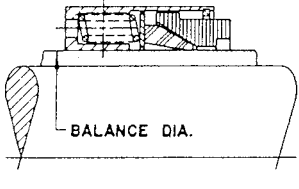
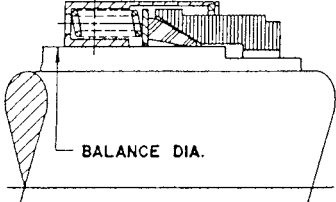
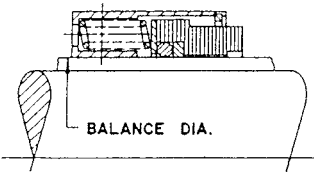
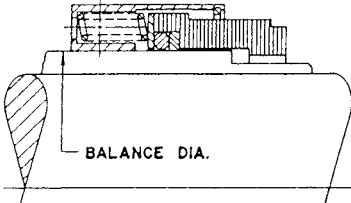
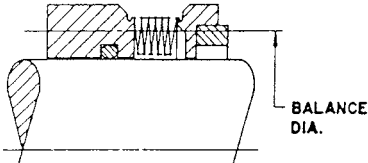
TYPE OF SEAL	TYPE OF BALANCE	
	UNBALANCED $b = 100\% \text{ OR } > 100\%$	BALANCED $b < 100\%$
FULL CONVOLUTION BELLOW SEAL		
WEDGE SEAL		
O RING SEAL		
MULTIPLE CONVOLUTION METAL BELLOWS SEAL	METAL BELLOWS ONLY AVAILABLE IN BALANCE CONFIGURATION	
		

FIGURE 32.19 Common unbalanced and balanced seals.

Some seals use multiple springs (Fig. 32.21*b*). This design requires a shorter axial space. The springs are placed are placed around the circumference of the shaft. These seals use normally o-ring or wedge as secondary seals.

The mechanical load of the seal and the hydraulic pressure in the stuffing box move the secondary seal of the pusher-type seal (Fig. 32.22). These seals use the following:

- O-ring
- Wedge
- V-ring

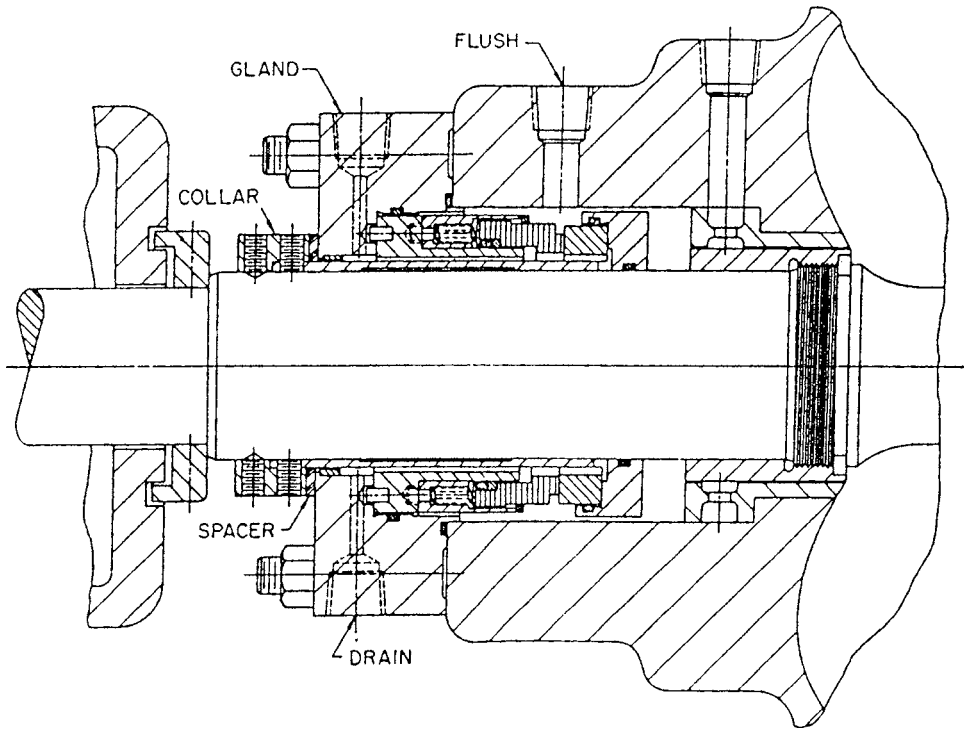
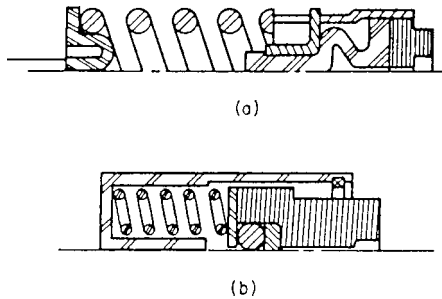


FIGURE 32.20 Stationary seal with rotating mating ring. (Source: John Crane-Houdaille, Inc.)

FIGURE 32.21 Comparison of (a) single-spring and (b) multiple-spring seals.



The gland plate and its mating ring are in the pump stuffing box. This is the stationary component of the seal assembly. The following are the sealing points in this design:

- Gasket between the gland plate and face of the stuffing box
- Surface contact between the mating ring and the primary ring
- O-ring between the primary ring and shaft or sleeve

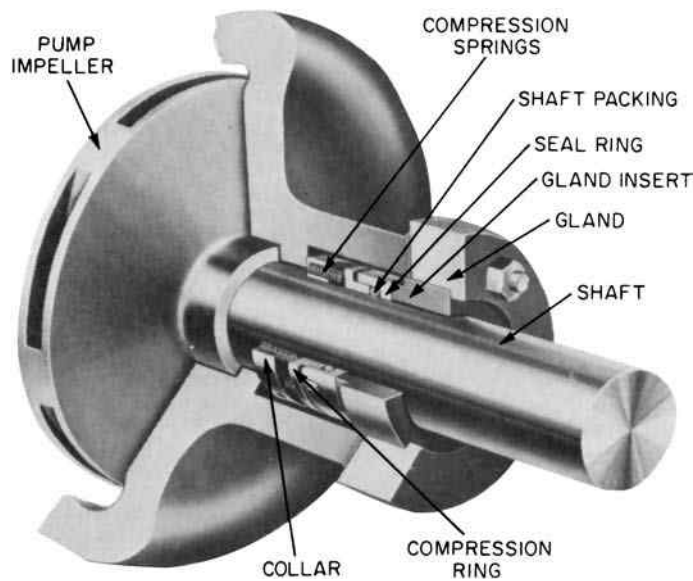


FIGURE 32.22 O-ring-type mechanical seal. (Source: Durametallic.)

The primary ring (rotating) has a hardened metal face. The compression ring holds this ring against the stationary ring through loading of the o-ring. The compression ring supports a series of springs. These springs are connected at the opposite end by a collar. This collar is fixed to the shaft. The primary ring is flexibly mounted. This is done to accommodate any shaft deflection or equipment vibration.

Figure 32.23 illustrates another pusher-type seal. The following are the components of this seal:

- F = Primary ring (rotating)
- A = A component that locks the metal retainer to the shaft

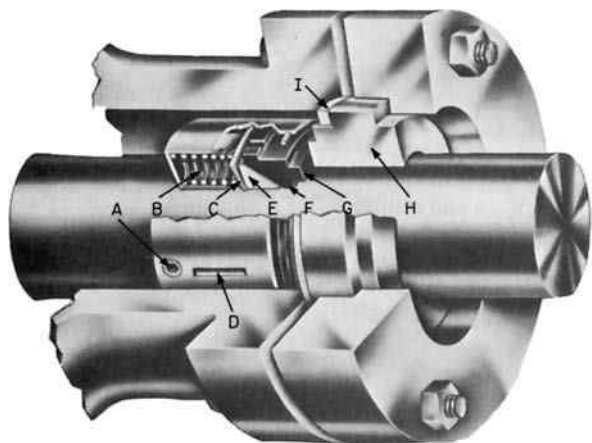


FIGURE 32.23 Wedge-type mechanical seal. (Source: John Crane-Houdaille, Inc.)

- D = Drive dents
- E = Wedge (seal between the primary ring and the shaft or sleeve)
- B = Multiple springs
- C = Mental disc (distributes a spring load evenly)
- G = Primary ring (rotating)
- H = Mating ring (stationary)
- I = pump casing

This design is selected when elastomers cannot be used in the seal. A wedge made from Teflon or Grafoil is used instead of the elastomer. Component A locks the metal retainer to the shaft to the shaft. This retainer provides positive drive to the primary ring F through drive dents D. These dents fit corresponding grooves. Wedge E constitutes the seal between the primary ring and shaft or sleeve. This wedge is preloaded by multiple springs B. Disc C distributes the spring load uniformly. The dynamic seal consists of the primary ring G, the mating ring H, and pump casing (I).

Figure 32.24 illustrates a different pusher seal. This external split seal is commonly used in applications involving sewage or slurries. The main seal parts are located outside the stuffing box. This permits the replacement of the seal members without dismantling the pump or installing new shaft sleeves. The gland sleeve (1) fits inside the stuffing box (2). The gland sleeve o-ring (3) seals the gland sleeve against the stuffing box. The split end seal (4) is fitted into the gland sleeve. Gland nut (5) holds the seal in place. The antirotation stud (6) passes through the guide lugs (7). This assembly moves axially in the stuffing box. The axial movement depends on the force exerted by the adjustable springs (8) on the yoke plate (9). The springs act on the gland nut and abutment plate. The springs force the end seal against the cone (10). This cone rotates with the shaft. O-ring (11) seals the cone. This O-ring provides the running seal.

The secondary seal is not pushed along the shaft by the mechanical load or hydraulic pressure in a nonpusher seal. The bellows convolution accommodates all the movement in this design. The definition presented above applies to seals that use half-, full-, and multiple-convolution bellows as a secondary seal.

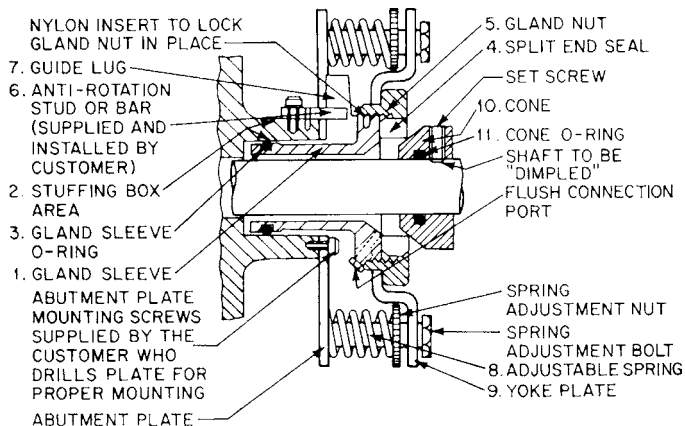


FIGURE 32.24 External mechanical seal for slurry or sewage service. (Source: Johns-Manville.)

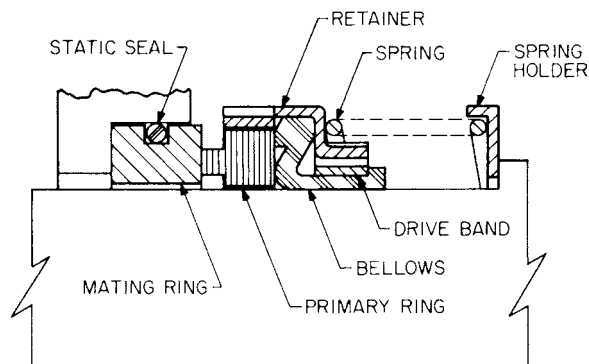


FIGURE 32.25 Half-convolution bellows seal. (Source: John Crane-Houdaille, Inc.)

Figure 32.25 illustrates a half-convolution bellows seal. These seals are always made of an elastomer. The drive band holds the tail end of the bellows to the shaft. These bellows seal the shaft. They also allow the unit to rotate with the shaft. This design is used normally in light-duty service applications.

Figure 32.6 illustrates a full-convolution bellows seal. The drive band holds the tail end of the bellows to the shaft. The drive for the seal assembly in this design is similar to the half-convolution seal. Heavier full-convolution bellows seals are also available. These seals tolerate larger shaft motion and runout up to a pressure of 8.28 MPa (1200 psi).

Teflon and metals require multiple-convolution bellows seal design. This is because of the mechanical characteristics of these materials. This design is necessary to add flexibility to this material that cannot be used in any other shape.

Figure 32.27 illustrates a Teflon bellows seal assembly. These seals are mounted outside the stuffing box. This is because of the large cross-sectional area. Hydraulic pressure is applied to the outside diameter of the seal. This pressure helps keep the faces closed. Small springs are installed on the atmosphere side of the seal. The springs provide the force required to keep the seal faces closed when the pump is depressurized.

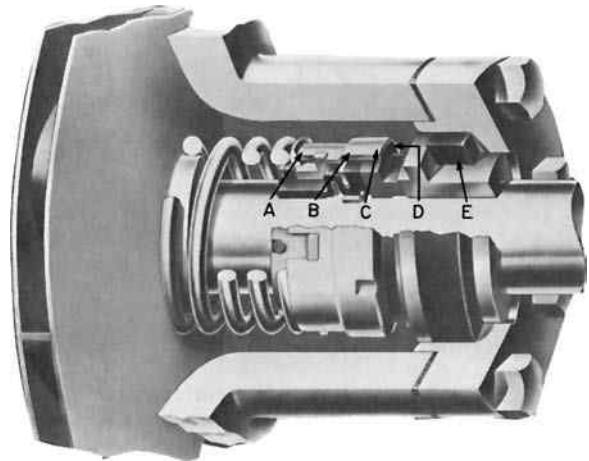
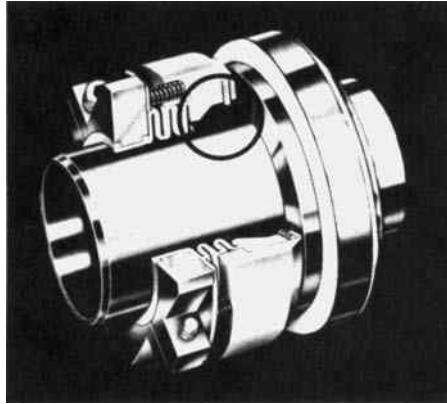


FIGURE 32.26 Full-convolution bellows seal. (Source: John Crane-Houdaille, Inc.)

FIGURE 32.27 Teflon bellows seal. (Source: John Crane-Houdaille, Inc.)

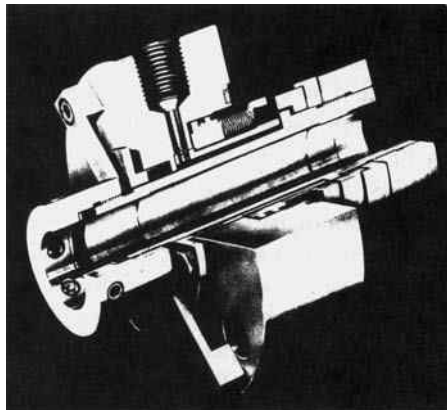


There is a variety of metals bellows seal design. The seal head is made from a metal in this design. A carbon or tungsten carbide insert is mounted on the seal face. This insert is shrunk to fit to the assembly. Figure 32.28 illustrates a stationary metal bellows seal. This design has a rotating mating ring. The seal head is mounted in the gland ring. This component is held in place by cap screws. This design employs high-temperature elastomers and Grafoil. This extends the temperature limit of the seal to 400°C (750°F). The seals having rotating mating rings are preferred in refinery service. The liquid to the seal must be at least 25°C (50°F) below its atmospheric boiling point. This is required to increase the longevity of the seal. Nitrogen gas or steam quench should be used at the atmosphere side of the seal in applications having a temperature of 232°C (450°F). This is required to prevent carbonization of the liquid being sealed.

Common Types of Mating Rings

The mating ring is a replaceable component of the seal assembly. This is because its face wears out during operation. Figure 32.29 illustrates common mating assemblies. The material of the static seal determines the temperature limit of the assembly. The mating ring is normally made from ceramic or a hard metal.

FIGURE 32.28 Stationary metal bellows seal. (Source: John Crane-Houdaille, Inc.)



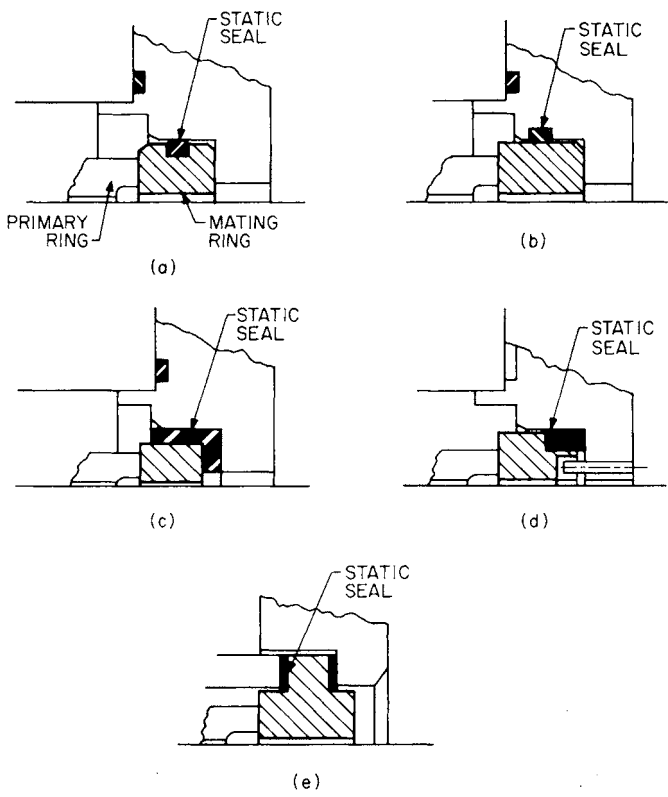


FIGURE 32.29 Common mating ring assemblies: (a) grooved o-ring, (b) square section, (c) cup-mounted, (d) floating, (e) clamped-in.

Materials of Construction

All components of the seal are selected based on their corrosion resistance to the liquid being sealed. NACE (National Association of Corrosion Engineers) have prepared a corrosion handbook. This handbook provides the corrosion rate of many seal materials used with a wide variety of liquids and gases. Double seals that keep the seal components in a neutral liquid should be selected in applications having a corrosion rate greater than 0.05 mm (2 mils) per year. This is done to reduce the corrosion. The inside diameters of the mating ring, primary ring, and secondary seal are exposed to the corrosive liquid in this design. These components should be constructed from corrosion-resistant materials. This material includes ceramic, carbon, and Teflon. Table 32.3 provides the common material of construction. Table 32.4 shows the properties of most seal face materials.

The design of the secondary and static seal depends mainly on the operating temperature. Table 32.5 provides the temperature limits for common secondary and static seal materials.

The *PV* (pressure $\times$ velocity) is the second consideration in the selection of the primary and mating ring materials. This parameter indicates the capability of the material combination to resist adhesive wear. This is the dominant wear mechanism in mechanical

Component	Material of Construction
Secondary seals	
O-ring	Nitrile, ethylene, propylene, chloroprene
Bellows	Nitrile, ethylene, propylene, chloroprene
Wedge	Fluorocarbon resin, Grafoil*
Metal bellows	Stainless steel, nickel-base alloy
Primary ring	Carbon, metal-filled carbon, tungsten carbide, silicon carbide, siliconized carbon, bronze
Hardware (retainers, disk, snap ring, set screws, springs)	18-8 stainless steel, 316 stainless steel, nickel base alloys, titanium
Mating ring	Ceramic, cast iron, tungsten carbide, silicon carbide

*A registered trademark of Union Carbide.

TABLE 32.3 Common Materials of Construction for Mechanical Seals

seals. Table 32.6 provides the limiting PV values for various face combinations. These values have been established for a wear rate that provides an equivalent seal life of 2 years. The PV value of an application can be compared with the limiting value shown in Table 32.6 to determine satisfactory service. These values apply to aqueous solutions at 49°C (120°F). Values 60% higher should be used for lubricating liquids.

Gland Plates and Piping Arrangements

The selection of the gland plate and associate piping is critical to the success or failure of a seal installation. The main purpose of the gland plate (or end plate) is to hold the mating ring assembly in the pump. This plate is also a pressure-containing component of the seal assembly. The fit of the gland plate to the stuffing box affects the alignment of the sealing surfaces. The American Petroleum Institute (API) specification requires a register fit with the inside or outside diameter of the stuffing box. This is required to ensure proper installation. The gland plate at the stuffing box must also confine the static seal completely. Figure 32.30 illustrates the three basic gland plate designs.

The *plain* gland plate is used on the following applications:

1. Seal cooling is supplied internally through the pump stuffing box.
2. The liquid being sealed is not hazardous.
3. The liquid being sealed will not crystallize or carbonize at the atmospheric side of the seal.

The *flush* gland plate is used in the following applications:

- Internal cooling is not available.
- The coolant (liquid being sealed or liquid from an external source) is directed to the seal faces to remove the heat.

Property	Cast Iron	Ni-Resist	Ceramic		Carbides		Carbon			
			85% (Al ₂ O ₃)	99% (Al ₂ O ₃)	Tungsten (6% Co)	Silicon Si-C	Resin	Babbitt	Bronze	Si-C Conv.
Modulus of elasticity, x10 ⁶ lb/in ² (x10 ³ MPa)	13–15.95 (90–110)	10.5–16.9 (72–117)	32 (221)	50 (245)	90 (621)	48–57 (331–393)	2.5–4.0 (17.2–27.6)	1.04–4.1 (7.2–28.3)	2.9–4.4 (20–30)	2–2.3 (13.8–15.9)
Tensile strength, x10 ³ lb/in ² (MPa)	65–120 (448–827)	20–45 (138–310)	20 (138)	39 (269)	123.25 (850)	20.65 (142)	4.5–9 (31–62)	8–8.6 (55–59)	7.5–9 (52–62)	2 (14)
Coefficient of thermal expansion x10 ⁻⁶ , in/in · °F (cm/cm · K)	6.6 (11.88)	6.5–6.8 (11.7–12.24)	3.9 (7.02)	4.3 (7.74)	2.53 (4.55)	1.88 (3.38)	2.3–3.4 (4.14–6.12)	2.1–2.7 (3.78–4.86)	2.4–3.1 (4.32–5.58)	2.4–3.2 (4.32–5.76)
Thermal conductivity, Btu · ft/h · ft ² · °F (W/m · K)	23–29 (39.79–50.17)	25–28 (43.25–48.44)	8.5 (14.70)	14.5 (25.08)	41–48 (70.93–83.04)	41–60 (70.93–103.8)	3.8–12 (6.57–20.76)	6–9 (10.38–15.57)	8–8.5 (13.84–14.70)	30 (51.9)
Density, lb/in ³ (kg/m ³)	0.259–0.268 (7169–7418)	0.264–0.268 (7307–7418)	0.123 (3405)	0.137 (3792)	0.59 (16,331)	0.104 (2879)	0.064–0.069 (1771–1910)	0.083–0.112 (2297–3100)	0.083–0.097 (2297–2685)	0.067–0.070 (1854–1938)
Hardness	Brinell		Rockwell A		Rockwell 45N		Shore			
	217–269	131–183	87	87	92	86–88	80–105	60–95	70–92	Rockwell 15T 90

TABLE 32.4 Properties of Common Seal Face Materials

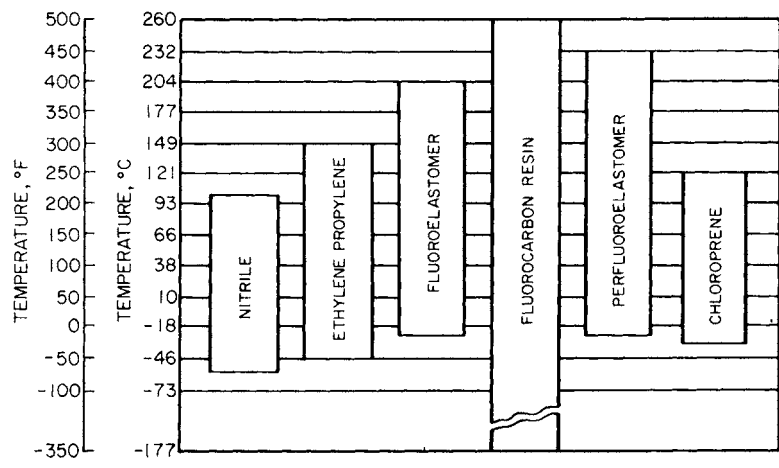


TABLE 32.5 Temperature Limits of Secondary Seal Materials

Sliding Materials		PV Limit, $\frac{\text{lb}}{\text{in}^2} \cdot \frac{\text{ft}}{\text{min}}$ (bar · m/s)	Comments
Rotating	Stationary		
Carbon-graphite	Ni-resist	100,000 (35.03)	Better thermal shock resistance than ceramic
	Ceramic (85% Al ₂ O ₃)	100,000 (35.03)	Poorer thermal shock resistance and much better corrosion resistance than Ni-resist
	Ceramic (99% Al ₂ O ₃)	100,000 (35.03)	Better corrosion resistance than 85% Al ₂ O ₃ ceramic
	Tungsten carbide (6% Co)	500,000 (175.15)	With bronze-filled carbon-graphite, PV is up to 100,00 $\frac{\text{lb}}{\text{in}^2} \cdot \frac{\text{ft}}{\text{min}}$ (35.03 bar · m/s)
	Tungsten carbide (6% Ni)	500,000 (175.15)	Ni binder for better corrosion resistance
	Silicon carbide converted carbon	500,000 (175.15)	Good wear resistance; thin layer of Si-C makes relapping questionable
	Silicon carbide (solid)	500,000 (175.15)	Better corrosion resistance than tungsten carbide but poorer thermal shock resistance

TABLE 32.6 Frequently Used Seal Face Materials and Their PV Limitations

Sliding Materials		PV Limit, $\frac{\text{lb}}{\text{in}^2} \cdot \frac{\text{ft}}{\text{min}}$ (bar · m/s)	Comments
Rotating	Stationary		
Carbon-graphite		50,000 (17.51)	Low PV , but very good against face blistering
Ceramic		10,000 (3.50)	Good service on sealing paint Pigments
Tungsten carbide		120,000 (42.04)	PV is up to 185,000 $\frac{\text{lb}}{\text{in}^2} \cdot \frac{\text{ft}}{\text{min}}$ (64.8 bar · m/s) will two grades that have different % of binder
Silicon carbide converted carbon		500,000 (175.15)	Excellent abrasion resistance; more economical than solid silicon carbide
Silicon carbide (solid)		500,000 (175.15)	Excellent abrasion resistance, good corrosion resistance, and moderate thermal shock resistance

TABLE 32.6 Frequently Used Seal Face Materials and Their PV Limitations (*Continued*)

The *flush-and-quench* gland plate is used in the following application: seals requiring direct cooling and a quenching fluid at the atmospheric side of the seal.

The quench fluid could be a liquid, gas, or steam. This fluid prevents the buildup of any carbonized or crystallized material along the shaft. The seal quench can increase the longevity of the seal. This is because it eliminates the loss of seal flexibility due to hang-up (formation of crystals, etc.).

The *flush, vent, and drain* gland plate provides the following features:

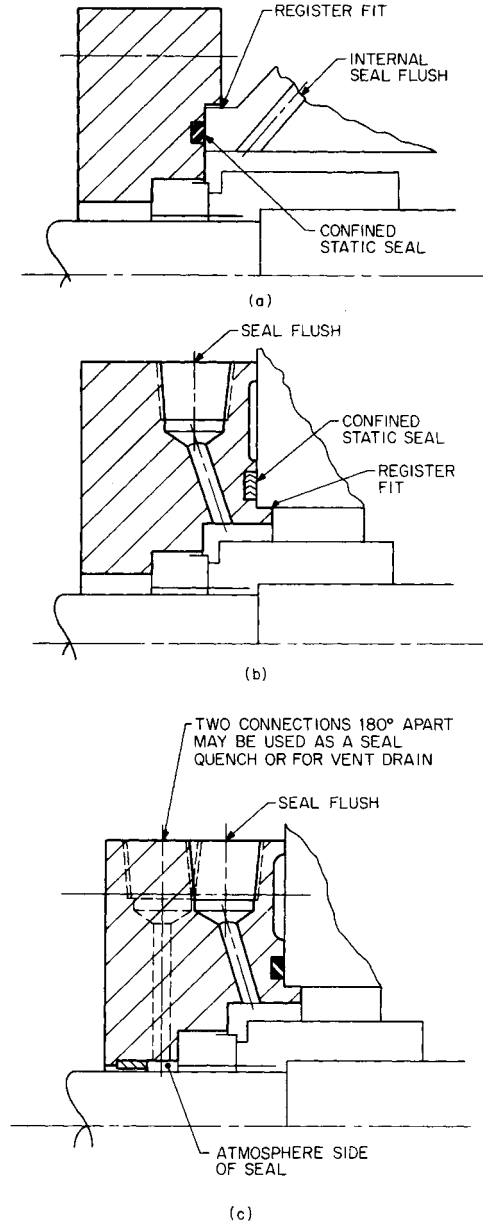
- 1. Control of seal leakage.
- 2. Venting and burning off flammable vapors leaking from the seals to a flair
- 3. Directing nonflammable liquid leakage to a safe ramp

Figure 32.31 illustrates restrictive devices used with quench, or vent-and-drain, gland plates. Figure 32.31*a* illustrates a bushing that can be pressed in place. Figure 32.31*b* and 32.31*c* illustrates bushings allowed to float. These floating bushings permit closer running fits with the shaft. This is because bushings are not restricted at the outside diameter. Figure 32.31*d* illustrates small packing rings. These designs can be used for a seal quench.

Installation and Troubleshooting of Mechanical Seals

The operation of the pump within the manufacture’s specifications is essentials for a successful seal installation. Figure 32.32 illustrates the types of motion that affect the seal performance. Relative movement between the seal components or

FIGURE 32.30 Basic gland plate designs: (a) plain gland plate, (b) flush gland plate, (c) quench, or vent-and-drain, gland plate.



shaft sleeve indicates that mechanical motion has been transmitted to the seal from the following:

1. Misalignment (angular or parallel)
2. End play
3. Radical runout of the pump

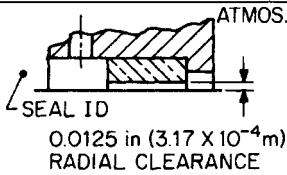
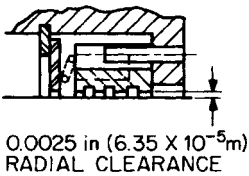
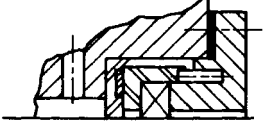
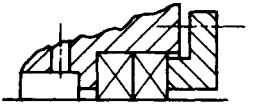
DESCRIPTION		COMMENTS
(a) THROTTLE BUSHING	 SEAL ID 0.0125 in (3.17 X 10 ⁻⁴ m) RADIAL CLEARANCE ATMOS.	USED ON VENT-AND-DRAIN DESIGNS OF NONSPARKING MATERIALS. MEETS API SPECIFICATION.
(b) FLOATING THROTTLE BUSHING	 0.0025 in (6.35 X 10 ⁻⁵ m) RADIAL CLEARANCE	MAY BE USED ON QUENCH AND VENT-AND-DRAIN GLANDS. SPRING LOADED TO FLOAT WITH SHAFT. MADE OF NONSPARKING MATERIALS. MEETS API SPECIFICATION. REQUIRES MORE SPACE THAN FIXED BUSHING.
(c) FLOATING BUSHING		SOFT PACKING SIZED TO SHAFT DIAMETER USED ON QUENCH AND VENT-AND-DRAIN GLANDS. SPRING LOADED. NO ADJUSTMENTS REQUIRED. EXCELLENT DRY RUN.
(d) PACKING RINGS		CREATES POSITIVE SEAL ON QUENCH DESIGNS. REQUIRES SOME ADJUSTMENT DURING OPERATION.

FIGURE 32.31 Common restrictive devices used with quench, or vent-and-drain, gland plates.

Angular misalignment occurs when the mating ring is not perpendicular to the shaft. This misalignment results in excessive vibrations of the internal components of the seal. This causes fretting of the sleeve or seal hardware on pusher-type seals. The distortion of a stuffing box by pipe strain will also cause misalignment. This problem will also damage the wearing rings. The improper alignment of the stuffing box with the rest of the pump causes parallel misalignment. Seal vibrations occur when the shaft strikes the inside diameter of the mating ring. Axial end play will result in fretting and damage to the seal faces. This is because the seal is being loaded and unloaded continuously. The seal faces will be damaged by heat checking (thermal damage) in this case. This damage will appear as radial lines on the seal faces. Radial runout will also cause seal vibrations. This will shorten the life of the seal. The seal drawings and instructions should be reviewed to determine the installation dimension or spacing. This is required to confirm that the seal is at the proper working height (Fig. 32.33). The cleanliness of the seal installation is critical for adequate operation. This is because the seal has precision-lapped faces.

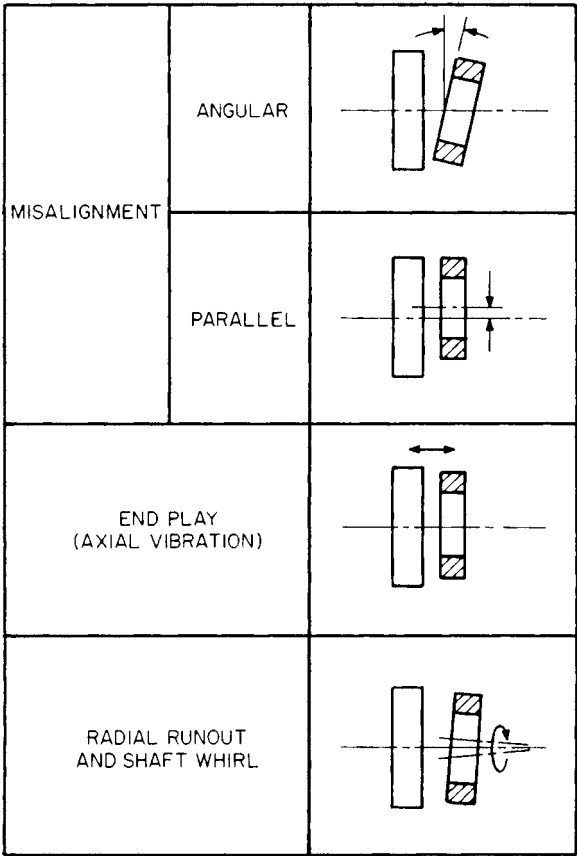


FIGURE 32.32 Common types of motion that influence seal performance.

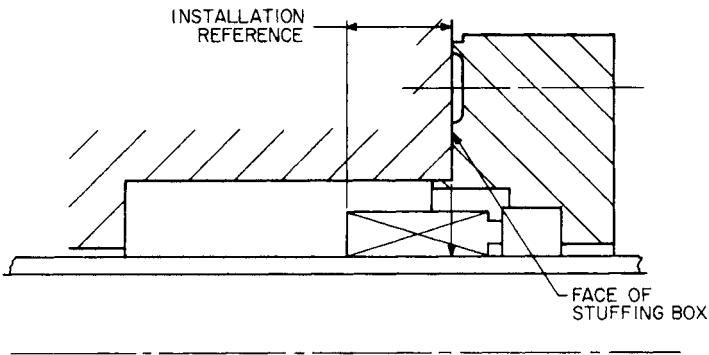


FIGURE 32.33 Typical installation reference dimensions.

There should be no static leakage from a proper installation of a mechanical seal. The dynamic leakage should range from zero to a few drops per minute. Under full vacuum, mechanical seals are used to prevent air ingress into the pump. The cause of excessive leakage should be indentified and corrected. Table 32.7 indentifies the causes of seal leakage with possible corrections. Figure 32.34 illustrates the common failure modes of mechanical seals.

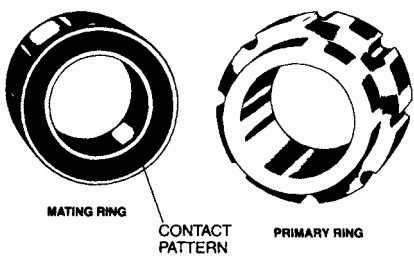
Symptom	Possible Causes	Corrective Procedures
Seal spits and sputters (“face popping”) in operation	Seal fluid vaporizing at seal interface	Increase cooling of seal faces Check seal proper seal balance with seal manufacturer Add bypass flush line if not in use Enlarge bypass flush line and/or orifices in gland plate Check for seal interface cooling with seal manufacturer
Seal drips steadily	Faces not flat Carbon graphite seal faces blistered Seal faces thermally distorted	Check for incorrect installation dimensions Check for improper materials or seals for the application Improve cooling flush lines Check for gland plate distortion due to overtorquing of gland bolts Check gland gasket for proper compression Clean out foreign particles between seal faces; relap faces in necessary Check for cracks and chips at seal face; replace primary and mating rings
	Secondary seals nicked or scratched during or scratched during installation O-rings overaged Secondary seals hard and brittle from compression set Secondary seals soft and sticky from chemical attack	Replace secondary seals Check for proper lead in chamfers, burrs, etc. Check for proper seals with seal manufacturer Check with manufacturer for other material
	Spring failure Hardware damaged by erosion Drive mechanisms corroded	Replace parts Check with seal manufacturer for other material

TABLE 32.7 Checklist for Identifying Causes of Seal Leakage

Symptom	Possible Causes	Corrective Procedures
Seal squeals during operation	Amount of liquid inadequate to lubricate seal faces	Add bypass flush line if not in use Enlarge bypass flush line and/or orifices in gland plate
Carbon dust accumulates on outside of gland ring	Amount of liquid inadequate to lubricate seal faces Liquid film evaporating between seal faces	Add bypass flush line in not in use Enlarge bypass flush line and/or orifices in gland plate Check for proper seal design with seal manufacturer if pressure in stuffing box is excessively high
Seal leaks	Nothing appears to be wrong	Refer to list under "Seal drips steadily" Check for squareness of stuffing box to shaft Align shaft, impeller, bearing, etc., to prevent shaft vibration and/or distortion of gland plate and/or mating ring
Seal life is short	Abrasive fluid	Prevent abrasives from accumulating at seal faces Add bypass flush line if not in use Use abrasive separator or filter
	Seal running too hot	Increases cooling of seal faces Increase bypass flush line flow Check for obstructed flow in cooling lines
	Equipment mechanically out of line	Align Check for rubbing of seal on shaft

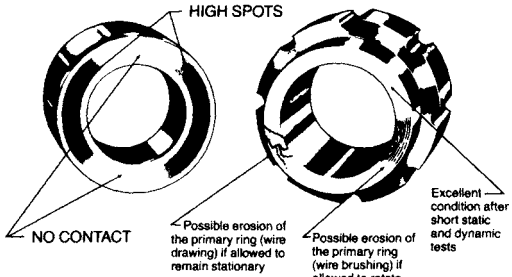
TABLE 32.7 Checklist for Identifying Causes of Seal Leakage (*Continued*)

1. FULL CONTACT PATTERN



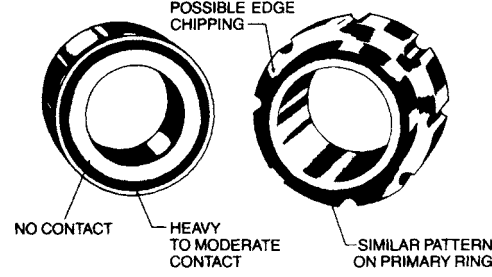
OBSERVATION:
Typical contact pattern for a **non-leaking** seal. Full contact on the mating ring surface through 360°. Little or no measurable wear on either seal ring. If leakage is present with this type face pattern, the secondary seals must be examined.

4. MECHANICAL DISTORTION



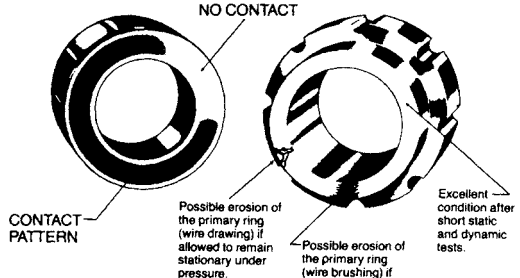
OBSERVATION:
Mating ring is distorted mechanically, creating two large contact spots. Pattern fades away between contact areas.

2. CONING (NEGATIVE ROTATION)



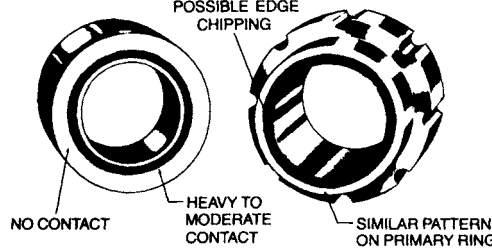
OBSERVATION:
Heavy contact on the mating ring pattern at the outside diameter of the seal. Fades away to no visible contact at the inside diameter of contact pattern. Possible edge chipping on the outside diameter of primary ring.

5. MECHANICAL DISTORTION



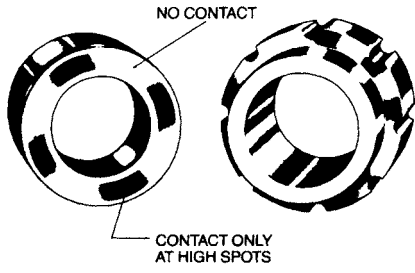
OBSERVATION:
Mating ring is being distorted mechanically, creating contact through approximately 270°. Pattern fades away at the low spot.

3. THERMAL DISTORTION (POSITIVE ROTATION)



OBSERVATION:
Heavy contact on the mating ring pattern at the inside diameter of the seal. Fades away to no visible contact at the outside diameter of contact pattern. Possible edge chipping on the inside diameter of the primary ring.

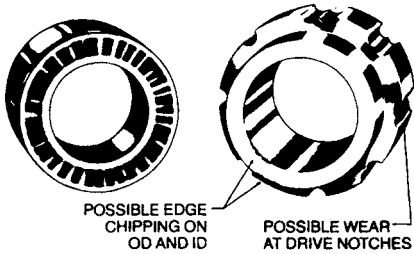
6. MECHANICAL DISTORTION



OBSERVATION:
Mating ring is being distorted mechanically, creating contact at bolts. High spots are at each bolt location.

FIGURE 32.34 Identifying causes of seal leakage. (Source: John Crane-Houdailee, Inc.)

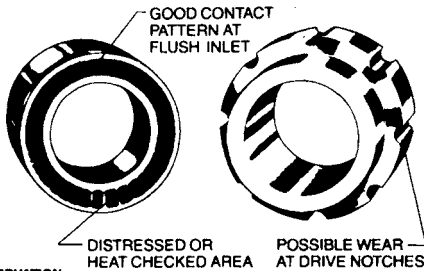
7. HIGH WEAR OR THERMALLY DISTRESSED SURFACE



OBSERVATION:

High wear of mating or thermally distressed surface (heat checking) through 360°. High primary ring wear with carbon deposits on atmosphere side of seal. Possible edge chipping of primary ring due to opening and closing of seal faces.

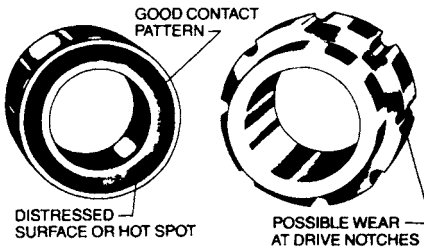
8. SECTION OF THERMALLY DISTRESSED SURFACE



OBSERVATION:

Thermally distressed area approximately 1/3 of the contact pattern. Distressed area 180° from inlet of seal flush. High primary ring wear with possible carbon deposits on atmosphere side of seal.

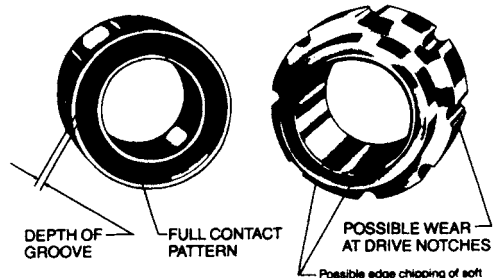
9. PATCHES OF THERMALLY DISTRESSED SURFACE



OBSERVATION:

Patches of thermally distressed surface (heat checking). 2, 3, 4, 5 or 6 hot spots are possible. High primary ring wear with possible carbon deposits on atmosphere side of seal. Failure due to hot spots (thermal asperities) is likely to occur on light specific gravity liquids at high speeds and pressures.

10. HIGH WEAR AND GROOVING

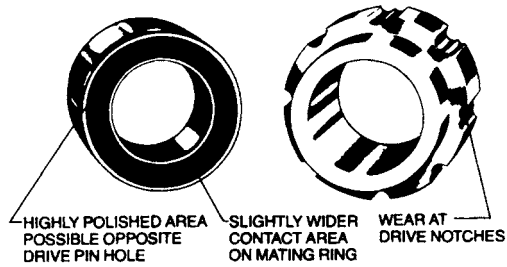


OBSERVATION:

High wear of the mating ring. Primary ring has grooved the mating ring evenly through 360°.

Possible edge chipping of soft carbon primary rings. Edges will be rounded for hard material of construction like tungsten carbide.

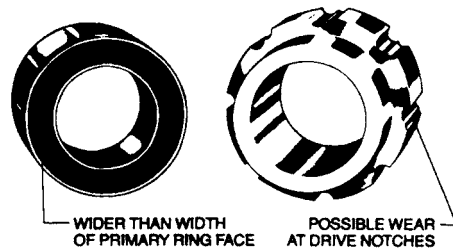
11. OUT-OF-SQUARE MATING RING



OBSERVATION:

Contact pattern through 360° slightly larger than primary ring face width. High spot may be present on the mating ring opposite a drive pin hole. Mating ring without static seals will rock or move in gland plate or holder.

12. WIDE CONTACT PATTERN



OBSERVATION:

Contact pattern considerably wider on the mating ring than the face width of the primary ring.

FIGURE 32.34 (Continued)

This page intentionally left blank

Positive Displacement Pumps

In positive displacement pumps, a fixed volume of liquid is pumped from the inlet zone into the discharge zone. The two main types of positive displacement pumps are *reciprocating* and *rotary*.

Positive displacement pumps produce the head imposed on them by the restriction to flow on the discharge side.

33.1 Reciprocating Pumps

The main characteristics of reciprocating pumps are:

- They produce pulsating flow.
- They develop high shutoff or stalling pressure.
- They display almost constant capacity when driven by a motor.
- They are subject to vapor binding at low net positive suction head (NPSH) conditions.

The three classes of reciprocating pumps are: piston, plunger, and diaphragm pumps.

33.1.1 Piston Pumps

Piston pumps consist of a reciprocating piston inside a cylinder. The piston is connected to a rod that passes through a gland at the end of the cylinder (Figs. 33.1, 33.2, and 33.3).

The reciprocating motion of the piston creates the condition for suction and delivery of the liquid. When the piston moves to the right, a vacuum is developed between the head of the cylinder and the piston. The liquid enters the cylinder due to the pressure difference between the suction tank and the cylinder. During this time, the delivery valve (5) is closed. It is subjected to the pressure of the liquid entering the cylinder. When the piston moves to the left, the pressure inside the cylinder closes the suction valve (4) and opens the delivery valve (5) and the liquid is pushed into the delivery piping. The rate of delivery starts from zero when the piston starts to move to the left. It reaches a maximum at the mid point of the stroke approximately when it is fully accelerated. Then the delivery rate gradually falls to zero. Then, there is a time interval during the return stroke when the delivery remains at

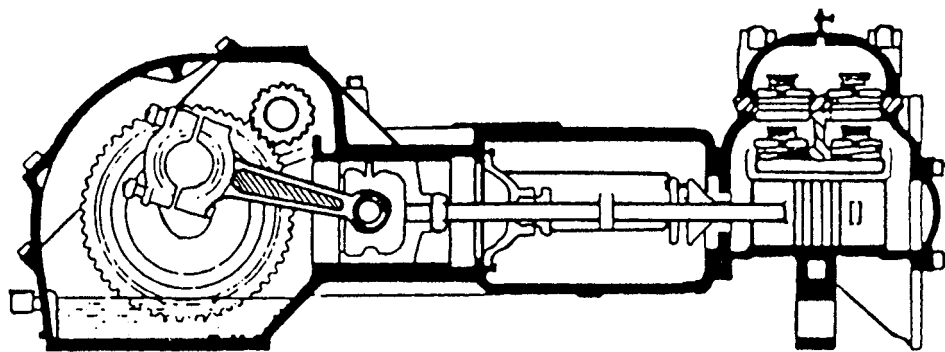


FIGURE 33.1 Basic design of the piston pump.

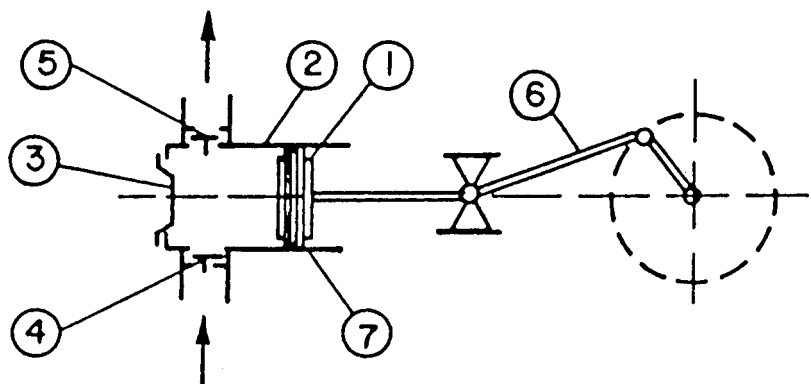


FIGURE 33.2 Operating principle for a horizontal, single-acting piston pump: (1) piston; (2) cylinder; (3) head of cylinder; (4) suction valve; (5) delivery valve; (6) crank and connecting rod assembly; (7) seal rings.

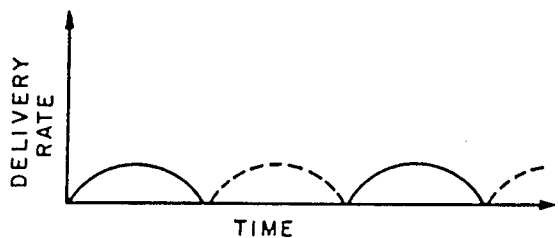


FIGURE 33.3 Liquid delivery rate for a simplex pump.

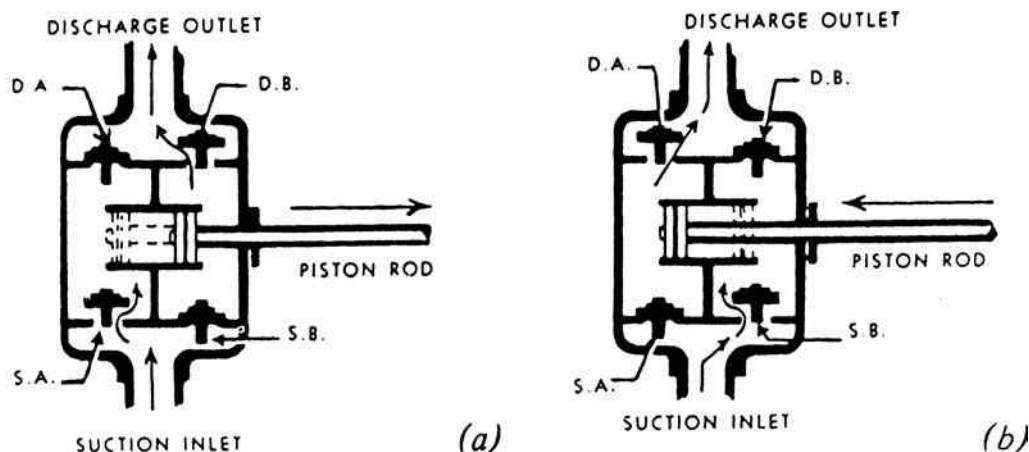


FIGURE 33.4 Double-acting piston pump.

zero (Fig. 33.3). The crankshaft assembly (6) converts the rotary motion of the drive wheel to the linear movement of the piston.

Figure 33.4 illustrates a double-acting piston pump. It has two discharge valves, D.A. and D.B. and two suction valves, S.A. and S.B.

When the piston moves to the right, the liquid is drawn into the cylinder via S.A. and simultaneously the liquid is pushed out via D.B. When the piston moves to the left, the liquid is drawn into the cylinder via S.B. and discharged via D.A. Since both sides of the piston are involved in the pumping action, the pump is called “double-acting.”

33.1.2 Plunger Pumps

In plunger pumps, the reciprocating motion of the plunger through packing glands cause the displacement of the liquid from the cylinder. In these pumps there is a considerable radial clearance between the plunger and the cylinder walls. They are usually single acting because only one end of the plunger is used for pumping liquid (Fig. 33.5).

The surface of the cylinder does not require a high degree of finishing as does a piston pump. The leakage is usually minimized by tightening the packing gland (3).

These pumps are usually suited for handling suspension and viscous fluids due to their large radial clearances and ability to generate high pressures.

FIGURE 33.5 Operating principle behind a single-acting horizontal plunger pump: (1) plunger; (2) cylinder; (3) packing gland; (4) suction valve; (5) delivery valve.

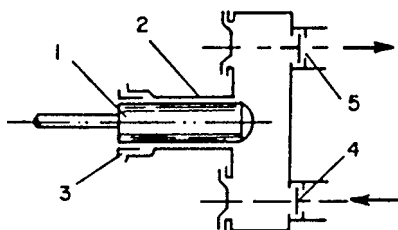
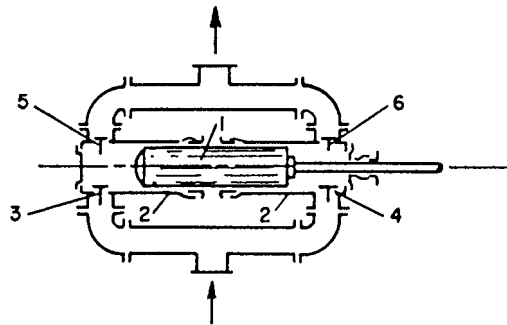


FIGURE 33.6 Operating principle behind a horizontal, double-acting plunger pump: (1) plunger; (2) cylinders; (3–4) suction valves; (5–6) delivery valves.



Piston and plunger pumps can provide a more even delivery when operating as double acting (Fig. 33.6). The double-acting plunger pump is considered to be an aggregate of two single-acting pumps having two suction and two delivery valves.

When the plunger moves to the right, the liquid enters the cylinder (2) through suction valve (3) and simultaneously the liquid is discharged through discharge valve (6). When the plunger moves to the left, the liquid enters the cylinder through suction valve (4) and is simultaneously discharged through delivery valve (5).

Hence, suction and delivery occurs during each stroke in double-acting pumps. Consequently, these pumps have higher capacity, and their delivery is smoother than single-acting pumps.

Figures 33.7 and 33.8 illustrate a three-plunger pump (triplex pump). It is a widely used design due to its ability to provide a pulsation-free flowrate and smooth torque at maximum stroke frequencies which can exceed 1500 strokes per minute (st/min).

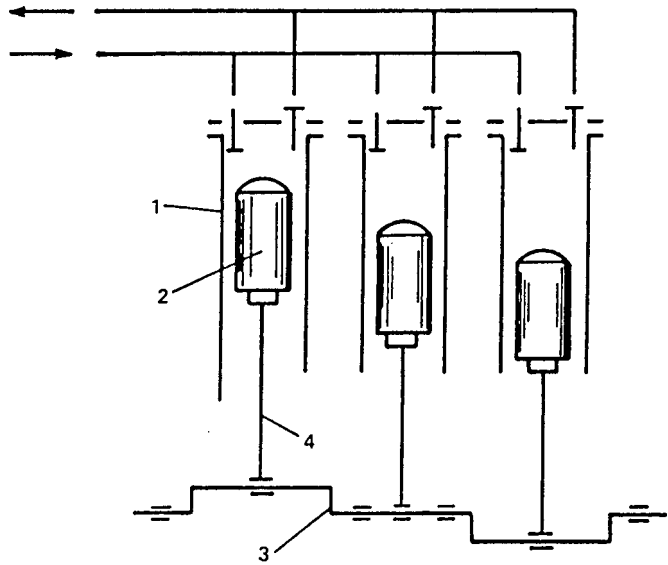


FIGURE 33.7 Operating principle of a triplex pump: (1) cylinders; (2) plungers; (3) crankshaft; (4) connecting rods.

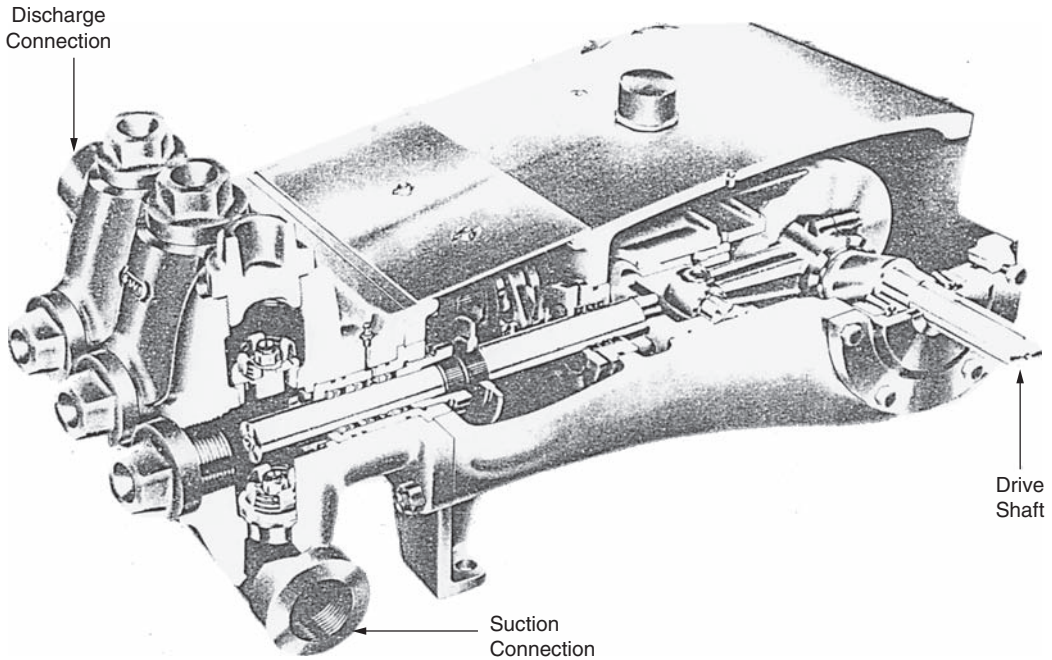


FIGURE 33.8 Power-driven triplex pump. (Source: Reprinted with permission from National Oilwell Varco.)

Triplex pumps are single-acting tripled pumps having three cranks located 120° from each other. The total delivery of a triplex pump is the sum of the deliveries of the three individual single acting pumps. During one rotation of the crankshaft, the liquid is suctioned and delivered once in each cylinder. Single-acting plungers are normally used in higher-pressure units. A triplex (three plungers) is used to reduce flow pulsations substantially compared with single or duplex pumps. In general, the reciprocating pumps-flowrate decreases when the liquid viscosity increases because the speed must be reduced. The efficiency of the pump drops when the viscosity increases. The differential pressure generated by reciprocating pumps is independent of the liquid density. This is in contrast to centrifugal pumps. The differential pressure in reciprocating pumps is entirely dependent on the magnitude of the force exerted on the piston. Reciprocating pumps are normally used for sludge and slurry services particularly where other types of pumps are inoperable or troublesome. The maintenance in these conditions is usually high due to wear of valve, cylinder, rod, and packing. The delivery of a piston pump is equal the total volume swept out by the piston in the cylinder times the number of strokes of the piston per unit time.

Thus, the theoretical delivery in a single-acting pump is:

$$Q_{th} = F \times S \times n$$

where F = piston cross-sectional area

S = stroke length

n = number of crankshaft revolutions (or number of double strokes)

In double-acting pumps, each crankshaft revolution has two suctions, and two deliveries. When the piston moves to the right (Fig. 33.6), the volume of liquid sucked from the left side is FS . The delivery from the right side is $(F - f)S$, where f is the cross-sectional area of the rod.

When the piston moves to the left, the volume delivered from the left side into the piping is FS . While the volume sucked into the side is $(F - f)S$.

Therefore, for n rotations of the crankshaft, the theoretical delivery of a double-acting pump is:

$$Q_{th} = FSn + (F - f)Sn = (2F - f)Sn$$

The actual delivery is less than the theoretical one due to leakage across the piston and valves. The actual pump delivery is:

$$Q = Q_{th} \eta_v$$

where, η_v is the volumetric efficiency (ratio of actual discharge to swept volume). For large pumps, the volumetric efficiency is typically 0.97–0.99; for medium flow rates ($Q = 20\text{--}300$ m³/h), $\eta_v = 0.9\text{--}0.95$; and for small flow rates, $\eta_v = 0.85\text{--}0.9$.

33.1.3 Diaphragm Pumps

The pumping action in a diaphragm pump is achieved by periodic movement of a flexible membrane. The main advantages of a diaphragm pump are:

1. No stuffing box is required
2. Ability to tolerate abrasive slurries and chemically aggressive liquids

Figure 33.9 illustrates a mechanically actuated diaphragm pump. The liquid being pumped is completely isolated from the reciprocating mechanism by the diaphragm. This eliminates leakage along the piston rod and plunger.

The diaphragm D is attached to the piston guide P by the disc B. An eccentric is used to produce the reciprocating motion of the guide P. This causes the diaphragm to move back and forth resulting in the pumping action.

Figure 33.10 illustrates the operation of another diaphragm pump. A noncorrosive liquid is displaced in a cylinder (2) where the plunger (1) is operating. A flexible membrane (3) made from soft rubber or a special steel moves the liquid.

The diaphragm moves to the right when the plunger moves upward and the liquid is sucked in through globe valve (4). Then, the diaphragm moves to the left when the plunger moves downward and the liquid is discharged through delivery valve (5). The liquid being pumped is in contact with the parts of the pump that are made of corrosion and erosion-resistant materials. New and improved designs of fully sealed diaphragm and bellows pumps have been developed to minimize leakage. These types vary from micrometering pumps whose capacity is in milliliters to large diaphragm pumps requiring drives of several hundred kilowatts for high-pressure service. These pumps are economically viable due to their low maintenance and high reliability.

The pumping characteristic of these pumps is usually not linear. Figure 33.11 illustrates the relationship between the flow and the pressure for various stroke lengths. However, the metering rate varies linearly with the stroke frequency.

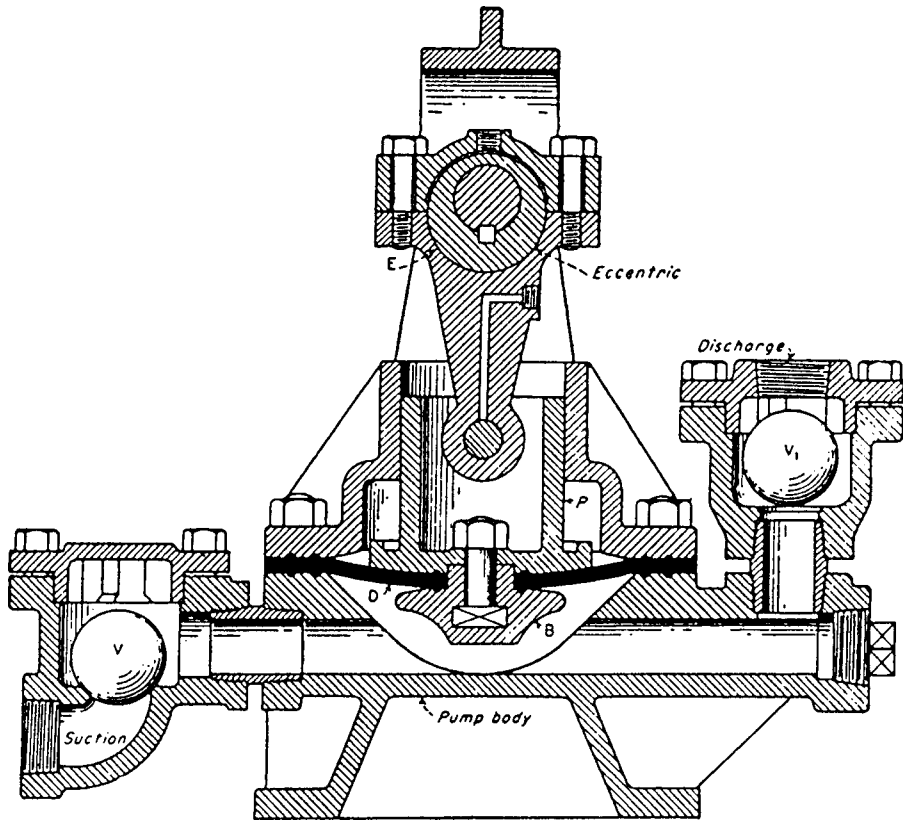
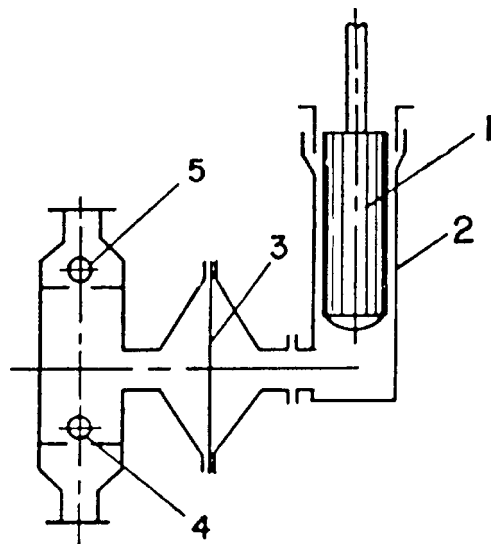


FIGURE 33.9 Mechanically actuated diaphragm pump.

FIGURE 33.10 Operating scheme for a diaphragm pump: (1) plunger; (2) cylinder; (3) diaphragm; (4) suction valve; (5) delivery valve.



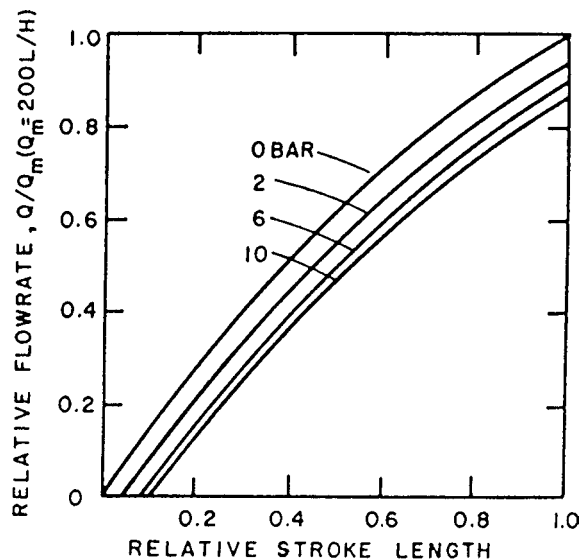


FIGURE 33.11 Diaphragm metering pump flow characteristics.

33.2 Rotary Pumps

In rotary pumps, the rotating parts move relative to the casing. It generates a reduced pressure in the suction lines to draw the liquid and then discharges it at a higher pressure.

The size and rotational speed determine the capacity of a rotary pump. The flow generated by these pumps is normally constant when compared to the fluctuating flows of reciprocating pumps. The main advantage of rotary over centrifugal pumps is their capability of handling highly viscous liquids.

Rotary pumps have a simple design and high efficiency in handling flow conditions that are considered too low for use by centrifugal pumps economically. Their delivery pressure is moderate, and their capacity is small or medium.

The five types of rotary pumps are: gears, screw, lobe, cam, and vane. However, there is considerable overlap among these types.

33.2.1 Gear Pumps

The *external gear pump* is the simplest type of rotary pumps. It is illustrated in Fig. 33.12. There are two gear wheels (2) inside the casing (1). They provide a tight fit to effectively seal the spaces between each pair of adjacent teeth.

The driver drives one of the gear wheels and the second rotates in mesh. When the gear wheels pass the suction opening “A,” the liquid is impounded between the teeth. It is then carried around and discharged at “B.”

The gear wheel teeth produce pulsations in the delivery. The frequency of the pulsations is equal to the product of the number of teeth on both wheels and the speed of rotation. The pulsations can be reduced by having *helical* teeth with a particular angle.

FIGURE 33.12 Operation of a gear-type rotary pump: (1) casing; (2) gear wheels.

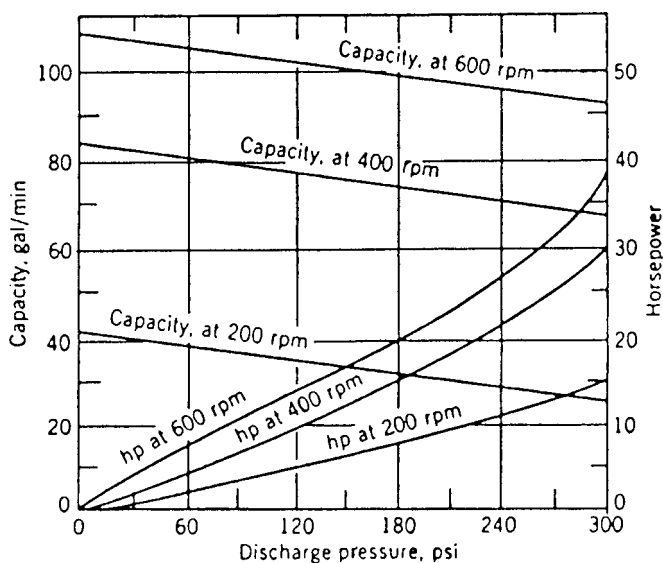
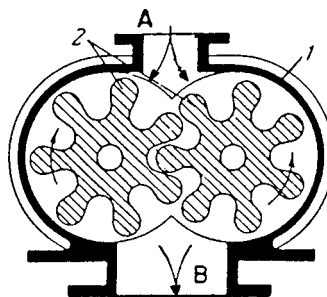


FIGURE 33.13 Typical performance characteristics of an external gear pump.

Gear pumps are most suited for liquids having high viscosity, but they cannot be used with suspensions. The spacing between the teeth is too small to handle solids without suffering significant erosion.

Figure 33.13 illustrates typical performance characteristics of external gear pumps. Note the decrease in capacity with increasing discharge pressure.

33.2.2 Screw Pumps

Screw pumps may have up to three screws turning along the pump axis. The liquid is pushed between the screw threads and the casing.

Screw pumps can be single-rotor or multiple-rotor. The liquid is pumped by the progressing cavities along the rotating screw from inlet to outlet. The flow is smooth because vibration is minimized by the axial flow pattern.

Slurries with relatively large particles can be handled by a single-rotor screw pump (also called a progressive cavity pump). This type of pump consists of a rotor that rotates within a stator, executing a compound movement. The rotor revolves around its axis while the axis rotates. Hence, the rotor has the shape of a helical screw, and the stator has a double internal helical thread. When the rotor rotates a complete turn, the eccentric movement permits the rotor to contact the entire surface of the stator. The space between the rotor and the stator contains the entrapped liquid which is pushed continuously toward the pump outlet. This pumping action produces a continuous, smooth, and relatively low flow. This type of action minimizes fracturing of particles and abrasion damage inside the pump. The food processing and chemical industries use single-screw pumps extensively for handling solid/liquid mixtures that are abrasive or require careful handling of the solids.

Figure 33.14 shows a twin-screw pump. It consists of two sets of screws that rotate and mesh within an accurately bored casing. The operating clearances maintained between the screws are relatively tight. A pair of timing gears mounted on the shafts maintain the tight clearance. The power is transmitted from the drive shaft to the driven shaft. The rotating screws are housed within the pump body which consists of a casing with two precision-machined bores.

Some screw pumps operate without timing gears. The driver in these pumps is used to turn the screws directly. Figure 33.15 illustrates typical designs of screw pumps.

Figure 33.16 illustrates the operation of a three-rotor screw pump. The center rotor drives the remaining two. The liquid enters at the end of the rotor, and is driven smoothly to the delivery end.

A slug of liquid is displaced mechanically from the inlet to the outlet by being trapped in the helical cavity (referred to as a "positive lock") which is created by meshing of the screws.

33.2.3 Two- or Three-Lobe Pumps

The design of two- or three-lobe pumps is similar to gear pumps (Fig. 33.17), but the lobes replace the gear wheels. An external gearing mechanism drives the lobes separately preventing them from touching each other. The small clearance (a few thousandths of an inch) between the lobes and the casing minimizes friction and wear but maintains a small leakage from the delivery to the suction side. The lobe pumps have characteristics similar to those of gear pumps.

33.2.4 Cam Pumps

Cam pumps have an eccentrically mounted circular rotor that sweeps a circle. The radius of the circle is equal to the sum of the radius of the rotor and the eccentricity.

Figure 33.18 illustrates a cam pump utilizing a plunger valve. The circular cam is mounted firmly on the shaft. It is housed inside the rotor ring which is free to rotate around the cam. The plunger acts as a discharge valve by sliding freely through a slide pin. The plunger can be substituted by any form of vane which provides a seal between the suction and discharge lines.

The pump design is such that the friction and wear due to contacts between the surfaces are minimized except for the cam inside the rotor. When the cam rotates, it expels the liquid from the space ahead of it and sucks in the liquid behind it. The characteristics of cam pumps are similar to those of gear pumps.

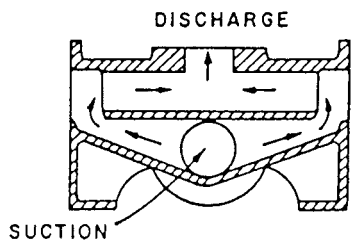
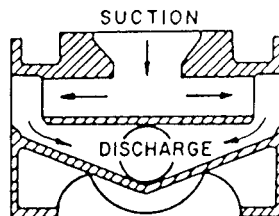
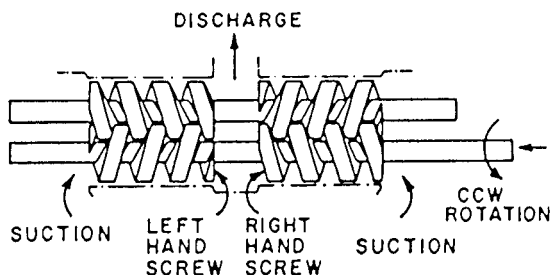
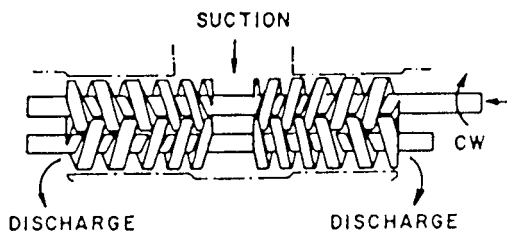
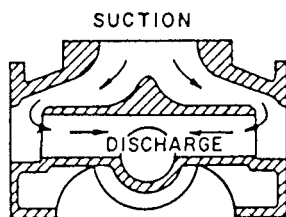
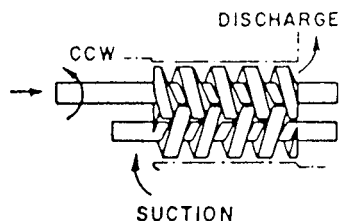
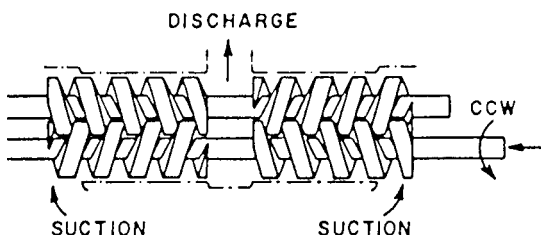
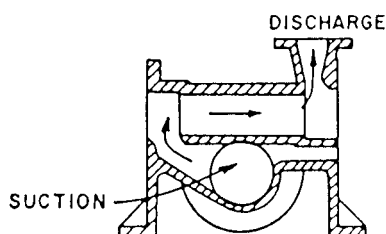
STANDARD END-FLOW DESIGNREVERSE-FLOW DESIGNEND-FLOW DESIGN WITH EXTENDED SCREWSINGLE-ENTRY END-FLOW DESIGN

FIGURE 33.14 Typical twin-screw pump bodies and flow patterns.

33.2.5 Vane Pumps

Figure 33.19 illustrates a vane pump including a massive cylinder (1), which carries rectangular vanes in slots arranged at intervals around the curved surface of the rotor which is located eccentrically within the casing (2).

Centrifugal forces throw the tip of the vanes (3) outward. When the cylinder rotates, the space behind a vane enlarges as it moves from the suction nozzle (5), forming a

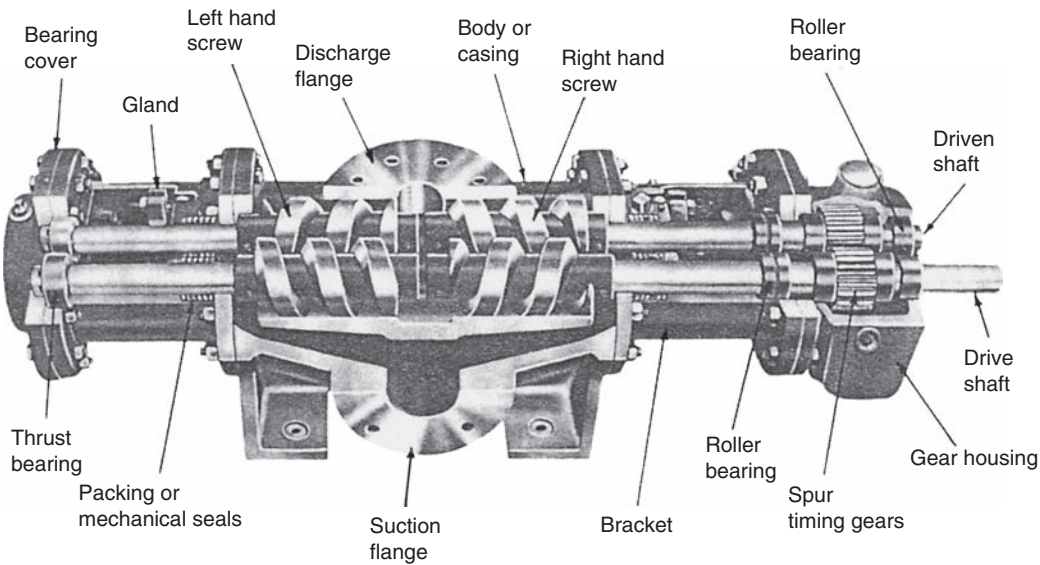


FIGURE 33.15 Cross-section of a twin-screw pump, showing main features. (Courtesy of Worthington Pump Inc., Mountainside, NJ.)

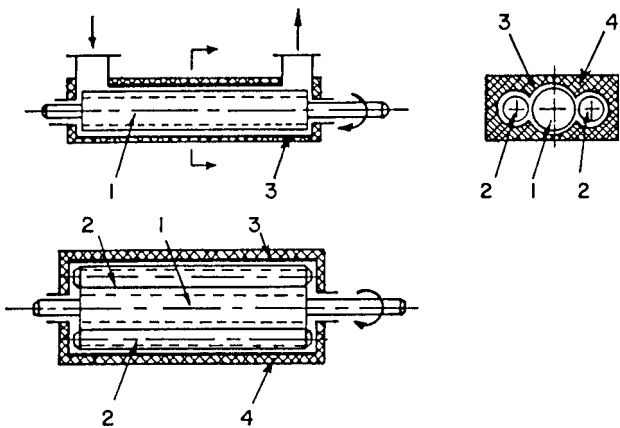


FIGURE 33.16 Operating scheme of a three-rotor screw pump: (1) driving screw; (2) driven screws; (3) sleeve; (4) casing.

vacuum in space (4) resulting in the liquid in the suction nozzle (4) being drawn in. The liquid is trapped between the vanes and the casing. When the vane moves from the vertical axis in the direction of rotation, the volume of the working space (4) decreases and the liquid is pushed out through the discharge nozzle (6).

The vane dimensions change due to wear. A new vane must be inserted before the seal is broken.

FIGURE 33.17 Sectional diagram of a three-lobe pump.

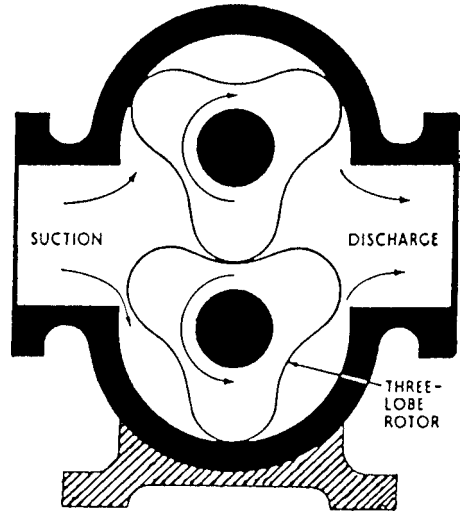
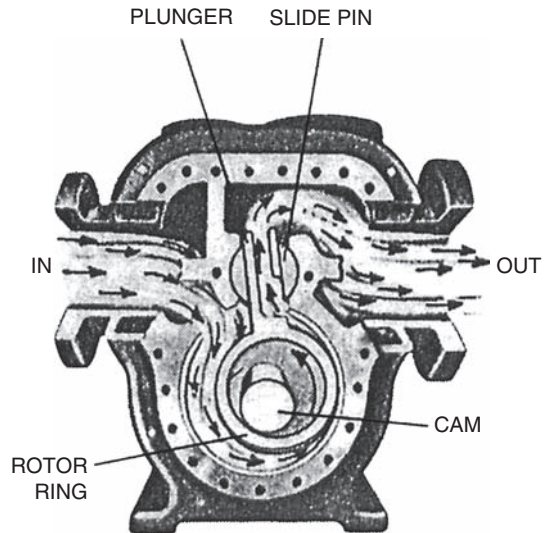


FIGURE 33.18 Cross-sectional view of a cam pump.



Bellows-Type Metering Pumps

Figure 33.20 illustrates a bellows-type metering pump. The components of these pumps are built from glass and polytetrafluoroethylene (PTFE) components. Their delivery pressure is limited to 5 bar (70 psi) because of the glass parts.

The output capacity can be large because fatigue-free bellows can be produced having several hundred millimeters in diameters. The normal life of the bellows can be up to 5000–10,000 hours at maximum load. The metering characteristic is linear due to the radial stiffness of the bellows. However, it depends slightly on the pressure. The failure

FIGURE 33.19 A sliding vane-pump: (1) rotor; (2) casing; (3) vanes; (4) working space; (5) suction nozzle; (6) delivery nozzle.

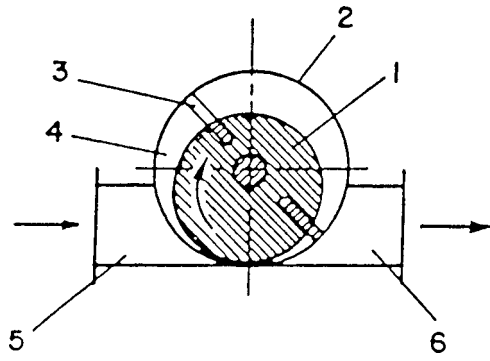
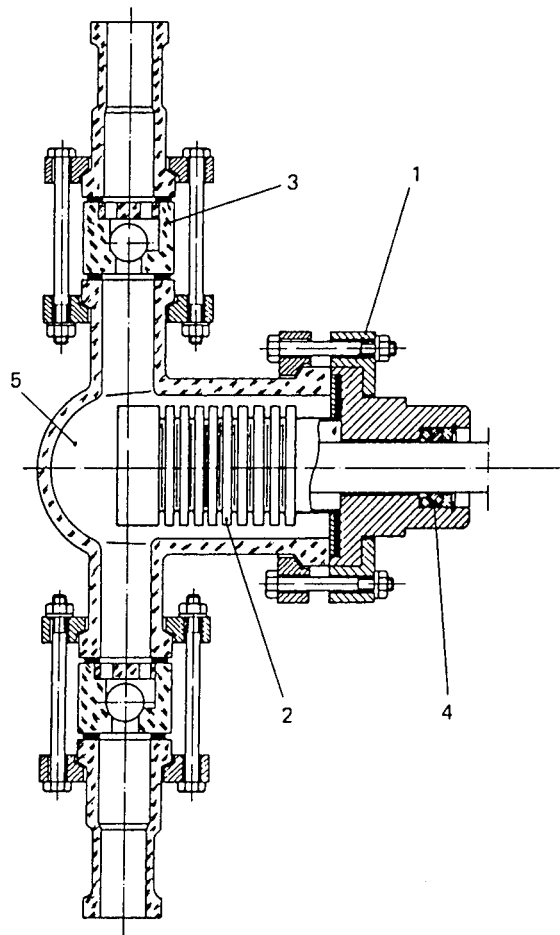


FIGURE 33.20 A bellows-type metering pump: (1) rupture chamber; (2) bellows; (3) check valve; (4) packed seal; (5) pump chamber.



Pump Type	Limits	Application	Wetted Parts Made of
Low-pressure diaphragm pump with direct mechanical drive	<6–10 bar (85–140 psi)	Metering, pumping	PVC or austenitic stainless steel, elastomer diaphragm
Low-pressure bellows pump with direct mechanical drive	<5 bar (70 psi)	Metering, pumping	Glass; PTFE bellows
High-pressure microdiaphragm metering pump with hydraulic drive	<700 bar (9.940 psi)	Metering	Acid-resistant steel
Diaphragm pump with hydraulic compression of tubular member	<50 bar (710 psi)	Metering, pumping	Acid-resistant steel: elastomer diaphragm and tube
Diaphragm pump with hydraulic drive	<350 bar (5000 psi) or PTFE diaphragm <3000 bar (42,600 psi) for metal diaphragm	Metering, pumping	PVC, PTFE, titanium, acid-resistant steel

TABLE 33.1 Operating Limits and Applications of Various Metering Pumps

of the bellows is monitored by level or pressure sensors located in the chamber below the bellows. The chamber is sealed by packing. Table 33.1 illustrates the operating limits and the advantages of various constructions. Leak-free metering or production pumps are required for critical, toxic, or abrasive media with an upper size limit of 300 kW. The stroke length, pressure, temperature, and compatibility with the fluid being processed affects the life of the diaphragm. The design of the diaphragm is empirical. It is based on extensive fatigue trials. The life of the diaphragm can exceed 3000 hours when operating in severe conditions. However, this value can be exceeded considerably at lower pressures, or shorter strokes etc. The failure of the diaphragm can be detected by a float switch that indicates the presence of the process liquid in the space behind the diaphragm. The failure of the diaphragm can also be detected by having two mechanically coupled diaphragms with a pressure switch between them that annunciates a pressure rise when the failure occurs.

The displacement of the diaphragm is greater in hydraulically driven pumps, but the operating conditions are limited to 5000 psi and 120°C by fatigue limit under compression for PTFE. Compact diaphragm metering pumps are suitable for the largest number of applications. The piston is operated by stroke-adjustable drives via a connecting link. The seal is established by a tight liner around the sliding piston. The diaphragm is deflecting between two perforated support plates. The hydraulic system has a relief valve for protection against excess pressure. The relief can be to the outside pump or to the fluid being pumped. The support plate and the relief valve protect the diaphragm against excessive deflection. An automatic vent valve located in the highest point in the hydraulic system provides degassing for the hydraulic medium. Compact diaphragm pumps using a hydraulic oil system usually require the use of sandwich diaphragms. These pumps are suitable for slurries of all types. Metal diaphragms are

used for services beyond the practical limits of PTFE. They are made dead-parallel cold-rolled sheet. These pumps are suited for:

1. Pressures higher than 5000 psi, and temperatures up to 400°C
2. Applications requiring all metal construction usually to ensure radiation resistance

Larger diameters for the diaphragm are needed due to the lower elasticity of the metal. For large capacities, pumps with metal diaphragms are more costly to manufacture than pumps with nonmetallic diaphragms.

Some types of diaphragm pumps have a minimum suction pressure requirements of 7 to 42 psi. These include pumps having metal diaphragms designed for larger output and faster stroking rates. Metal diaphragm pumps are more sensitive to cavitation than pumps having PTFE diaphragms. Cavitation can have significant adverse effects on the metering accuracy and performance. Also, loading shocks, and significant pressures can be generated when cavitation occurs. It should be minimized by having the proper layout for the piping system and pressure conditions. The instantaneous delivery flow is not in-phase with the plunger displacement due to the elasticity of the diaphragm. Hence, the delivered flow starts jerkily at the end of the compression phase and results in a corresponding shock wave.

All high-pressure pumps have this shock problem. It can be overcome by having shock dampers that have been sized based on pulsation theory.

33.3 Bibliography

Cheremisinoff, P. and Cheremisinoff N. P., *Pumps/Compressors/Fans Pocket Handbook*, Technomic Publishing Co., Inc., Lancaster, Pennsylvania, 1989.

Diaphragm Pumps

34.1 Introduction

Diaphragm pumps are displacement pumps with flexible membranes clamped at their peripheries. The central portion moves in a reciprocating motion by mechanical means such as a crank or an eccentric cam, or by fluid means, such as compressed air or liquid under alternating pressure. The inlet and outlet check valves control the flow into and out of the pumping chamber. The distinguishing features of diaphragm pumps are

1. They have no seals or packing.
2. They can be used in applications requiring zero leakage.
3. They are self-priming and can run dry without damage.

34.2 Mechanically Driven Diaphragm Pumps

Figures 34.1 and 34.2 show a mechanically driven single-diaphragm pump with a spring on the plunger rod. The operating pressure cannot exceed the maximum recommended pumping pressure because the spring compresses and will not move the diaphragm. The construction industry uses this type of pumps extensively due to its ability to suck in pebbles and other debris. The compression of the spring prevents a pebble from being pushed through the wall of the pumping chamber or damaging the cranking mechanism.

If the discharge line is relatively long, the pumped liquid can have large inertia. The pump draws liquid from the accumulator and suction line simultaneously. The accumulator is refilled with liquid during the discharge stroke. Thus, the liquid can impose large loads on the diaphragm and cranking mechanism.

The spring can absorb some of the cranking energy early in the discharge stroke and “give it back” during the latter part of the discharge stroke. This reduces the inertia loading on the diaphragm and the cranking mechanism. Diaphragm pumps are used in the automotive fuel system. A cam moves the diaphragm on a suction stroke and the spring returns it on the discharge stroke. Thus, the discharge pressure is determined by the spring. The pressure to the carburetor remains constant regardless of engine (pump) speed or fuel consumption rate.

The diaphragms are usually made of fabric-reinforced elastomers (usually synthetic rubbers) similar to the materials used in pneumatic tires. The forces on the central movable portion of the diaphragm determine the suction and discharge pressures. Mechanically driven diaphragm pumps use reciprocating action to act alternately on

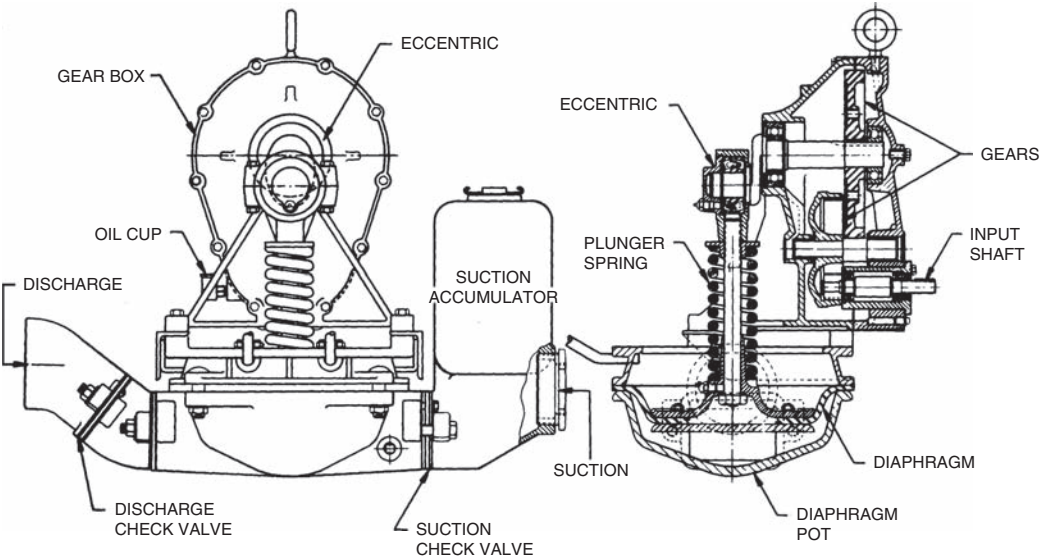
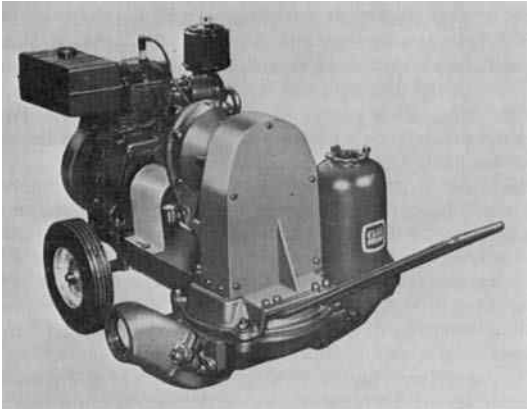


FIGURE 34.1 Cross section of a mechanically driven single-diaphragm pump.

FIGURE 34.2 Mechanically driven single-diaphragm pump, engine-powered.
(Courtesy of Gorman-Rupp.)



two diaphragms. When one diaphragm is pushed on a discharge stroke, the other diaphragm is pulled on a suction stroke. The pumping chambers have common inlet and outlet. The main advantage of duplex diaphragm pumps is that their flow is almost constant.

Mechanically driven diaphragm pumps have the following applications:

1. For dewatering foundations and cofferdams in the construction industry.
2. For pumping lime slurries in sewage treatment plant. Their head is normally limited to 50 ft (15.2 m) and 25 ft (7.6 m) of suction lift.

Hand-operated diaphragm pumps have applications as bilge pumps on sailboats, where hand operation is essential due to loss of power.

34.3 Hydraulically Actuated Diaphragm Pumps

Hydraulic diaphragm pumps use an adjustable-stroke reciprocating piston to force hydraulic fluid into and out of a chamber behind a diaphragm. They are commonly used as metering pumps. Since the diaphragm is balanced (the diaphragm separates the hydraulic fluid from the pumped liquid), the discharge pressure and flow rate can reach 4000 psi (275.2 bar) and 2000 gal/h (7.76 m³/h). The check valves are of the spring-loaded type. They are used with a metering accuracy of $\pm 1\%$ of maximum flow rate. A bypass arrangement must be provided in case the discharge is blocked. These pumps have applications in water and waste treatment plants, pulp and paper, chemical, petrochemical, and petroleum industries. They handle liquids such as hydrazine, phosphate, acids, aluminum chloride, caustic solvents, vinyl acetates, meraptan, and glycol.

34.4 Pneumatically Powered Diaphragm Pumps

The most common type of these pumps are double-diaphragm pumps (duplex pumps). Figure 34.3 illustrates an air-operated double-diaphragm pump. A photograph of the same pump is shown in Fig. 34.4. There are two flexible membranes and two diaphragm

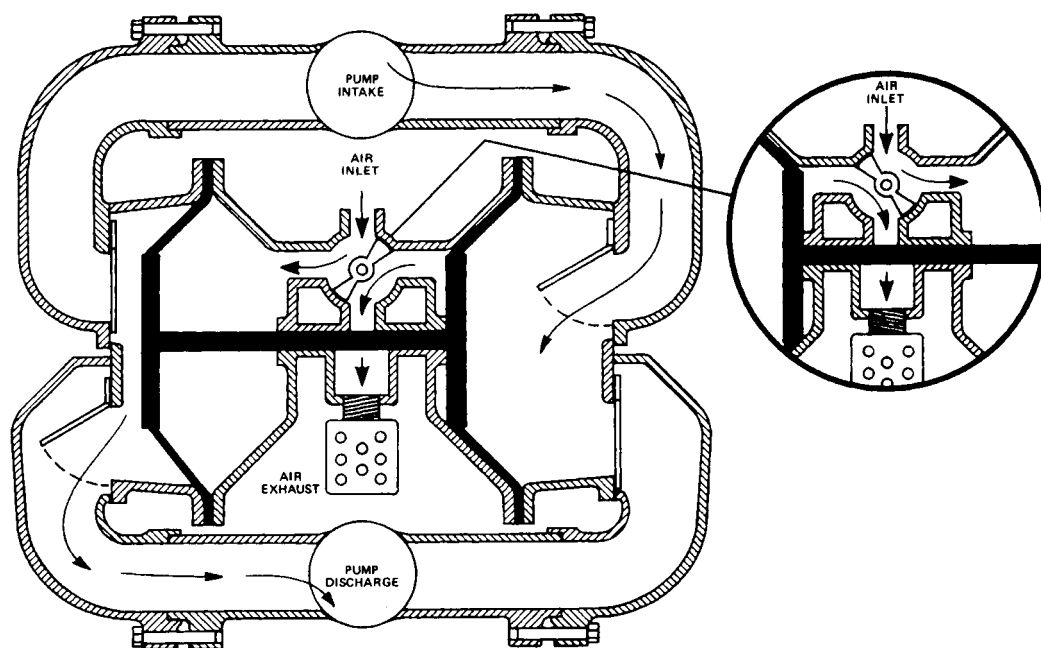
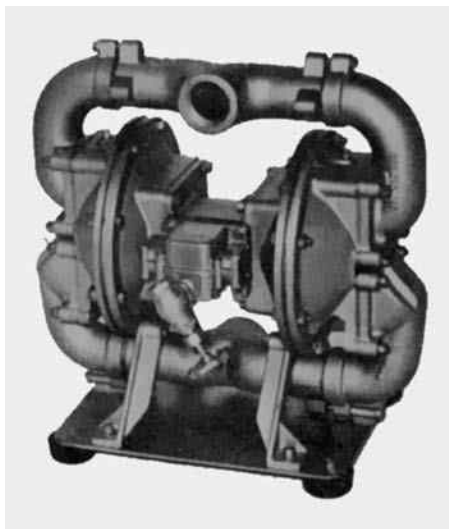


FIGURE 34.3 Cross section of an air-operated double-diaphragm pump with flap check valves. The air distribution valve is schematic for ease of understanding. Circular inset shows reversed air flow to and from chambers behind diaphragms. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

FIGURE 34.4 SandPIPER air-operated double-diaphragm pump with flap check valves. (Source: Reprinted with permission from IDEX Chemical, Food & Pharma.)



chambers. Both diaphragms are moved by compressed air directed to the back side of the left diaphragm while air is exhausted from the back side of the right diaphragm. Upon completion of a stroke, the air distribution valve automatically transfers the compressed air to the back side of the right diaphragm and allows air to exhaust from the left chamber. This pump can handle almost pipe-sized solids.

Since the discharge is from the bottom, this type of pumps is suitable for pumping solids in suspension which may tend to settle out if the pumping rate is reduced or when the pump is shut down. Figure 34.5 illustrates an air-operated double-diaphragm pump with ball-type check valves. A photograph of this pump is shown in Fig. 34.6. The discharge is located at the top to allow air or vapor to be expelled from the chambers easily. The volumetric displacement of these pumps is reduced when air or vapor is trapped in pumps having bottom outlets. The air or vapor is compressed repeatedly instead of the liquid being pumped. This problem is of concern in applications requiring relatively high pumping pressure. If the flow rate is relatively high, there is sufficient turbulence and mixing of air or vapor with the pumped liquid to purge the chambers from the gases.

Figure 34.7 illustrates the performance chart of a typical 2-in air-operated double-diaphragm pump. Note that for a given pressure of compressed air, the discharge pressure decreases with increasing flow.

The performance of these pumps is not affected by the specific gravity of the liquid being pumped (the discharge pressure remains the same for a given capacity and air inlet pressure). However, for centrifugal pumps, the discharge pressure is directly proportional to the specific gravity of the liquid being pumped.

The air consumption rate decreases with capacity. The air consumption is almost proportional to the flow. This is a very important feature of these pumps. It allows the pumps to be used in applications requiring variable flowrate.

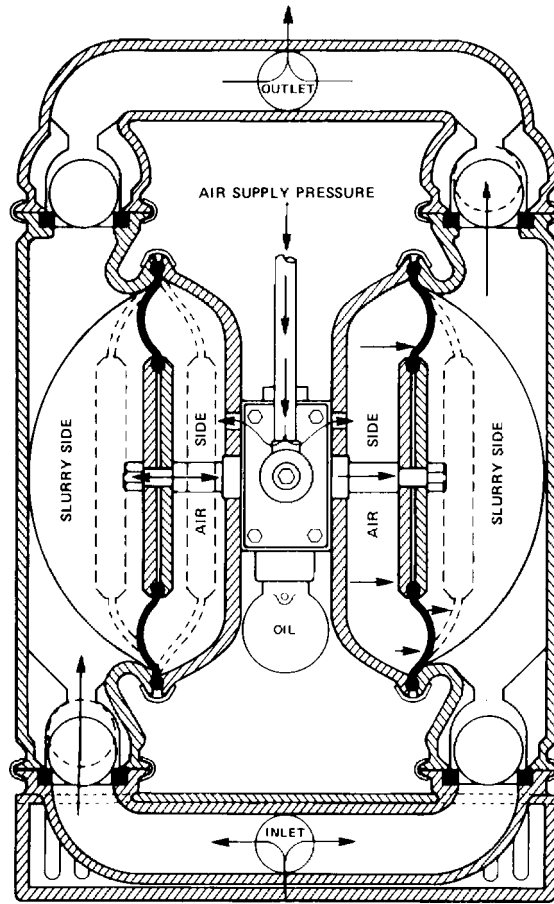


FIGURE 34.5 Cross section of an air-operated double-diaphragm pump with ball check valves.

FIGURE 34.6 CHAMP air-operated double-diaphragm pump with ball check valves. (Courtesy of Wilden Pump & Engineering.)



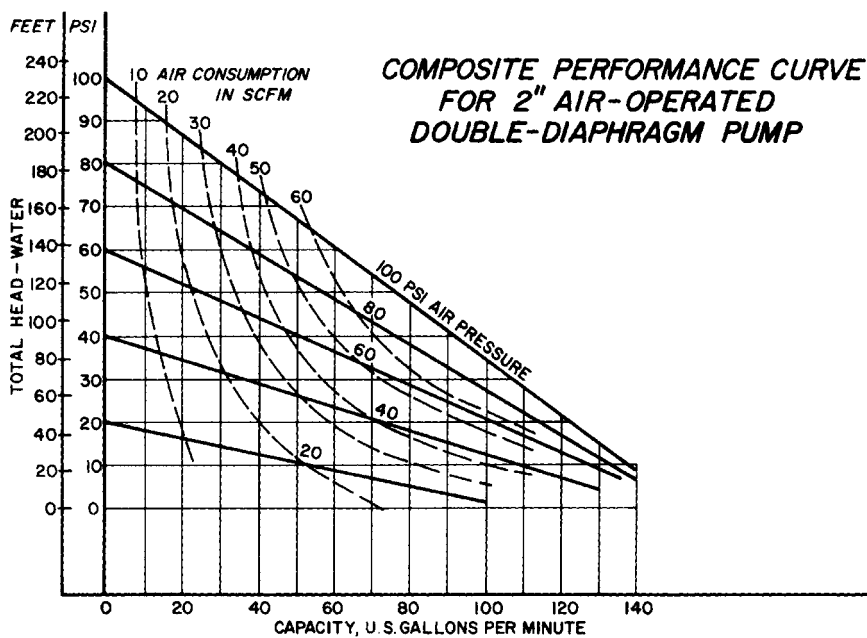


FIGURE 34.7 Performance chart. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

34.5 Materials of Construction

Figure 34.8 illustrates the three types of check valves used for diaphragm pumps (flap, ball, and poppet). The main advantage of flap valves is their ability to handle large objects in suspension. Elastomeric hinges are normally used with flap valves. These valves normally have elastomeric faces or are elastomeric balls because the pumps handle abrasive slurries.

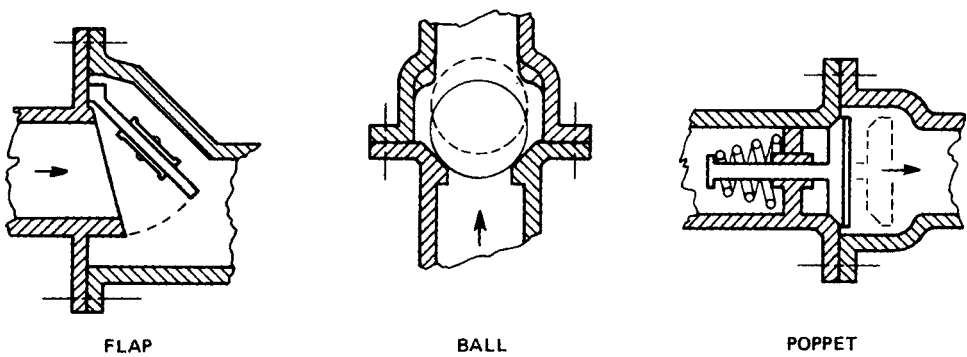


FIGURE 34.8 Three basic types of check valves. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

Diaphragms are normally made of fabric-reinforced synthetic rubber such as neoprene, Buna N, butyl, Viton, and Teflon. Teflon diaphragms are normally used for some solvents and aggressive acids or alkalis.

The diaphragms of air-operated double-diaphragm pumps act as membranes that separate the compressed air from the liquid being pumped. They are balanced. The pump is normally made of cast iron, aluminum, stainless steel, Hastelloy C, Carpenter 20, and plastics reinforced with glass fiber.

34.5.1 Advantages and Limitations

Since the liquid velocity through the check valves of a diaphragm pump does not exceed pipeline velocity, there is minimal scouring and abrasion from a slurry. Thus, diaphragm pumps are ideally suited for handling abrasive slurries. The pumps can be used for liquids having a viscosity up to 50,000 SSU (11,000 cSt). They can also be used for shear sensitive materials, such as latex because there is minimum turbulence and mixing. The maximum pumping pressure of an air-operated pump cannot exceed the pressure of the compressed air powering the pumps.

Pumping Dry Powders

Diaphragm pumps are used to pump dry powders because they can pump air as well as powders. The fluid medium for the powders in suspension is the air. The pump moves the suspended powder in air. In some cases, air is injected into the powder to lower the apparent specific gravity and to get the powder into suspension.

Size Limitations of Air-Operated Diaphragm Pumps

Air-operated diaphragm pumps have a practical size limit. The cost of compressed air and pump components becomes excessive for flows higher 250 gal/min (58.3 m³/h). For larger flows, it is cheaper to use several pumps in parallel.

Pressure Limitations

The pump pressure does not have an upper limit. Compressed air pressure is usually limited to 125 psi (8.6 bar). Pumps can be designed to be operated at higher pressures.

Pump Controls

A pressure regulator in the compressed air supply line is used to control the pumping pressure. An air line valve is used to control the pumping rate.

Liquid Handled

Diaphragm pumps can handle the following liquids and slurries: ceramic slurry, paint, cement grout, chemicals, glue, resin, petroleum products, driller's mud, mill scale, ore concentrates, printer's ink, sewage, filter aids, latex, waste oils, wood preservatives, core washes, asphaltic coatings, bilge waste, radioactive waste, lapping compounds, porcelain frit, mine tailings, volatile solvents, coolant with metal fines, varnish, acids, coatings, soapstone slurries, explosives, lime slurries, yeast, chocolate, and wine.

34.5.2 Limitations of Diaphragm Pumps

In summary, the limitations of diaphragm pumps are

1. Not practical for pumping rates above 250 gal/min (58.3 m³/h)
2. Not manufactured for operating pressures above 125 psi (8.6 bar)

34.5.3 Advantages of Diaphragm Pumps

1. Self-priming from dry start
2. Infinitely variable pumping rate and pressure within capacity of pump
3. No seals or packing
4. Can run dry indefinitely
5. Discharge can be throttled to zero flow indefinitely
6. Explosion-proof for use in hazardous environments
7. Power used in proportion to pumping rate
8. Can be used in confined areas; no heat buildup
9. Can pump abrasive slurries and solids in suspension
10. Can pump viscous liquids up to 50,000 SSU (11,000 cSt)
11. Gentle pumping of shear-sensitive material
12. Can pump dry powders in air suspension
13. No close-fit, sliding, or rotating parts in contact with the liquid being pumped
14. No bypass required as in other displacement pumps
15. Zero leakage
16. Simple to maintain and repair
17. No bedplate or couplings to align
18. Can be used in handling aggressive chemical solutions
19. Can handle a wider range of materials than any other type of pumps

34.6 Bibliography

Karassik, I. J., Krutzsch, W. C., Fraser, W. H., and Messina, J. P., *Pump Handbook*, 2nd ed. McGraw-Hill, New York, 1986.

Canned Motor Pumps

35.1 Canned Motor Pumps Design and Applications

Figure 35.1 illustrates a canned motor pump. These pumps are completely leak-proof and seal-less. They are used for handling corrosive, radioactive, and volatile liquids. The rotors of the motor and pump impeller are assembled on the same shaft. This shaft is placed in a hermetically sealed unit. This design does not include stuffing boxes or mechanical seals. The main advantages of canned motor pumps are the elimination of the requirements for alignment and external lubrication. The pumped liquid is circulated through the rotor of the motor. However, it is isolated from the stator windings by a thin corrosion-resistant nonmagnetic alloy liner. The circulated liquid cools the rotor and lubricates the sleeve bearings. Some designs include a stator jacket. This jacket is filled with dielectric oil. This feature enhances the heat dissipation from the pump. The circulated liquid must be free from solid particles. This is because the rotor and bearing clearances are small. A self-cleaning filter is installed in the discharge of the pump to accomplish this requirement. This filter is shown in Fig. 35.1. An external circulating tube connects the filter to the rear of the motor rotor. Thrust bearings are not included in this design. The axial hydraulic balance of the rotor is achieved using the following features:

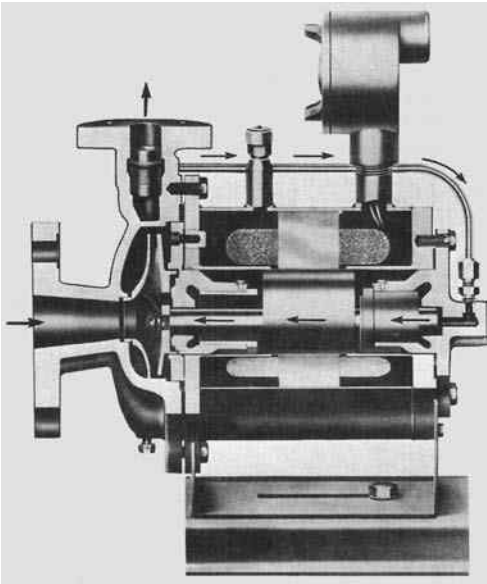
1. Back wearing rings
2. A balancing chamber
3. Balancing holes in the impeller

The length of the back wearing-ring engagement changes when the rotor position varies. This creates a change in the balancing chamber hydraulic pressure. This change in pressure equalizes the axial forces on the rotor.

Figure 35.2 illustrates a different design of a canned motor pump. A small portion of the liquid discharged from the impeller passes around the rotor. The pressure at the back of the rotor is lower than the pressure on the impeller-side of the rotor. This is due to the drop in pressure across the annulus (between the rotor and stator). Thus, the thrust acting on the rotor is toward the right. The thrust acting on the impeller is toward the left (refer to Chap. 31 "Pumps," for detailed explanation on balancing of the axial thrust). The machine is designed to balance the two axial thrusts. The flow passing around the rotor returns to the impeller suction through the following paths:

1. Grooves in the bearings (this flow provides lubrication for the bearings).
2. A servo-thrust control valve and a hollow shaft.

FIGURE 35.1 Canned motor pump.
(Source: Reprinter with permissions from Teikoku USA, Houston, TX.)



The servo-thrust control valve opens when the shaft moves toward the left (Fig. 35.2). This is due to a reduction in flow through the impeller leading to an increase in pressure on the right-hand side of the impeller. This increases the flow through the hollow shaft. The pressure on the right-hand side of the rotor will decrease as a result of the increase in flow through the shaft. This decrease in pressure will continue until a new equilibrium position is reached.

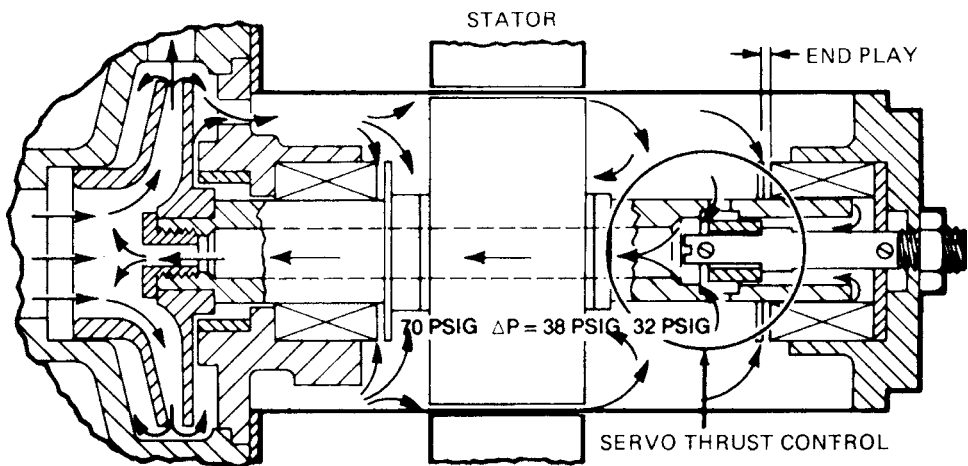


FIGURE 35.2 Canned motor pump with built-in valve for servo-thrust control. (Source: Laurence Pump and Engine.)

Cryogenic liquids in the range of 205°C (400°F) can be handled in these pumps. Liquid temperatures of 538°C (1000°F) can also be handled when the following modifications are made:

1. A thermal heat barrier is added between the pump and the motor.
2. An auxiliary impeller is mounted on the shaft. This impeller circulates the liquid in the motor rotor chamber through a heat exchanger.

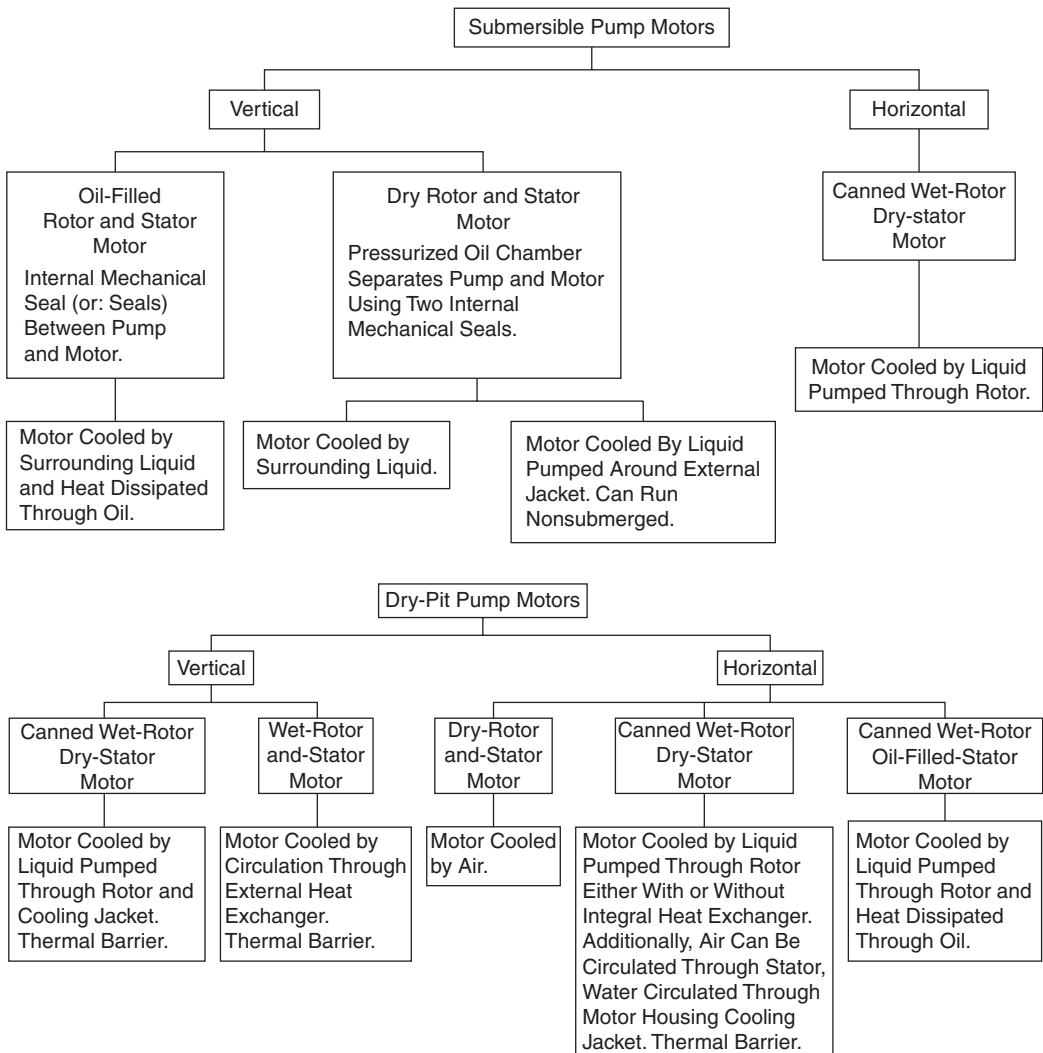


FIGURE 35.3 Integral pump motors for seal-less centrifugal pumps.

The motor is cooled in some designs by forced air circulation through the stator windings and external motor housing jackets. Some pumps can deliver a pressure of 345 bar (5000 psi). However, most of these pumps are designed to handle a pressure of 20 bar (300 psi). Some designs involve multistage construction. These pumps can be used in submersible applications.

35.2 Seal-Less Pump Motors

Seal-less pump motors are completely leak proof. These centrifugal pumps do not require shaft sealing. Some designs are submersible. These pumps are used in the following applications:

1. When leakage cannot be tolerated.
2. When conventional seals cannot be used.

The pump and motor housings are joined in this design. This eliminates the shaft seal. Figure 35.3 illustrates the various motor designs used for seal-less centrifugal pumps.

35.3 Bibliography

Karassik, I. G., *Pump Handbook*, 2nd ed. McGraw-Hill, New York, 1985.

Troubleshooting of Pumps

36.1 Pump Maintenance

The recommendations made in this chapter for the maintenance of pumps are limited to the pumps which are most commonly encountered in industry. This is because of the wide variations in pump type, designs, materials, and applications. A thorough review of the manufacturer's maintenance manual is highly recommended before attempting to service any pump.

36.1.1 Daily Observations of Pump Operation

A daily inspection of the pumping system is recommended. Any irregularities discovered should be reported immediately. This applies to variations in the following parameters:

1. Sound of pump operation
2. Bearing temperatures
3. Stuffing box and seal leakages

An hourly check of all pump instrumentation is recommended. This includes all the pressure gauges, temperature gauges, flowmeters, motor current gauges, etc. Daily checks of the following parameters are also recommended:

1. Pump capacity
2. Pump suction and discharge pressures
3. Pump seal temperatures
4. Power consumption of the pump

Immediate corrective action is recommended if any of these parameters drift outside the normal expected range.

36.1.2 Semiannual Inspection

The following checks are recommended every 6 months:

1. Check the free movement of stuffing box glands.
2. Clean and oil the gland bolts.

3. Inspect the packing and determine if it requires replacements.
4. Check the pump and driver alignment. Correct any misalignment discovered between the pump and driver.
5. Drain all oil-lubricated bearings. Refill these bearings with fresh oil.
6. Check all grease-lubricated bearings.

Determine the following:

1. The bearings contain the correct amount of grease.
2. The grease is still in suitable condition.

36.1.3 Annual Inspection

A thorough inspection of the pump should be performed yearly. The following activities are recommended during this inspection:

1. Remove and clean the bearings.
2. Examine the bearings for flaws or defect.
3. Clean the bearing housings.
4. Examine antifriction bearings for scratches and wear.
5. Place a layer of oil or grease of antifriction bearings.
6. Remove the packing.
7. Examine the shaft sleeves and the shaft for wear.
8. Check the vertical shaft movement of a pump with sleeve bearings at both ends when the packing has been removed. This activity is normally done following the disconnection of the coupling halves for the alignment check. Investigate and determine the cause of any vertical shaft movement exceeding 150% of the original clearance.
9. Check the endplay of the bearings. Determine and correct the cause of the bearing endplay if it exceeds the value recommended by the manufacturer.
10. Check and flush all auxiliary piping, including drains, sealing water piping, and cooling water piping.
11. Flush and clean all auxiliary coolers.
12. Repack the pump stuffing boxes.
13. Realign and reconnect the pump and driver.
14. Recalibrate all instruments and flowmeters.
15. Test the pump to ensure adequate performance.
16. The pump should also be tested following the completion of any repairs.

36.1.4 Complete Overhaul

The frequency of a complete pump overhaul varies significantly. This is due to the significant differences in the following:

1. Pump application
2. Pump construction and materials
3. Liquid handled, etc.

Monthly overhauls are required for pumps used in severe service applications. However, other pumps should be overhauled every 5 years or even less frequently.

The pump should not be overhauled unless there is factual evidence that this work is required. This implies any of the following:

1. The pump performance has deteriorated.
2. The pumping system is noisy.
3. The pump driver is overloaded.

The pump should also be overhauled if there is circumstantial evidence. This indicates that from past experience with this pump or similar pumps on similar service, an overhaul was required.

All the spare parts recommended by the manufacturer should be maintained in stores at all times. This is essential to ensure that the pump is repaired quickly when it fails. The pump can also be repaired at the manufacturer's plant if the repairs required are relatively complex.

36.1.5 Spare and Repair Parts

The minimum number of spare parts required for a pump depends on the severity of the application. The pump manufacturer should be consulted regarding this subject. The spare parts required should be purchased before commissioning the pump. A reorder point (ROP) and reorder quantity (ROQ) should be established for each part. Some replacement parts may have to be supplied larger or smaller than the original part. This depends on the overhaul required for the pump. The serial number and size of the pump should be provided to the manufacturer when ordering spare parts after the pump has been in service. This information is normally stamped on the nameplate of the pump. The pump manufacturer requires this information to provide the correct spare parts.

36.1.6 Record of Inspections and Repairs

All the activities that were performed during the semiannual and annual inspections should be recorded in the pump maintenance file. This file should contain the following:

1. A complete record of all the parts that required maintenance since the pump has been in service
2. Comments about the condition of the parts that were repaired or replaced. This includes the following:
 - a. Rate of wear of the parts
 - b. Repair methods implemented
3. Photographs of all the defective parts that were found in the pump
4. The cost of maintenance and repairs that were performed on the pump

- 5. Operating hours of the pump
- 6. The reasons for each pump failure

A review of this file normally reveals if any modifications are required.

36.1.7 Diagnoses of Pump Troubles

Pumps experience either hydraulic or mechanical problems. Hydraulic problems include the following:

- 1. Failure to deliver the required discharge pressure
- 2. Failure to deliver the required flow
- 3. Failure to maintain its prime after starting

The mechanical problems include the following:

- 1. Excessive power consumption
- 2. Vibration or noise from the pump or bearings
- 3. Failure of a pump component

There is interdependence between the problems encountered in both categories. For example, excessive wear of a pump component is classified as a mechanical problem. However, this problem may lead to a reduction in pump capacity (a hydraulic problem). Thus, a list of potential contributory causes should be provided for each symptom of a problem.

36.2 Troubleshooting of Centrifugal Pumps

Table 36.1 lists all of the symptoms of troubles in centrifugal pumps and the possible cause of each symptom. This table is recommended for troubleshooting centrifugal pumps.

36.3 Troubleshooting of Rotary Pumps

Table 36.2 lists all the symptoms of problems in encountered in rotary pumps and all possible causes of each symptom. This table is recommended for troubleshooting rotary pumps.

36.4 Troubleshooting of Reciprocating Pumps

Table 36.3 lists all symptoms of troubles encountered in reciprocating pumps and all possible causes of each symptom. This table is recommended for troubleshooting reciprocating pumps.

36.5 Troubleshooting of Steam Pumps

Table 36.4 lists all the symptoms of problems encountered in steam pumps and all possible causes of each symptom. This table is recommended for troubleshooting steam pumps.

Symptom	Possible Cause of Trouble (Each number is defined in the list below)
Pump does not deliver water Insufficient capacity delivered Insufficient pressure developed Pump loses prime after starting Pump requires excessive power Stuffing box leaks excessively Packing has short life Pump vibrates or is noisy Bearings have short life Pump overheats and seizes	1, 2, 3, 4, 6, 11, 14, 16, 17, 22, 23 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 14, 17, 20, 22, 23, 29, 30, 31 5, 14, 16, 17, 20, 22, 29, 30, 31 2, 3, 5, 6, 7, 8, 11, 12, 13 15, 16, 17, 18, 19, 20, 23, 24, 26, 27, 29, 33, 34, 37 13, 24, 26, 32, 33, 34, 35, 36, 38, 39, 40 12, 13, 24, 26, 28, 32, 33, 34, 35, 36, 37, 38, 39, 40 2, 3, 4, 9, 10, 11, 21, 23, 24, 25, 26, 27, 28, 30, 35, 36, 41, 42, 43, 44, 45, 46, 47 24, 26, 27, 28, 35, 36, 41, 42, 43, 44, 45, 46, 47 1, 4, 21, 22, 24, 27, 28, 35, 36, 41
Suction troubles 1. Pump not primed 2. Pump or suction pipe not completely filled with liquid 3. Suction lift too high 4. Insufficient margin between suction pressure and vapor pressure 5. Excessive amount of air or gas in liquid 6. Air pocket in suction line 7. Air leaks into suction line 8. Air leaks into pump through stuffing boxes 9. Foot valve too small 10. Foot valve partially clogged 11. Inlet of suction pipe insufficiently submerged 12. Water seal pipe plugged 13. Seal cage improperly located in stuffing box, preventing sealing fluid from entering space to form seal System troubles 14. Speed too low 15. Speed too high 16. Direction of rotation wrong 17. Total head of system higher than design head of pump 18. Total head of system lower than pump design head 19. Specific gravity of liquid different from design 20. Viscosity of liquid different from design 21. Operation at very low capacity 22. Parallel operation of pumps unsuitable for such operation Mechanical troubles 23. Foreign matter in impeller 24. Misalignment 25. Foundations not rigid 26. Shaft bent	27. Rotating part rubbing on stationary part 28. Bearings worn 29. Wearing rings worn 30. Impeller damaged 31. Casing gasket defective, permitting internal leakage 32. Shaft or shaft sleeves worn or scored at packing 33. Packing improperly installed 34. Type of packing incorrect for operating conditions 35. Shaft running off center because of worn bearings or misalignment 36. Rotor out of balance, causing vibration 37. Gland too tight, resulting in to flow of liquid to lubricate packing 38. Cooling liquid not being provided to water-cooled stuffing boxes 39. Excessive clearance at bottom of stuffing box between shaft and casing, causing packing to be forced into pump interior 40. Dirt or grit in sealing liquid, leading to scoring of shaft or shaft sleeve 41. Excessive thrust caused by mechanical failure inside pump or by failure of hydraulic balancing device, if any 42. Excessive grease or oil in antifriction bearing housing or lack of cooling, causing excessive bearing temperature 43. Lack of lubrication 44. Improper installation of antifriction bearings (damage during assembly, incorrect assembly of stacked bearings, use of unmatched bearings as a pair, etc.) 45. Dirt in bearings 46. Rusting of bearings from water in housing 47. Excessive cooling of water-cooled bearing, resulting in condensation of atmospheric moisture in bearing housing

TABLE 36.1 Check Chart for Centrifugal Pump Troubles

Symptom	Possible Cause of Trouble (Each number is defined in the list below)
Pump fails to discharge	1, 2, 3, 4, 5, 6, 8, 9, 16
Pump is noisy	6, 10, 11, 17, 18, 19
Pump wears rapidly	11, 12, 13, 20, 24
Pump not up to capacity	3, 5, 6, 7, 9, 16, 21, 22
Pump starts, then loses suction	1, 2, 6, 7, 10
Pump takes excessive power	14, 15, 17, 20, 23
Suction troubles 1. Pump not properly primed 2. Suction pipe not submerged 3. Strainer clogged 4. Foot valve leaking 5. Suction lift too high 6. Air leaking into suction 7. Suction pipe too small System problems 8. Wrong direction of rotation 9. Low speed 10. Insufficient liquid supply 11. Excessive pressure 12. Grit or dirt in liquid	13. Pump runs dry 14. Viscosity higher than specified 15. Obstruction in discharge line Mechanical troubles 16. Pump worn 17. Drive shaft bent 18. Coupling out of balance or alignment 19. Relief valve chatter 20. Pipe strain on pump casing 21. Air leak at packing 22. Relief valve improperly seated 23. Packing too tight 24. Corrosion

TABLE 36.2 Check Chart for Rotary Pump Troubles

36.6 Vibration Diagnostics

36.6.1 Analysis Symptoms

Pump vibration can be generated by a mechanical or hydraulic problem. Table 36.5 lists the following:

1. All causes of vibration encountered in pumps
2. The predominant frequencies of vibration
3. The direction and location of the predominant vibration amplitude
4. The amplitude response to speed variation during vibration-test runs

The following are the comments on Table 36.5 (the numbers correspond to the “Cause of Vibration” in Table 36.5):

1. Long, high-speed rotors often require field balancing at full speed to make adjustments for rotor deflection and final support conditions. Corrections can be made at balancing rings or at coupling bolts.

Symptom	Possible Cause of Trouble (Each number is defined in the list below)
Liquid end noise	1, 2, 7, 8, 9, 10, 14, 15, 16
Power end noise	17, 18, 19, 20
Overheated power end	10, 19, 21, 22, 23, 24
Water in crankcase	25
Oil leak from crankcase	26, 27
Rapid packing or plunger wear	11, 12, 28, 29
Pitted valves or seats	3, 11, 30
Valves hanging up	31, 32
Leak at cylinder valve hole plugs	10, 13, 33, 34
Loss of prime	1, 4, 5, 6
Suction troubles	16. Obstruction under valve
1. Insufficient suction pressure	17. Main bearings loose
2. Partial loss of prime	18. Bearings worn
3. Cavitation	19. Oil level low
4. Lift too high	20. Plunger loose
5. Leaking suction at foot valve	21. Main bearings tight
6. Acceleration head requirement too high	22. Ventilation inadequate
System problems	23. Belts too tight
7. System shocks	24. Driver misaligned
8. Poorly supported piping, abrupt turns in piping, pipe too small, piping misaligned	25. Condensation
9. Air in liquid	26. Seals worn
10. Overpressure or overspeed	27. Oil level too high
11. Dirty liquid	28. Pump not level and rigid
12. Dirty environment	29. Packing loose
13. Water hammer	30. Corrosion
Mechanical troubles	31. Valve binding
14. Valves broken or badly worn	32. Valve spring broken
15. Packing worn	33. Cylinder plug loose
	34. O-ring seal damaged

TABLE 36.3 Check Chart for Reciprocating Pump Troubles

- Bent rotors can sometimes be straightened by the "hot-spot" procedure, but this should be regarded as a temporary solution because bow will come back in time. Several rotor failures have resulted from this practice. If blades or disks have failed, check for corrosion fatigue, stress corrosion, resonance, off-design operation.
- Straighten bow slowly, running on turning gear or at low speed. If rubbing occurs, trip unit immediately and keep the rotor turning 90° using a shaft wrench every 5 minutes until the rub clears; then resume slow run. This may take 12 to 24 hours.

Symptom	Possible Cause of Trouble (Each number is defined in the list below)
Pump does not develop rated pressure	4, 5, 7, 8
Pump loses capacity after starting	1, 2, 6
Pump vibrates	9, 10, 11, 14
Pump has short strokes	12, 13, 14
Pump operation is erratic	1, 2, 3, 6
Suction troubles 1. Suction line leaks 2. Suction lift too high 3. Cavitation System problems 4. Low steam pressure 5. High exhaust pressure 6. Entrained air or vapors in liquid	Mechanical troubles 7. Worn piston rings in steam end 8. Binding piston rings in liquid end 9. Misalignment 10. Foundation not rigid 11. Piping not supported 12. Excessive steam cushioning 13. Steam valves out of adjustment 14. Liquid piston packing too tight

TABLE 36.4 Check Chart for Steam-Pump Troubles

- Often requires complete rework or new case, but sometimes a mild distortion corrects itself with time (requires periodic internal and external realignment). Usually caused by excessive piping forces or thermal shock.
- Usually caused by poor mat under the foundation or thermal stress (hot spots) or unequal shrinkage. May require extensive and costly repairs.
- Slight rub may clear, but trip the unit immediately if a high-speed rub gets worse. Turn by hand until clear.
- Unless thrust bearing has failed, this is caused by rapid changes of load and temperature. Machine should be opened and inspected.
- Usually caused by excessive pipe strain and/or inadequate mounting and foundation, but is sometimes caused by local heat from pipes or the sun heating the base and foundation.
- Most trouble is caused by poor pipe supports (should use spring hangers), improperly used expansion joints, and poor pipeline up at casing connections. Foundation settling can also cause severe strain.
- Bearings may become distorted from heat. Make a hot check, if possible. Observing contact.
- Watch for brown discoloration, which often precedes recurring failures. This indicates very high local oil film temperatures. Check rotor for vibration. Check bearing design and hot clearances. Check condition of oil, especially viscosity.
- Check clearances and roundness of journal, as well as contact and tight bearing fit in the case. Watch out for vibration transmission from other sources and

Cause of Vibration	Predominant Frequencies											Vert.	Hor.
	0-40%	40-50%	50-100%	1X Running Frequency	2X Running Frequency	Higher Multiples	$\frac{1}{2}$ Running Frequency	$\frac{1}{4}$ Running Frequency	Lower Multiples	Odd Frequency	Very High Frequency		
1. Initial unbalance.....	..	..	..	90	5	5	..	..	..	..	..	40	50
2. Permanent bow or lost rotor parts (vanes).....	..	..	..	90	5	5	..	..	..	..	..		
3. Temporary rotor bow.....	..	..	..	90	5	5	..	..	..	..	..		
4. Casing distortion { Temporary..... Permanent		10		80	5	5	..	..	..	..	..		
		10		80	5	5	..	..	..	..	..		
5. Foundation distortion.....	..	20	..	50	20	..	..	..	..	10	..		
6. Seal rub.....	10	10	10	20	10	10	..	..	10	10	10	30	40
7. Rotor rub, axial.....		20		30	10	10	..	..	10	10	10	30	40
8. Misalignment.....	..	..	..	40	50	10	..	..	..	..	..	20	30
9. Piping forces.....	..	..	..	40	50	10	..	..	..	..	..	20	30
10. Journal and bearing eccentricity.....	..	..	..	80	20	..	..	..	..	..	..	40	50
11. Bearing damage.....		20		40	20	..	..	..	..	..	20	30	40
12. Bearing and support excited vibration (oil whirls, etc.).....	10	70	..	..	..	..	10	10	..	..	..	40	50
13. Unequal bearing stiffness horizontal-vertical.....	..	..	..	..	80	20	..	..	..	..	..	40	50

TABLE 36.5 Vibration Analysis Symptoms

Cause of Vibration	Predominant Frequencies												Hor.	Vert.
	0-40%	40-50%	50-100%	1x Running Frequency	2x Running Frequency	Higher Multiples	1/2 Running Frequency	1/4 Running Frequency	Lower Multiples	Odd Frequency	Very High Frequency			
14. Thrust bearing damage.....	90						..	..	..	..	10	20	30	
Insufficient tightness in assembly of:	Predominant frequency will show at lowest critical or resonant frequency													
15. Rotor (shrink fits).....	40	40	10	..	..	..	..	..	..	10	..	40	50	
16. Bearing liner.....	90		..	..	..	..	..	..	..	10	..	40	50	
17. Bearing cases.....	90		..	..	..	..	..	..	..	10	..	40	50	
18. Casing and support.....	50		..	..	..	..	..	..	..	50	..	40	50	
19. Gear inaccuracy.....	..	..	..	..	..	20	..	..	..	20	60	30	50	
20. Coupling inaccuracy or damage.....	10	20	10	20	30	10	..	..	..	..	..	30	40	

TABLE 36.5 Vibration Analysis Symptoms (*Continued*)

Direction and Location of Predominant Amplitude							Amplitude Response to Speed Variation During Vibration-Test Runs											
							Coming Up					Slowing Down						
Axial	Shaft	Bearings	Casing	Foundation	Piping	Coupling	Stays Same	Increases	Decreases	Peaks	Comes Suddenly	Drops Out Suddenly	Stays Same	Increases	Decreases	Comes Suddenly	Drops Out Suddenly	Cause
10	90	10	..	..	..	..	100	..	Peaks at critical	..	..	..	..	..	100	..	..	1
			..	..	..	..	100	..		..	..	..	..	..	..	2		
			..	..	..	..	60	5		5	30	5	50	5	10	3		
			..	..	..	..	50	5		5	10	30	5	50	5	10	4	
			..	..	..	..	60	..		..	40	..	60	..	..	..	5	
30	80	10	10	10	10	..	80	..	..	..	..	..	20	..	80	..	..	6
			10	10	10	..	70	..	10	10	10	..	70	10	10	7		
			20	20	20	..	40	10	10	10	10	..	50	20	20	8		
			10	10	10	..	30	10	10	20	20	..	40	20	20	9		
			10	10	10	..	40	..	20	20	20	..	40	20	20	10		
10	90	10	..	..	..	..	50	10	Peaks at critical	..	..	..	40	10	50	..	..	11
			..	..	..	..	50	10		10	10	10	10	50	10	20	11	
			..	..	..	..	50	10		10	10	10	10	50	10	20	11	
			..	..	..	..	50	10		10	10	10	10	50	10	20	11	
			..	..	..	..	50	10		10	10	10	10	50	10	20	11	

TABLE 36.5 Vibration Analysis Symptoms (Continued)

Amplitude Response to Speed Variation During Vibration-Test Runs														
Direction and Location of Predominant Amplitude		Coming Up						Slowing Down						
		Shaft	Bearings	Casing	Foundation	Piping	Coupling	Stays Same	Increases	Decreases	Comes Suddenly	Drops Out Suddenly	Cause	
10	50	20	20	20	20	..	..	90	..	..	10	..	90	12
10	40	30	30	..	..	..	50	10	..	..	40	..	10	13
50	60	20	20	..	..	..	10	10	10	20	10	10	10	14
10	60	20	20	..	..	..	..	90	10	..	..	10	90	15
10	80	10	10	..	..	..	..	90	10	..	..	10	90	16
10	70	20	10	..	..	..	..	90	10	..	..	10	90	17
10	50	20	30	..	..	..	..	90	10	..	..	10	90	18
20	80	10	10	..	..	..	20	20	10	20	20	10	10	19
30	70	20	..	..	..	10	20	Loose sleeve, friction or dirt 40 in teeth	10	..	20	10	40	20

TABLE 36.5 Vibration Analysis Symptoms (Continued)

Cause of Vibration	Predominant Frequencies												Vert.	Hor.	
	0-40%	40-50%	50-100%	1x Running Frequency	2x Running Frequency	Higher Multiples	½ Running Frequency	¼ Running Frequency	Lower Multiples	Odd Frequency	Very High Frequency				
21. Rotor and bearing system critical	..	..	..	100	..	..	..	..	..	..	..	40	50		
22. Coupling critical.....	..	..	..	100	Also make sure tooth fit is tight								20	40	
23. Overhang critical.....	..	..	..	100	..	..	..	..	..	..	..	40	50		
Structural resonance of: { 24. Casing 25. Supports 26. Foundation	..	10	..	70	10	..	10	..	..	..	..	40	50		
	..	10	..	70	10	..	10	..	..	..	..	40	50		
	..	20	..	60	10	..	10	..	..	..	..	30	40		
27. Pressure pulsations.....	Most troublesome if combined with resonance												30	40	
28. Electrically excited vibration.....	..	..	..	..	..	↑						30	40		
29. Vibration transmission.....	..	..	..	..	..							90	..	30	40
30. Valve vibration.....	..	..	..	..	..							..	100	30	40

TABLE 36.5 Vibration Analysis Symptoms (Continued)

Cause of Vibration	Predominant Frequencies											Vert.	Hor.	
	0-40%	40-50%	50-100%	1x Running Frequency	2x Running Frequency	Higher Multiples	½ Running Frequency	¼ Running Frequency	Lower Multiples	Odd Frequency	Very High Frequency			
Problem	The section below is meant to identify basic mechanisms													
31. Subharmonic resonance.....	..	Rare—Look for aerodynamic origin (seals)		100						..	..	30	30	
32. Harmonic resonance.....	..	..	..	..	100		..	..	..	..	..	40	40	
33. Friction-induced whirl.....	80	10	10	..	..	..	..	..	..	..	..	40	50	
34. Critical speed.....	..	..	..	100	..	..	..	..	..	..	..	40	50	
35. Resonant vibration.....	..	..	..	100	..	..	..	..	..	..	..	40	40	
36. Oil whirl.....	..	100	Watch for aerodynamic rotor-lift (partial admission, etc.)										40	50
37. Resonant whirl.....	..	100	..	..	..	..	..	..	..	..	..	40	50	
38. Dry whirl.....	..	..	..	..	..	..	..	..	..	..	100	30	40	
39. Clearance-induced vibrations.....	10	80	10	..	..	..	..	..	..	..	..	40	50	
40. Torsional resonance.....	..	..	..	40	20	20	..	..	..	20	..	Torsional		
41. Transient torsional.....	..	..	..	50	..	..	..	..	..	50	..	..	↓	

TABLE 36.5 Vibration Analysis Symptoms (Continued)

Amplitude Response to Speed Variation During Vibration-Test Runs																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																										
Direction and Location of Predominant Amplitude						Coming Up						Slowing Down																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																														
Axial	Shaft	Bearings	Casing	Foundation	Piping	Coupling	Stays Same	Increases	Decreases	Peaks	Comes Suddenly	Drops Out Suddenly	Stays Same	Increases	Decreases	Comes Suddenly	Drops Out Suddenly	Cause																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																								
10	70	30	..	..	..	..	..	20	..	80	..	..	..	..	20	..	..	21																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																								
40	10	10	..	..	..	80	..	20	..	80	..	If loose	..	..	20	50 If loose	..	22																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																								
10	70	10	..	..	..	20	..	30	..	70	..	..	..	..	30	..	..	23																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																								
10	..	40	40	10	10	..	..	20	..	80	..	..	..	..	20	..	..	24																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																								
10	..	20	50	20	10	..	..	20	..	80	..	..	..	..	20	..	..	25																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																								
30	..	10	40	40	10	..	..	20	..	80	..	..	..	..	20	..	..	26																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																								
30	Can excite whirls or resonance	↓	40	30	40	..	90		10%—Depending on origin of disturbance	90	10	→	90	10	→			27																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																								
30																		↓	40	40	20	..	90	..	↓	..	..	90	..	..	28																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																											
30																		↓	40	40	20	..	90	..	↓	..	..	90	..	..	29																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																											
30	..	..	80	10	10	..	80	..	..	..	10	10	80	..	..	10	10	30																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																								
40	20	80	20	20	20	..	..	20	..	20	30	30	..	..	20	30	30	31																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																								
	If bearing is excited	20	20	20	20	..				60	..	..	20	..	20		..	32																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																								
																		20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20

TABLE 36.5 Vibration Analysis Symptoms (Continued)

Amplitude Response to Speed Variation During Vibration-Test Runs											
Direction and Location of Predominant Amplitude						Coming Up					
Axial	Shaft	Bearings	Casings	Foundation	Piping	Coupling	Stays Same	Increases	Decreases	Comes Suddenly	Drops Out Suddenly
10	80	20	..	..	..	..	..	..	..	10	90
10	60	40	..	..	..	80	..	..	20	..	..
20	20	10	20	30	20	80	..	20	20	..	..
10	80	20	..	..	..	..	..	..	..	..	100
10	20	20	20	20	20	..	..	..	..	20	80
30	40	20	20	10	..	10	..	..	..	..	20
10	70	10	10	..	..	10	..	..	..	20	60
	100	Lateral amplitude	..	..	..	30	20	..	..	20	30
							40				
	Torsion 100	40	40	..	..	50	30	..	..	30	20
							..	..			41

Numbers indicate percent of cases showing above symptoms, for causes listed in vertical column at left.
Source: From *The Practical Vibration Primer* by Charles Jackson. Copyright © 1979 by Gulf Publishing Company, Houston, Texas. Used with permission.
All rights reserved.

TABLE 36.5 Vibration Analysis Symptoms (*Continued*)

check the frequency. May require antwhirl bearings or tilting-shoe bearings. Check especially for resonances at whirl frequency (or multiples) in foundation and piping.

13. This can excite resonances and criticals and combinations thereof at two times running frequency. Usually difficult to field balance because, when horizontal vibration improves, vertical vibration gets worse and vice versa. It may be necessary to increase horizontal bearing support stiffness (or mass) if the problem is severe.
14. Usually the result of slugging the machine with fluid, solids buildup on rotor, or off-design operation (especially surging.)
15. The frequency at rotor support critical is characteristic. Disks and sleeves may have lost their interference fit by rapid by rapid temperature changes. Parts usually are not loose at standstill.
16. Often confused with oil whirl because the characteristics are essentially the same. Before suspecting any whirl, sure everything in the bearing assembly is absolutely tight with an interference fit.
17. This should always be checked.
18. Usually involves sliding pedestals and casing feet. Check for friction, proper clearance, and piping strains.
19. To obtain frequencies, tape a microphone to the gear case and record noise on magnetic tape.
20. Loose coupling sleeves are notorious troublemakers, especially in conjunction with long, heavy spacers. Check tooth fit by placing indicators on top, then lifting by hand or a jack and noting looseness [should not be more than 1 to 2 mils (0.025–0.05 mm) at standstill, at most]. Use hollow coupling spacers. Make sure coupling hubs have at least 1 mil/in (1 mm/m) interference fit on shaft. Loose hubs have caused many shaft failures and serious vibration problems.
21. Try field balancing; more viscous oil (colder); larger, longer bearings with minimum clearance and tight fit; stiffen bearing supports and other structures between bearing and ground. This is basically a design problem. It may require additional stabilizing bearing or a solid coupling. It difficult to correct in the field. With high-speed machines, adding mass at the bearing case helps considerably.
22. These are critical of the spacer-teeth-overhang subsystem. Often encountered with long spacers. Make sure of tight-fitting teeth with a slight interference at standstill and make the spacer as light and stiff as possible (tubular). Consider using a solid or membrane coupling if the problem is severe. Check coupling balance.
23. Overhang critical can be exceedingly troublesome. Long overhangs shift the nodal point of the rotor deflection line (free-free mode) toward the bearing, robbing the bearing of its damping capability. This can make critical speeds. Shorten the overhang or put in an outboard bearing for stabilization.
24. Casing resonance is also called case drumming. It can be very persistent, but is sometimes harmless. The danger is that parts may come loose and fall into the

- machine. Also, rotor/casing interaction may be involved. Diaphragm drumming is serious, since it can cause catastrophic failure of the diaphragm.
25. Local drumming is usually harmless, but major resonances, resulting in vibration of the entire case as a unit, are potentially dangerous because of possible rubs and component failures, as well as possible excitation of other vibrations.
 26. Similar problems as in 24 and 25 with the added complications of settling, cracking, warping, and misalignment. This cause may also produce piping troubles and possible case warpage. Foundation resonance is serious and greatly reduces unit reliability.
 27. Pressure pulsations can excite other vibrations with possible serious consequences. Eliminate such vibrations using restraints, flexible pipe supports, sway braces, shock absorbers, etc., plus isolation of the foundation from piping, building, basement, and operating floor.
 28. Occurs mostly at two times the line frequency (7200 cpm), coming from motor and generator fields. Turn the fields off to verify the source. Usually harmless, but if the foundation or other components (rotor critical or torsional) are resonant, the vibrations may be severe. There is a risk of catastrophic failure if there is a short circuit or other upsets.
 29. This can excite serious vibrations or cause bearing failures. Isolate the piping and foundation and use shock absorbers and sway braces.
 30. Valve vibration is rare but sometimes very violent. Such vibrations are aerodynamically excited. Change the valve shape to reduce turbulence and increase rigidity in the valve gear. Make sure the valve cannot spin.
 31. The vibration is exactly one-half, one-quarter, and one-eighth of the exciting frequency. It can be excited only in nonlinear systems; therefore, look for such things as looseness and aerodynamic or hydrodynamic excitations. It may involve rotor "shuttling." If so, check the seal system, thrust clearances, couplings, and rotor-stator clearance effects.
 32. The vibrations are at two, three, and four times the exciting frequency. The treatment is the same as for direct resonance: change the frequency and add damping.
 33. If the cause is intermittent, look into temperature variations. Usually the rotor must be rebuilt, but first try to increase stator damping, add larger bearings (tilting-shoe), increase stator mass and stiffness, and improve the foundation. This problem is usually caused by maloperation, such as quick temperature changes and fluid slugging. Use membrane-type coupling.
 34. This is basically a design problem, but is often aggravated by poor balancing and a poor foundation. Try to field-balance the rotor at operating speed, lower oil temperature, and use larger and tighter bearings.
 35. Add mass or change stiffness to shift the resonant frequency. Add damping. Reduce excitation and improve system isolation. Reducing mass or stiffness can leave the amplitude the same even if resonant frequency shifts because of stronger amplification. Check "mobility."

36. Stiffen the foundation or bearing structure. Add mass at the bearing, increase critical speed, or use tilting-shoe bearings (which is the best solution). First, check for loose fit of bearings in bearing case.
37. Same comments as 36 with additional resonance of rotor, stator, foundation, piping, or external excitation; find the resonant members and the sources of excitation. Tilting-shoe bearings are the best. Check for loose bearings.
38. Sometimes you can hear the squeal of a bearing or seal, but frequency is usually ultrasonic. Very destructive, Check for rotor vanes hitting the stator, especially if clearances are smaller than the oil film thickness plus rotor deflection while passing through the critical speed.
39. Usually accompanied by rocking motions and beating within clearances. It is serious especially in the bearing assembly. Frequencies are often below running frequency. Make sure everything is absolutely tight, with some interference. Line-in-line fits are usually riot sufficient to positively prevent this type of problem.
40. This problem is very destructive and difficult to end. The symptoms are gear noise, wear on the back side of teeth, strong electrical noise or vibration, loose coupling bolts, and fretting corrosion under the coupling hubs. There is wear on both sides of coupling teeth and possibly torsional-fatigue cracks in keyway ends. The best solution is to install properly tuned torsional vibration dampers.
41. Similar to 40 but encountered only during start-up and shutdown because of very strong torsional pulsations. It occurs in reciprocating machinery and synchronous motors. Check for torsional cracks.

Table 36.6 lists the pump operation modes for a range of unfiltered bearing cap velocities.

36.6.2 Impeller Unbalance

Impeller unbalance generates vibrations normally at 1× operating speed. This problem is a main cause of the failure in the following components:

1. Mechanical seals
2. Bearings

Smooth	Acceptable	Marginal	Planned Shutdown for Repairs	Immediate Shutdown
0.1 in/s (p) and less	0.1–0.2 in/s (p)	0.2–0.3 in/s (p)	0.3–0.5 in/s (p)	0.5 in/s (p)

Note: For gearing, add 0.1 in/s to all values.

P = peak

mm/s = 25.4 × in/s

TABLE 36.6 Bearing Cap Data—Velocity Unfiltered

The impeller should be inspected during every pump overhaul. Pitting may appear in some areas around the impeller. This is due to nonhomogeneous nature of most castings. This allows corrosion to be more aggressive in one area of the impeller. Unbalance occurs due to uneven etching or surface pitting in the impeller. All impellers larger than 25 cm (10 in) should be balanced in the shop at operating speed.

36.6.3 Hydraulic Unbalance

Vibration occurs when the flow entering the impeller is not evenly distributed. This vibration appears at 1× operating speed. The installation of an elbow in the horizontal plane followed by a short, straight run to a double-suction pump can generate this vibration. This is due to the difference between the flows entering both sides of the impeller. The flow does not have enough time to straighten before it enters the pump in this case. This condition will also occur if the pump suction is taken from a tee. Axial thrust on the impeller and high axial vibrations will occur due to this problem. This problem can be avoided by installing the elbows vertically in double-suction pumps.

Noise and/or vibration may occur when the flow is lower than the design flow. Pressure pulsations and recirculation may also occur in this condition. The increase in the net positive suction head available (NPSHA) has helped to reduce these problems in some applications.

36.7 Bibliography

Karassik, I. J., Krutzsch, W. C., Fraser, W. H., and Messina, J. P., *Pump Handbook*, 2nd ed. McGraw-Hill, New York, 1986.

CHAPTER 37

Water Hammer

37.1 Introduction

Water hammer is a pressure surge. This phenomenon can occur in any pumping installation. It is caused by abrupt changes to the flow rate (e.g., when the fluid-flow rate is forced to stop or change direction suddenly). Water hammer commonly occurs when a valve is closed suddenly. For example, a valve closing within 1.5 seconds can generate a pressure spike (acoustic wave) five times higher than the system operating pressure. Pressure waves propagate in the piping system as a result of the sudden closure of the valve. Water hammer can generate a highly destructive force. This force can lead to the following:

1. Blown diaphragm
2. Failure of seals, gaskets, instrumentation, expansion joints, and pipes

37.2 Nomenclature

The following is a list of the variables used in this chapter. Table 37.1 provides the SI conversion factors.

- a = velocity of pressure wave, ft/s
 D = inside diameter of conduit, ft
 e = thickness of pipe wall, ft
 E = Young's modulus for pipe material, lb/ft²
 g = acceleration of gravity, ft/s²
 H_0 = pumping head for initial steady pumping conditions, ft
 H_R = rated pumping head, ft
 $K_1 = 91,600 H_R Q_R / WR_R^2 N_R^2, s^{-1}$
 K = volume modulus of liquid, lb/ft²
 L = total length of conduit, ft
 $2L/a$ = round-trip wave travel time, s
 N_R = rated pump speed, rpm
 η_R = pump efficiency at rated speed and head, decimal form
 ρ = pipe line constant = $a\bar{V}_0/gH_0$

To Convert	To	Multiply by
$CD^2(\text{kg} \cdot \text{m}^2)$	$WR^2(\text{lb} \cdot \text{ft}^2)$	23.73
kg/m^2	lb/ft^2	0.2048
kg/m^3	lb/ft^3	0.06243
m	ft	3.281
m/s	ft/s	3.281
m/s^2	ft/s^2	3.281
m^3/s	ft^3/s	35.32
mm	ft	3.281×10^{-3}

TABLE 37.1 SI Conversions

- Q_0 = initial flow through pump, ft^3/s
- Q_R = rated flow through pump, ft^3/s
- μ = Poisson’s ratio of pipe material
- V_0 = velocity in conduit for initial steady conditions, ft/s
- w = specific weight of water, lb/ft^3
- WR^2 = flywheel effect of rotating parts of motor, pump, and entrained water, $\text{lb} \cdot \text{ft}^2$

37.3 Basic Assumptions

The following assumptions were made in this chapter:

1. The fluid in the pipe system is elastic, of homogeneous density, and always in the liquid state.
2. The pipe wall material or conduit is homogeneous, isotropic, and elastic.
3. The velocities and pressures in the pipeline, which is always flowing full, are uniformly distributed over any transverse cross section of the pipe.
4. The velocity head in the pipeline is negligible relative to the pressure changes.
5. At any time during the pump transient, when operation is in the zones of pump operation, energy dissipation, and turbine operation, there is an instantaneous agreement at the pump, as defined by the steady-state complete pump characteristics, of the pump speed and torque corresponding to the transient head and flow which exist at that moment at the pump.
6. The length between the inlet and outlet of the pump is so short that water hammer waves propagate between these two points instantly.
7. Windage effects of the rotating elements of the pump and motor during the transients are negligible.
8. Water levels at the intake and discharge reservoirs do not change during the transient period.

37.4 Effects of Water Hammer in High- and Low-Head Pumping Systems

The effects of water hammer are more significant in low-head pumping systems than high-head pumping systems. This is because the relative increase in pressure due to a drop in flow velocity in low-head pumping systems is much greater than high-head pumping systems. (The pressure increase in the pumping system is proportional to the square of the drop in flow velocity.) This is the first law of thermodynamics. Thus, since the flow velocity is about the same in low-head and high-head pumping systems, the pressure increase will also be the same due to the same drop in flow velocity. However, since the pressure is lower in low-head pumping systems than high-head pumping systems, the relative increase in pressure is greater in low-head pumping systems than high-head pumping systems.

37.4.1 Magnitude of the Pulse

The magnitude of the pressure pulse caused by water hammer can be calculated using the Joukowsky equation:

$$\Delta P = \rho a \Delta c$$

where ΔP = magnitude of the pressure wave, Pa

ρ = density of the fluid, kg/m³

a = speed of sound in the fluid, m/s

Δc = change in fluid velocity, m/s

The peak pressure reached in the system depends on the compressibility of the fluid. The following equation provides the maximum pressure in a water filled line:

$$P = 0.07 VL/t + P_1$$

where P_1 = inlet pressure

V = flow velocity, ft/s

t = valve closing time, s

L = length of the upstream pipe, ft

Example 37.1 Calculate the pressure spike that will occur in a water-filled line when the solenoid valve closes. The system has the following characteristics:

$L = 60$ ft

$V = 5.0$ ft/s (recommended velocity for PVC piping design)

$t = 40$ ms (solenoid valve closing time)

$P_1 = 60$ psi (inlet pressure)

$$P = 0.07 \times 5 \times 60 / 0.04 + 60 = 585 \text{ psi}$$

Solution Pipe-sizing charts recommend a flow velocity at or below 1.5 m/s (5 ft/s) for some applications. This is done to reduce the magnitude of the pressure pulse that would occur when the valve is closed suddenly.

The pressure pulse and shock waves caused by water hammer generate a loud bang that resembles a hammering noise. Water hammer can break pipelines. The force of the moving fluid is cushioned using air traps and stand pipes (open at the top) in some applications. (What appears to be a water tower in some hydroelectric generating stations is actually one of these devices.)

Water hammer can cause a catastrophic failure of water pipes in hydroelectric power plants. These plants must be carefully designed and maintained to prevent water hammer damage.

The fluid downstream of the valve will continue to flow following the closure of the valve. This creates a vacuum that may cause the pipe to collapse or implode. This problem can be significant if the pipe is on a downhill slope. Air and vacuum relief valves, or air vents, are installed just downstream of the valve to prevent this problem. These devices allow air to enter the line to prevent this problem.

37.4.2 Possible Causes of Water Hammer

The following are the possible causes of water hammer:

1. Sudden valve closure
2. Pump failure
3. Check valve slamming shut rapidly (this is due to sudden deceleration of the flow)

37.4.3 Mitigating Measures to Water Hammer

Water hammer damage has normally been limited to pipe and equipment failure. However, water hammer has caused accidents and fatalities in some cases. The risk of a pipeline failure should be assessed during the design phase. Special care should be given to systems containing hazardous fluids. The following features are used to reduce or eliminate water hammer problems:

1. Reduce the flow velocity.
2. Increase the time required to close valves (e.g., a pulsing circuit is used in electric actuators. It uses an electronic timer to interrupt the current to the motor periodically. For example, it allows the current to reach the motor for 5 seconds. It then stops the current to the motor for 5 seconds. The cycle is then repeated until the valve is closed).
3. Increase the pressure rating of the pipelines. However, this feature increases the capital cost of the system.
4. Install water towers in the system. This feature is used in many drinking water system applications. These towers help in maintaining a steady flow rate. They also prevent large pressure variations in the system.
5. Install accumulators in the system. These accumulators are air vessels. They are similar to water towers. However, they have a pressurized air cushion above the fluid vessel. A bladder is used to separate the air and fluid in the vessel.
6. Install an uninterruptible power system (UPS). This feature has the following advantages:
 - a. The system is shut down properly.
 - b. The valves are closed within the required times.
7. Install a flywheel on the pump shaft. This feature ensures that the flow does not stop suddenly following an interruption of power.

37.4.4 Applications of Water Hammer

Water hammer has been used in the following applications:

1. To create a pump. This pump is called hydraulic ram.
2. To detect leaks from systems.

37.5 Power Failure at Pump Motors

37.5.1 Pumps with No Valves at the Pump

Following a power interruption to the pump motor, the only energy remaining to drive the rotor (for the pump motor assembly) is its kinetic energy. The rotor speed drops quickly following the interruption. This is because the kinetic energy of the rotor is relatively small relative to that required to maintain the flow against the discharge head.

Water hammer waves of increasing subnormal pressure are formed in the discharge line of the pump due to this sudden change in flow. These subnormal pressure waves propagate quickly through the discharge line toward the discharge outlet. Complete wave reflections occur at the discharge outlet. The reduction in pump speed prevents the flow from moving forward against the existing head. This occurs within a short period of time after the power interruption. The flow through the pump will reverse direction if there is no control valve at the pump. This occurs while the pump is still rotating in the forward direction in some cases. The speed of the pump will drop at a higher rate at this stage. The pump will then reverse direction. The maximum reverse flow will pass through the pump. The pump will reach the runaway speed in reverse, a short time later. The pump will now be acting as a turbine. The reverse flow through the pump will drop when it approaches runaway speed. The pressure at the discharge of radial-flow pumps and along their discharge line will increase. This is due to the rapid reduction in reverse flow.

Figures 37.1 to 37.8 illustrates the results of a large number of water hammer cases. These results are for a set of radial-flow pumps. These pumps have low specific speed. These results can be used to determine the pressure transient that occurs at a radial-flow pump and its discharge line when there are no control valves associated with this pump.

The following two independent parameters are used in these charts:

1. ρ , the pipeline constant
2. $K(2L/a)$, this is a constant that includes the following effects:
 - a. Pump and motor inertia
 - b. Water hammer wave travel time of the pump discharge line

The maximum head at the pump with reverse flow will not usually exceed the initial pumping head under the following conditions:

1. The frictional head in the pump discharge line is more than 25% of the total pumping head during normal operation.
2. Separation of the liquid column does not occur in the discharge line.

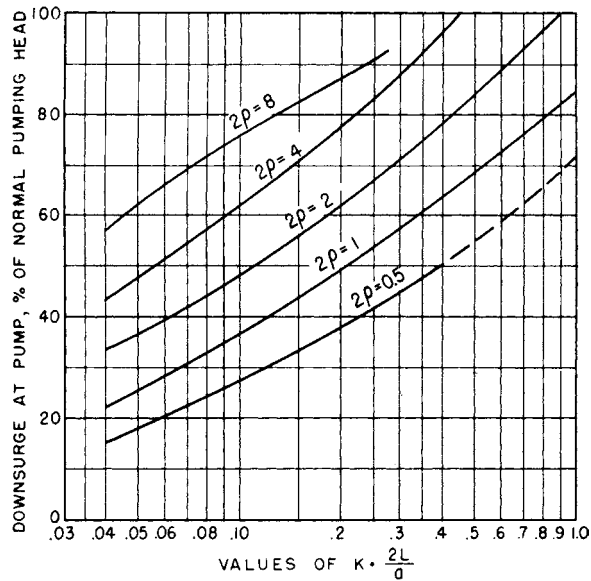


FIGURE 37.1 Downsurge at pump. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

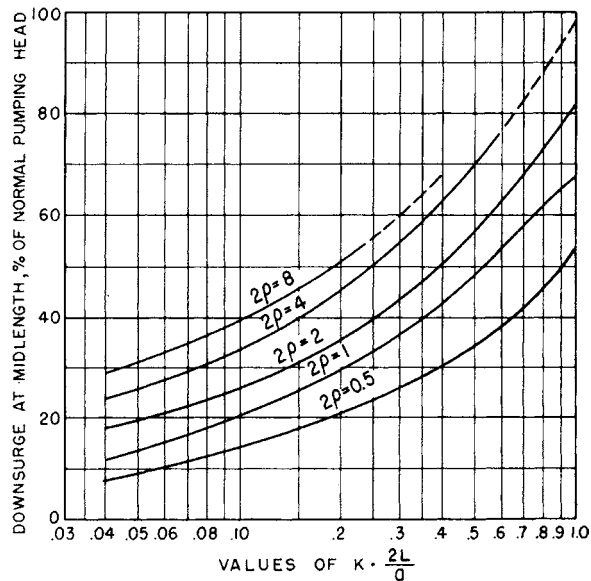


FIGURE 37.2 Downsurge at midlength. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

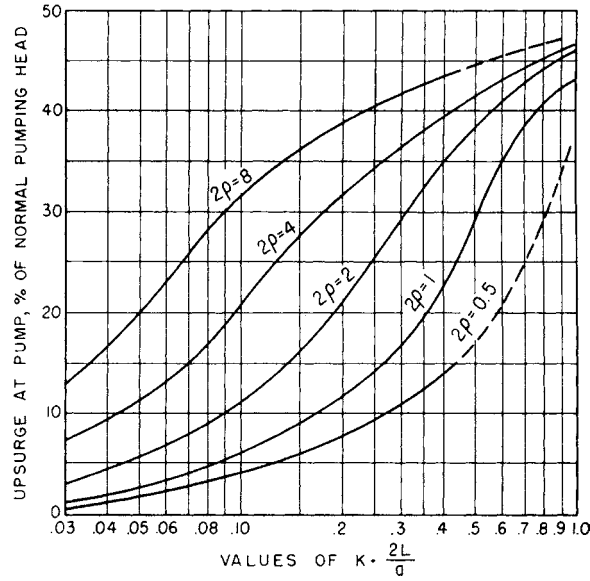


FIGURE 37.3 Upsurge at pump. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

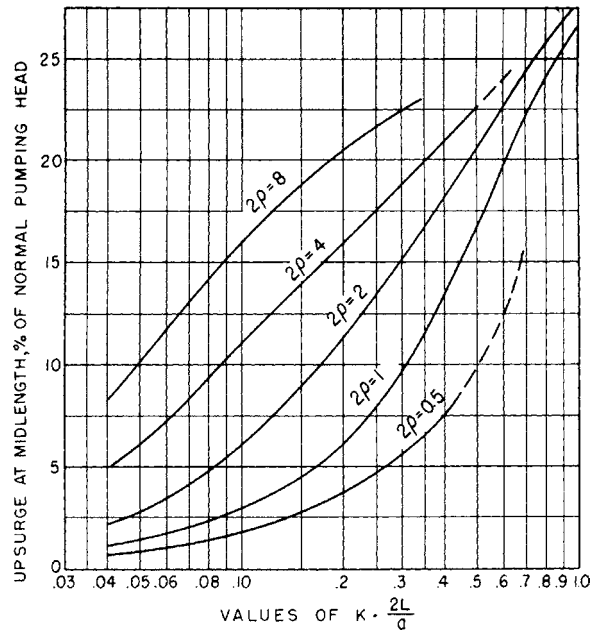


FIGURE 37.4 Upsurge at midlength. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

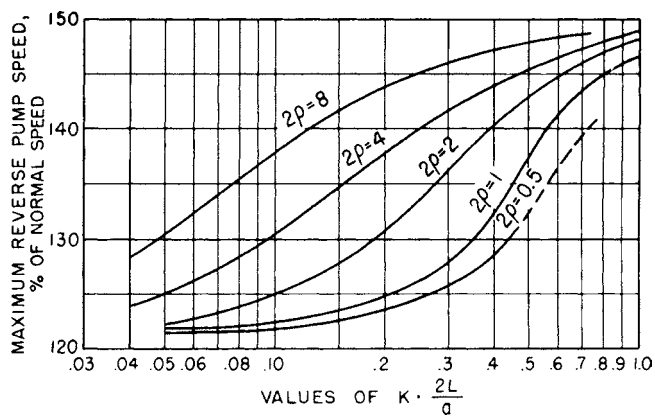


FIGURE 37.5 Maximum reverse speed. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

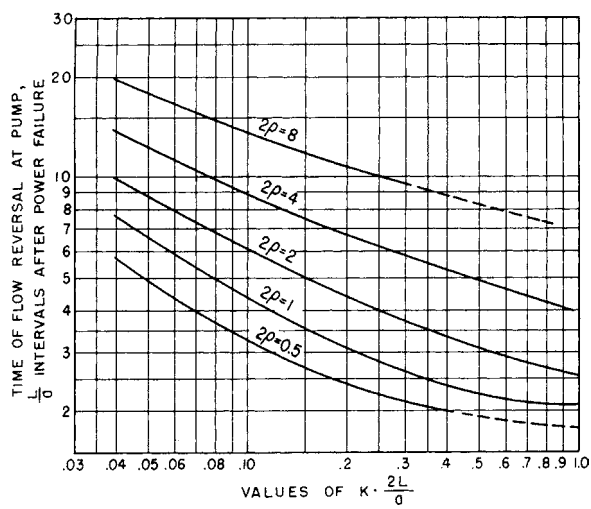


FIGURE 37.6 Time of flow reversal at pump. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

37.5.2 Pumps Equipped with Check Valves

Check valves installed in the pump discharge lines can generate pressure transients. The pump discharge flow keeps the check valve open during normal operation. However, following a power failure, the pump flow reverses direction. This results in a rapid closure of the check valve. The head will increase in the discharge line following the closure of the valve. This head will almost reach the head drop which existed when the flow reversal occurred. However, the maximum increase of the head in the

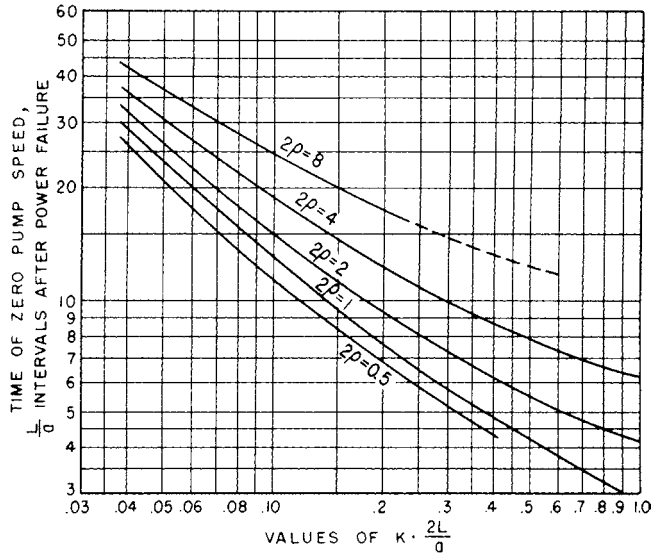


FIGURE 37.7 Time of zero pump speed. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

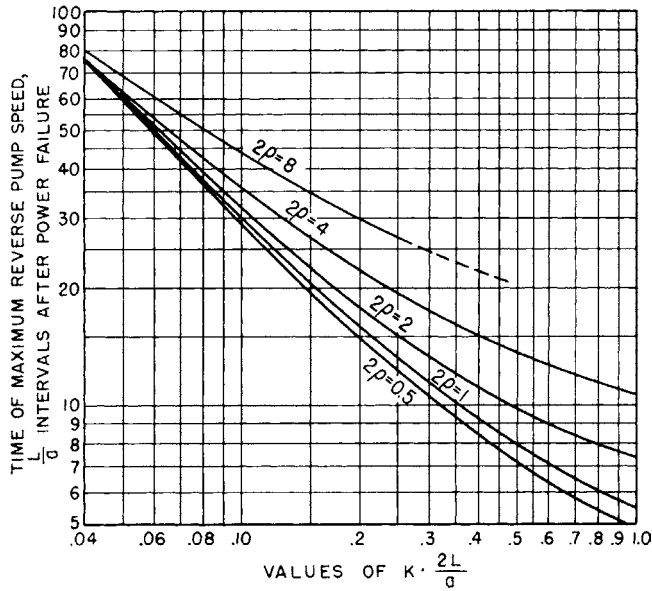


FIGURE 37.8 Time of maximum reverse pump speed. (Source: Reprinted with permission from Karassik, I., *Pumps Handbook*, 4th ed., McGraw-Hill, New York, 2008, with permission from McGraw-Hill.)

discharge line will be significantly higher than the head drop which existed when the flow reversal occurred if the closure of the check valve is delayed due to any of the following reasons:

1. Hinge friction
2. Valve malfunction
3. Valve characteristics

The head increase in the pump discharge line will be significantly lower than the one obtained when the check valve closes momentarily upon flow reversal if the check valve is closed slightly ahead of the flow reversal. Several manufacturers incorporate this feature in their check valves. This includes a spring-loaded or lever-arm-weighted devices on the check valve hinge pins. This feature closes the valve disk ahead of the flow reversal.

The following are types of check valves:

1. Rapid-closing check valves
2. Slow-closing check valves

Rapid-closing check valves are the best option for pump discharge lines. This is because they prevent reverse flow upon a power failure.

37.5.3 Controlled Valve Closure

The head rise that occurs in pump discharge lines following a power failure is normally limited in most applications by using single-speed discharge valve closure. However, there will be some reverse rotation below the maximum runaway speed in some cases. A two-speed valve closure is used in some cases to prevent or limit the reverse speed. This feature closes the valve very rapidly for most of its stroke when the flow reverses at the pump. The valve is closed slowly for the remainder of its stroke. This is done to reduce the pressure rise in the pump discharge line.

37.5.4 Surge Suppressors

Surge suppressors are devices used to control the pressure increase that occurs in pump discharge lines following a power interruption. The surge suppressor includes a pilot-operated valve. This valve opens quickly following a power interruption due to any of the following reasons:

1. Loss of power to a solenoid valve
2. Sudden pressure increase at the surge suppressor
3. Sudden pressure decrease at the surge suppressor

This valve releases the liquid from the pump discharge line to a collection system. The valve is slowly closed later using a dashpot. This is done to limit the pressure increase as the flow decreases. The pressure increase in a pump discharge line can be limited to any desired value using a proper-sized and field-adjusted surge suppressor.

Improper field adjustments of a surge suppressor can have serious consequences. For example, the down surge at the pump and in the discharge line will be larger than a system that does not have a surge suppressor if the surge suppressor opens immediately following a power interruption. Water column separation (refer to Sec. 37.5.5) could occur in this case. A significant pressure increase will occur in the system if the surge suppressor closes quickly when the maximum reverse flow has been established.

37.5.5 Water Column Separation

Water column separation occurs in a pump discharge line when the pressure drops below the vapor pressure (saturation pressure) of the liquid. Normal water hammer solutions cannot be used in this case. The liquid in the pump discharge line becomes separated by a section of liquid and vapor when this condition persists for a period of time. Significant pressure increase normally occurs when the two liquid columns rejoin. Thus, this condition should be avoided.

37.5.6 Quick-Opening, Slow-Closing Valves

The pressure increase that occurs at the high points in the pump discharge line can be limited using a quick-opening, dashpot-controlled, slow-closing valve. Liquid column separation occurs frequently at these locations. The valve opens quickly when the pressure in the region of liquid column separation drops below its set point. This allows a small amount of air to enter the pipeline. The valve should be in the fully open position when the upper water column in the pipeline reverses direction and returns to the separation area near the valve. The air and liquid mixture will discharge from the valve. This valve reduces the pressure increase resulting from the rejoining of the liquid columns. The valve is closed later slowly using a dashpot. This is done to reduce the pressure increase that occurs in the pipeline near the valve when the flow is interrupted. These valves should be sized properly. They should also be field-adjusted properly to open and close at the right times. The valves should be protected against freezing as well.

37.5.7 One-Way Surge Tanks

The one-way surge tank is a device used to control the pressure. This feature is used in applications where water column separation occurs. The one-way surge tank is relatively small. This tank is filled partially by the same liquid being pumped in the system. The pressure in this tank is lower than the system pressure. The one-way surge tank is connected to the system with check valves. These valves are held closed by the system pressure. The pressure in the system drops below the pressure in the one-way surge tank following a power failure. This opens the check valve connecting the one-way surge tank and the system. The one-way surge tank starts to drain. This fills the void created by the separation of the water columns in the system. The check valve at the one-way tank will close when the flow reverses direction. Thus, the system pressure will not increase due to the rejoining of the water columns. A float control valve is used normally to maintain the initial level of liquid in the one-way surge tank. The one-way surge tank will not act during the start-up phase of the pump. This tank should also be protected from freezing.

37.5.8 Air Chambers

The air chamber is used to limit the pressure increase in a long pump discharge line. This device is also known as a hydro-pneumatic tank. The air chamber contains the following:

1. Water in its lower portion
2. Compressed air in its upper portion

This device is normally located near the pumps. It can be installed vertically, horizontally, or in a sloping position. The water level in the chamber is maintained using a float level control system and compressed air. The head and flow downstream of the pump drop quickly following a power interruption to the pump motor. This allows the compressed air in the chamber to force water out into the pump discharge line. This minimizes the change in velocity and water hammer effects in the pump discharge line. The pump will not be able to deliver the flow against the system head following the interruption of power. This normally occurs within a fraction of a second after the interruption of power to the pump motor. The check valve located downstream of the pump closes rapidly at this stage. The pump shaft stops rotating quickly. The liquid in the pump discharge line will stop flowing. This liquid will reverse direction a short time later. It will start to flow back into the air chamber. The volume of air in the chamber decreases when the reverse flow enters it. The air flows out of the chamber through a throttling orifice. The head in the chamber increases above the head that was in the pump discharge line. This increase in the head depends on the following:

1. The throttling orifice
2. The initial volume of air in the chamber

37.5.9 Surge Tanks

The surge tank is a device used to reduce water hammer caused by the rapid changes of the flow in a pump discharge line following a power interruption to the motor. The surge tank is a very reliable device. This is because it does not have any moving parts which can fail. The liquid in the surge tank provides a source of potential energy following the power failure to the motor. This will reduce the rate of change in the flow and water hammer in the pump discharge line. The main disadvantages of the surge tank are its size and cost. This is because the tank must be very tall in high-head pumping applications.

37.5.10 Nonreverse Ratchets

The nonreverse ratchet prevents reverse rotation of the pump upon a power failure. This device is used to reduce water hammer in the pump discharge line. This device has proved to be effective in limiting water hammer. The nonreverse ratchet has also another significant advantage. This device protects the motor. The inrush current drawn by the motor will damage the motor significantly if the motor were started while the pump is rotating in reverse. The nonreverse ratchet reduces the inrush current drawn by the motor by preventing the shaft from rotating in the reverse direction.

37.6 Normal Pump Shutdown

During a normal pump shutdown, the control valve in the pump discharge line should be closed first. The power to the pump motor should then be interrupted. This sequence of events results in the least water hammer effect in the pump discharge line. However, if the discharge side of the pumps has check valves and the power to one of several pump motors connected to the same discharge line is interrupted, the flow will reverse rapidly at the pump which has been shutdown. The check valve in the discharge line of this pump will close rapidly due to the reverse flow. Anti-slam or slow-closing features of the check valves should be used in this case. These devices will reduce the water hammer effect in the discharge line.

37.7 Water Hammer Example

Example 37.2 Consider a power failure at the pumping plant installation shown in Fig. 37.9. This installation consists of three pumps which discharge into a steel pipeline. Aside from isolation valves, there are no check valves in the system. The basic data for this installation are as follows:

$$D = 32 \text{ in}; e = 3/16 \text{ in}; Q_0 = Q_R = 33.7 \text{ ft}^3/\text{s} \text{ (for three pumps)}$$

$$V_0 = 6.03 \text{ ft/s (for three pumps)}; H_0 = H_R = 220 \text{ ft}$$

$$400\text{-hp motors at each pump}$$

$$WR^2 \text{ of each pump and motor} = 385 \text{ lb} \cdot \text{ft}^2$$

$$N_R = 1760 \text{ rpm}; \eta_R = 0.847$$

$$\frac{D}{e} = \frac{32}{0.1875} = 171$$

$$Ma = 3000 \text{ ft from Fig. 37.10}$$

$$\frac{2L}{a} = \frac{2(3940)}{3000} = 2.63 \text{ seconds}$$

$$2p = \frac{aV_0}{gH_0} = \frac{(3000)(6.03)}{(32.2)(220)} = 2.55$$

$$K = \frac{(91,600)(H_R Q_R)}{WR_{\eta R}^2 N_R^2} = \frac{(91,600)(220)(33.7)}{3(385)(0.847)(1760)^2} = 0.224$$

$$K \frac{2L}{a} = 0.59$$

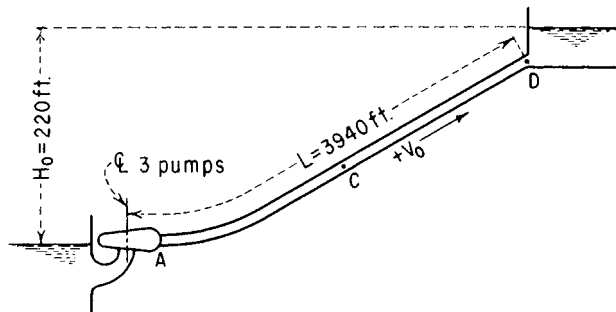
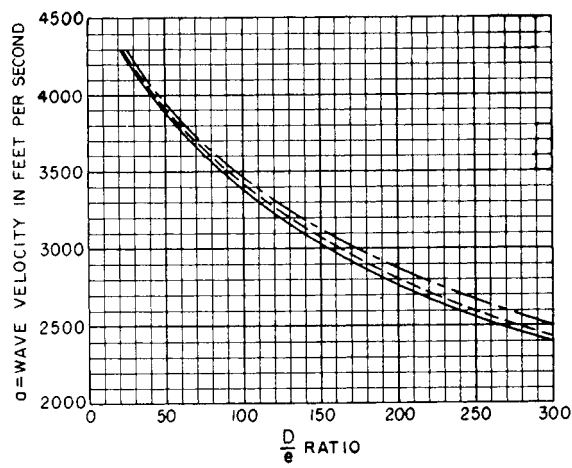


FIGURE 37.9 Pipeline profile.



$$a = \sqrt{\frac{1}{\frac{w}{g} \left(\frac{1}{K} + \frac{Dc_1}{Ee} \right)}}$$

where:

- a = wave velocity (ft. per sec.)
- g = acceleration of gravity (ft. per sec²)
- $\frac{D}{e}$ = $\frac{\text{diameter of pipe}}{\text{thickness of pipe}}$
- E = Young's modulus for steel pipes = 4.32×10^9 (lb. per ft.²)
- K = volume modulus of water = 43.2×10^6 (lb. per ft.²)
- w = 62.4 = specific weight of water (lb. per ft.³)
- μ = 0.3

$c_1 = \frac{5}{4} - \mu$
 $c_1 = 1 - \mu^2$
 $c_1 = 1 - \frac{\mu}{2}$

FIGURE 37.10 Pressure wave velocity in steel pipes.

From Figs. 37.1 to 37.8 the following results are obtained:

1. Downsurge at pump = $(0.92)(220) = 202$ ft
2. Downsurge at midlength = $(0.64)(220) = 141$ ft
3. Upsurge at pump = $(0.42)(220) = 92$ ft
4. Upsurge at midlength = $(0.23)(220) = 51$ ft
5. Maximum reverse speed = $(1.45)(1760) = 2550$ rpm
6. Time of flow reversal at pump = $3.5 L/a = 4.6$ seconds
7. Time of zero pump speed = $5.8 L/a = 7.6$ seconds
8. Time of maximum reverse speed = $10.0 L/a = 13.1$ seconds

As noted in the discussion on pumps equipped with check valves, the upsurge, or head rise, at the pump above the normal head would have been about 202 ft if there were check valves at the pumps which closed at the time of flow reversal.

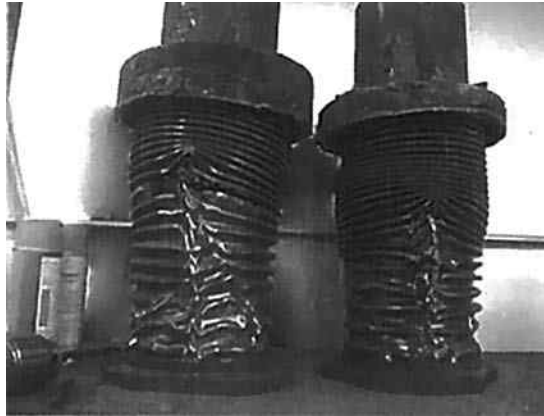


FIGURE 37.11 Expansion joints on a steam line that have been destroyed by steam hammer.

37.8 Steam Hammer

Steam hammer occurs in steam distribution systems. This phenomenon is similar to water hammer. The steam condenses in horizontal sections of the steam piping in some applications. The flowing steam picks up the condensed water. This water is now in the form of slugs. These slugs will entrain with the steam at high velocity. They can cause damage to the equipment downstream. Figure 37.11 illustrates steam hammer damage to expansion joints. Steam hammer generates a loud hammering noise and great stresses on the pipes. This condition is normally caused by poor design or malfunction of the condensate drainage system.

37.9 Bibliography

Karassik I. J., Krutzsch W. C., Fraser W. H., and Messina J. P., *Pump Handbook*, 2nd ed. McGraw-Hill, New York, 1986.

Wikipedia encyclopedia, Water hammer. http://en.wikipedia.org/wiki/Water_hammer.

This page intentionally left blank

Selection and Procurement of Pumps

38.1 Introduction

The following are the sequence involved in obtaining a pumping system:

1. Engineering of system requirements
2. Selection of pump and driver
3. Specification of pump
4. Bidding and negotiation
5. Evaluation of bids
6. Purchasing of selected pump

38.2 Engineering of System Requirements

The first decision is to determine the requirements and conditions under which the pumping system must operate.

38.2.1 Fluid Type

The development of the following physical and chemical data of the fluid handled is the first step: viscosity, density, corrosiveness, lubricating properties, chemical stability, volatility, and amount of suspended particles. These properties will have a significant influence on pump and system design. For example, the choice of materials depends on the level of corrosiveness of the fluid. Also, if the fluid contains suspended solids, suitable seal designs and abrasion-resistant construction materials are required.

The physical and chemical properties should cover the entire operating range of the pumping system. The effect of temperature, pressure, and time on the fluid properties should also be considered.

38.2.2 System-Head Curves

A preliminary design of the system should be done, including the equipment layout, piping and instrumentation diagram showing the various flow paths, their size and length, elevation of system components, all valves, equipment, piping specialties, etc. which establish the system-head losses.

The next step is to determine the flow paths, flow rates, pressures, and temperatures for all system operating conditions, and calculate pipe sizes and estimate pipe length. This information is required to develop system-head curves that illustrate the relationship between flow and hydraulic losses in the piping system. Allowances for corrosion and scale deposits must be included in calculating the hydraulic losses in the system over the life of the plant. The effect of static pressure and elevation differences should be added to the system-head curve to form a combined system-head curve. The total head required by the pumping system to overcome system resistance will be obtained from the resultant curve. The pump head must be at or higher than the combined system curve for all operating conditions and flow paths.

38.3 Alternate Modes of Operation

The following considerations should be examined when specifying a pumping system:

1. Is the pump operating continuously or intermittently?
2. Is the flow or head changing or constant?
3. Would the different flow paths require great difference in flow and head?

These and other questions originating from different modes of operation have a great influence on the number of pumps, their capacities, and whether booster pumps are required for some flow paths. Normally, reliability considerations will require the use of multiple pumps, such as two full-size pumps ($2 \times 100\%$) or three half-size pumps ($3 \times 50\%$). An impairment of a pump can occur due to a pump malfunction or a motor failure, loss of power supply, external damage, bearing, or seal failure, etc. The probability of these failures should be evaluated as part of the pump selection process.

38.4 Margins

Pumps are normally specified with margins above the normal rating. During normal operation, disturbances such as these can occur:

1. Changes in operating modes
2. Component failure
3. Electric system disturbances (under-voltage, etc.)

The probability and duration of these transients should be examined to specify adequate margins to minimize their effects on the pumping system. This includes an evaluation of these combined effects:

1. Equipment cost
2. Degree of criticality of the system
3. Consequences of unavailability of the equipment (impairment of the pumping system)

The following transients are normally considered:

1. Pressure and temperature fluctuations
2. Voltage and frequency variations
3. Loss of cooling water

Adequate margins should be incorporated in the pump rating if it is important to maintain a continuous flow. For example, margins are included in the pump head and capacity to allow the pumping system to maintain rated flow when the frequency or voltage drops. Other margin features may also be included to allow the pumps to operate without damage through transients such as cavitation. Pumps should not be purchased for capacities greater than 15 to 20% of requirements. Oversize pumps are more expensive and may require a higher voltage level.

38.5 Wear

The operating characteristics of the pumping system can be expected to change due to wear. The extent of wear over the life of the plant should be evaluated and adequate margins should be included to ensure that the pumps can provide the expected flow even at the end of the system life. Replaceable liners should be specified when abrasive or suspended materials are handled. These liners are usually made of resilient material, such as rubber compounds, or extremely hard alloys of cast iron. In addition, plastic lining (including impellers) is frequently used for these types of services. In applications that do not handle abrasive, suspended, or corrosive materials, the amount of wear is usually minimal over the life of the pump. The head-flow characteristics of the pump are not expected to be affected in this type of service. However, if cavitation occurs, component wear will have significant effect on the pump characteristics. In some applications, especially in power plants, the expected life of the pump is specified the same as plant life. However, the design life of a pump is based on evaluating economic factors. The wear margin included depends on factors such as mode of operation and fluid properties (abrasiveness, corrosiveness) and cavitation.

38.6 Future System Changes

The possibility of future system changes should be considered in the engineering system requirements. The future system changes to the system should be predicted and the system should be designed to enable the implementation of the changes with minimum disturbance to operation.

The future requirements should be presented based on projection of available data. Then, the possibility of designing the equipment to allow for the changes should be evaluated (e.g., providing extra flow or head margins, or specifying an impeller smaller than the maximum for a given casing size). The possibility of modifying the whole system, including the pumps, when the changes are required should also be evaluated.

38.7 Selection of Pump and Driver

The following factors affect the selection of the pump class and type for a specific application system requirements, system layout, fluid characteristics, intended life, energy cost, code, requirements, and materials of construction.

The pump requirements are

1. Providing a given capacity for specific length of time
2. Providing the head or pressure imposed by the system

The following questions should be answered before selecting a pump:

1. What are the required heads and capacities when the load changes?
2. Does the required head change when the capacity changes?

38.7.1 Pump Characteristics

For applications requiring constant capacity when the head changes, constant-speed reciprocating pumps are suitable. These pumps are available in a wide range of design pressures. However, their capacity is relatively small.

Since reciprocating pumps generates pulsations in the flow, rotary pumps are used in applications where pulsations cannot be tolerated. However, rotary pumps are limited to low to medium pressure ranges. In variable head and capacity applications, centrifugal pumps are often used. In low to medium capacity and medium to high pressure applications, straight centrifugal pumps are used. Axial-flow pumps are used in low-head, high-flow conditions. In intermediate situations, mixed-flow impellers are used.

For applications requiring self-priming pumps, reciprocating and rotary pumps can be used. However, centrifugal pumps can also be used if they are specifically designed to perform this function.

In some situations, the system layout will have an impact on the type of pump selected. In general, vertical pumps require less floor space than horizontal pumps. However, more head room is required for the maintenance and installation of these pumps. Also, centrifugal pumps require less floor space than reciprocating pumps.

38.7.2 Code Requirements

The American Petroleum Institute (API) Standard 610 or the Standards of Hydraulic Institute provide the minimum specifications for the design, construction, ratings, and testing of most industrial pumps. However, regulatory agencies' codes may impose additional requirements on the rating and construction of a pump. For example, the ASME Boiler and Pressure Vessel Code requires that the boiler feed pumps be capable of feeding the boilers when the highest set of safety relief valves are discharging. In nuclear power plants, the Nuclear Regulatory Commission (NRC) has imposed more stringent requirements on the quality assurance of critical pump components than specified in normal industrial codes.

38.7.3 Fluid Characteristics

The viscosity, density, volatility, chemical stability, and solids content should be considered when selecting pumps. Some applications may rule out some types of pumps at once. For example, for fluids having high content of solids, reciprocating piston pumps or pumps with close clearances cannot be used. For viscous fluids, such as oil or grease, rotary pumps are suitable. Centrifugal pumps can be used for clear fluids and fluids

having high content of solids. Diaphragm pumps are used for applications where the process liquid cannot come in contact with the moving parts.

38.7.4 Pump Materials

The process fluid and the environment affect the materials. The most important properties are resistance to corrosion and wear. The material selected should be the most suitable and economical for the application. The desirability of specifying expensive long-life materials is weighed against cheaper materials that must be frequently replaced.

Operating factors, such as type of service (continuous or intermittent, critical or noncritical, running speed, and intended life), will also have an effect on the material selection. For example, for continuous and/or critical service applications heavier duty design and construction equipment are used. High-speed applications will normally allow the use of smaller, less expensive equipment.

38.7.5 Driver Type

The main driver type for the pumping equipment are electric motors, steam turbines, steam engines, gas turbines, and internal combustion engines. Variable-speed drives are used to allow centrifugal pumps to operate along the system-characteristic curve to save power during part-load operations.

Electric motors are normally used in constant-speed applications unless hydraulic coupling or variable-speed drives are used. Internal combustion engines are normally used in applications where there is no electric power or when portability or redundancy (loss of power backup) is required.

In large, complex applications requiring continuous operation, the requirement of variable-speed drives should be based on economics. The operating costs of pumping equipment having variable-speed drives should be based on economics. The operating costs of pumping equipment having variable-speed drives are usually lower. However, the capital cost of variable-speed drives should be taken into consideration. A comparison of the total operating and capital costs of the pumping equipment over the intended plant life for both alternatives (including variable-speed or not) should be done. The operating cost should include the energy costs and maintenance and replacement costs. The alternative providing the lowest cost over the life of the plant should be selected.

38.8 Pump Specifications

38.8.1 Specification Types

The first decision to be made when selecting a pump is whether a formal specification or some abbreviated form of requirements will be suitable. A specification is not normally used for relatively simple or inexpensive pumps, or for replacement pumps where duplication is desired. The time and cost required to write a specification for inexpensive pumps or for replacement pumps and obtain and evaluate competitive bids exceed the potential cost savings. In these situations and where the pump supplier is already known, a direct quotation is normally requested from the supplier. The principal requirements should be well defined when requesting this quotation so that they can be included in the technical and priced offering. Tables 38.1 and 38.2 illustrate a pump data sheet. It is helpful for establishing the requirements.

Where a formal specification is required, there are two alternatives: a performance-type specification establishing the performance required by the pump or a construction specification establishing the type of design, construction, and methods that must be used in designing the pump. A performance-type specification establishes “what” is required, while a construction-type establishes “how” the pump must be designed. In most cases, a performance-type specification is used.

38.8.2 Data Sheet

The data sheets have been illustrated below.

LINE NO.	Service	Req'n. No.	Item No.
	Plant	Spec. No.	No. Req'd.
	Location	P. O. No.	Cost Code
	Draw'g. Ref.	(Layout)	(Process)
1	Liquid Pumped		
2	Viscosity (SSU) / Vapor Pressure (PSIA)		
3	Temperature (°F): Max. / Min. // Specific Gravity @ °F.		
4	Flow: Rating / Min. / Max. (GPM) (Lbs./Hr.)		
5	Suction Pressure @ (Flange) (Water Level) (PSIA)		
6	Discharge Pressure @ Flange (PSIA)		
7	Diff. Press.: Rating / Shutoff (Feet) (PSI)		
8	NPSH or Submergence: Available / Req'd @ Elev. (Ft.)		
9	Type of Pump / Model / No. Stgs.		
10	RPM / Rotation (View from Motor Facing Pump)		
11	Efficiency / BHP at Rating / BHP Max. @ GPM		
12	Impeller Diameter: Bid / Max. / Min.		
13	Impeller: Eye Area / Periph. Vel.		
14	Working Press. Max./Hydrotest PSIG		
15	Clearance: Wear Ring / Bearing / Impeller		
16	Hydraulic Thrust: Rating / Max. / Up		
17	WK ² / Speed-Torque: Rating / Speed-Torque Max. (Lb.Ft. ²)/(Lb.Ft.)		
18	Suction: Size / Rating / Facing / Position		
19	Discharge: Size / Rating / Facing / Position		
20	Base Plate / Sole Plate		
21	Coupling: Type / Mfr. / Furn. By		
22	Suction: Strainer Mtl. / Splitter Mtl.		
23	Bearing Lube: Type / GPM / Pressure / Max. Micron		
24	Shaft Seal: Type / Sealing Conn. / Cooling (GPM)		
25	Coupling Guard Type (Horiz. only)		
26	Material: Case or Bowl		
27	(& Size) Barrel		
28	Shaft: Case or Bowl / (Dia.)		
29	Shaft: Sleeve: Brg. / Stuff. Box		
30	Wear Ring: Case / Impeller		
31	Impellers / Liners		
32	VERT. ONLY	Disch. Head / Column (Dia. x Thickness)	
33		Shaft Encl. Tube / (Dia. x Thickness)	
34		Lineshaft / (Dia.) / (Brg. Spacing)	
35		Sleeve Bearing: Bottom / Bowl / Lineshaft	
36	Driver: Type (Motor-Turbine) (Solid-Hollow Sh.) / RPM / HP		
37	Furn. By / Weight / Dwg. Ref. / Mfr.		
38	Bearing Description / Thrust Rating		
39	Lubrication: Thrust / Radial / Cooling		
40	Drawing No.: Outline / Sectional / Performance Curve		
41	Net Weight: Pump / Removable / Rotating Elem.		
42	Inspection: Std. / Nuc. Class I, II or III / ASME III or VIII		
43	Testing: Ultrasonic / Eddy Cur. / Mag. Part. / Liq. Pen. / Radio.		
44	Hydrostatic / Witness // NPSH / Witness		
45	Performance / Witness / Field		
46	Quality Assurance: Mfr. Std. / Documented		
47	Seismic Design Req's: Class I / Class II		
FILL IN ALL BLANKS: IF NOT APPLICABLE MARK "NA"			
NOTES			
NO.	DATE	REVISIONS	JOB NO.
ORIGIN		CENTRIFUGAL PUMP DATA SHEET	DATA SHEET NUMBER
			REV.

TABLE 38.1 Centrifugal Pump Data Sheet

2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40 41 42 43 44 45 46 47	Service	Req'n. No.	Item No.
	Plant	Spec. No.	No. Req'd.
	Location	P.O. No.	Cost Code
	Draw'g. Ref.	(Layout)	(Process)
1	Liquid Pumped / Newtonian or Non-N.		
2	Viscosity (SSU) (CS): Norm. Temp. / Min. Temp. / Max. Temp.		
3	Vapor Press. (PSIA): Norm. Temp. / Min. Temp. / Max. Temp.		
4	Temperature (°F): Norm. / Min. / Max.		
5	Specific Gravity / Pour Point / Flash Point @ _____ °F		
6	Solids: Type / % by vol. / Abrasive / Shape / Distr. / Density		
7	Gases: Entr. / Dis'd. / Solubility (% by Vol. at 30" Hg. abs.)		
8	Flow: Rating / Min. / Max. (GPM)		
9	Pressure: Outlet / Inlet / Diff. Max ΔP (PSIG)		
10	NPSH: Available / Req'd. @ ½ Suction (PSIG)		
11	Type of Pump		
12	RPM / Rotation Facing Coupling		
13	Efficiency / BHP Rating / BHP Max		
14	Relief Valve: Internal - Setting / External - Setting (PSIG)		
15	Suction Conn.: Size / Rating / Facing / Position		
16	Discharge Conn.: Size / Rating / Facing / Position		
17	Mfr. Brg. No.: Radial / Thrust		
18	Base Plate: Pump (& Gear) Only / Comb. W. Driver (& Gear)		
19	Coupling: Mfr. / Model Size / Guard / Dr. Half Mtd.		
20	Seal: Packing Size, No., Type / Mech. Seal Mfr., No.		
21	Lubrication: Main, Type / Secondary, Type / Max. Micron		
22	Cooling: Water GPM., Pr. / Quench / Steam Jacket		
23	Material: Case		
24	Rotors		
25	Shaft / Shaft Sleeve		
26	Throat Bushing		
27	Lantern Ring / Gland		
28	Relief Valve		
29	Base Plate		
30	Cooling Piping / Steam Piping		
31	Seal Piping / Lubrication Piping		
32	Driver: Type (Motor - Turbine - Gear) / RPM / HP		
33	Furn. By / Weight / Dwg. / Mfr.		
34	Bearing Description: Radial, No. / Thrust, No.		
35	Lubrication: Radial / Thrust / Cooling		
36	High Speed Cplg.: Mfr. / Model / Furn. By		
37	Drawings: Outline / Sectional / Perf. Curve		
38	Net Weight		
39	Inspection: Std. / QA		
40	Testing: Ultrasonic / Eddy Cur. / Mag. Part. / Liq. Pen. / Radiogr.		
41	Hydrostatic / Witness // NPSH / Witness		
42	Performance / Witness / Disassembly		
43	Quality Assurance: Mfr. Std. / Documented		
44			
45			
46			
47			
FILL IN ALL BLANKS: IF NOT APPLICABLE MARK "NA"			
NO.		DATE	REVISIONS
ORIGIN		ROTARY PUMP DATA SHEET	
		JOB NO.	
		DATA SHEET NUMBER	
		REV.	

TABLE 38.2 Rotary Pump Data Sheet

38.8.3 Codes and Standards

The codes and standards used when specifying a pump are of great importance. Standards that determines the quality of materials should be included. Other industrial standards, for example ANSI, which establishes the metallurgy, dimensions, tolerances, and flange facing and drilling should also be included. If the pump application has some critical service requirements, other industrial codes should apply to the design and construction. For example, for nuclear service applications, the ASME Boiler and Pressure Vessel

Code, Section III establishes specific requirements for the materials, design features, and quality of manufacturing control of pressure-retaining components such as gland rings.

38.8.4 Bidding Documents

The bidding documents consist of a technical specification and commercial terms. The technical specification determines the performance requirements, materials of construction, and major technical features. The commercial terms include the location of the work, requirements for guarantees/warranties, shipping method, time of delivery, method of payment, normal inspection, and expediting requirements.

38.8.5 Technical Specification

A typical specification should include the following:

1. *Scope of work.* Pump, baseplate, driver (if included), interconnecting piping, lubricating oil pump and piping, spare part, instrumentation (pump-mounted), erection supervision
2. *Work not included.* Foundations, installation labor, anchor bolts, external piping, external wiring, motor starter
3. *Rating and service conditions.* Fluid pumped, chemical composition, temperature, flow, head, speed range preference, load conditions, overpressure, runout, off-standard operating requirements, transients
4. *Design and construction.* (Care should be taken to provide latitude in this section, as this borders on dictating construction requirements.) Codes, standards, materials, type of casing, stage arrangement, balancing, nozzle orientation, special requirements for nozzle forces and moments (if known), weld-end standards, supports, vents and drains, bearing type, shaft seals, baseplates, interconnecting piping, resistance temperature detectors, instruments, insulation, appearance jacket
5. *Lubricating oil system (if applicable).* System type, components, piping, mode of operation, interlocks, instrumentation
6. *Driver.* Motor voltage standards, power supply and regulation, local panel requirements, wiring standards, terminal boxes, electric devices; for internal combustion drivers, fuel type preferred (or required), number of cylinders, cooling system, speed governing, self-starting or manual, couplings or clutches, exhaust muffler
7. *Cleaning.* Cleaning, painting, preparation for shipment, allowable primers and finish coats, flange and nozzle protection, integral piping protection, storage requirements
8. *Performance testing.* Satisfactory for the service, smooth-running, free of cavitation and vibration, shop tests (Hydraulic Institute standards) for pump and spare rotating elements, overspeed tests, hydrostatic tests, test curves, field testing
9. *Drawing and data.* Drawings and data to be furnished, outline, speed versus torque curves, WK^2 data, instruction manuals, completed data sheets, recommended spare parts

10. *Tools.* One set of any special tools, including wheeled carriage for rotor if needed for servicing and maintenance.
11. *Evaluation basis.* Power, efficiency, proven design

The following additional technical specifications may also be required:

1. Electric motor or other type of driver
2. Shipment
3. Painting
4. Quality control testing (important for nuclear applications)

Unusual requirements should also be listed in the technical specification, for example, a sketch of the intake arrangement for wet-pit applications, special requirements regarding unique testing, such as metallurgical testing, etc. The system-head curves, sketches of the piping system, and listings of piping and accessories should also be included. Pump data sheets (Tables 38.1 and 38.2) are useful in providing a summary of information to the bidder. A section of these sheets is filled by the purchaser and the balance by the bidder to provide a summary of pump characteristics, materials to be furnished, accessories, weight, etc. These sheets should be included with the technical specification.

38.8.6 Commercial Terms

The commercial terms included in the bidding documents should provide this information:

- General: Name of buyer, place to which proposals are to be sent, information on ownership of documents, time allowed to bid, governing laws and regulations.
- Location or plant site: This establishes the geographic area in which the equipment is to perform and in a broad way the scope of the work. It should also state maximum temperatures, humidity, storage provisions (indoor or outdoor), and altitude (so that the motor drivers can be selected for the proper cooling).
- Definitions establishing buyer, agent, engineer, seller, whether "or equal" is intended to be exclusive or not.
- Proposal, which establishes the format of the proposal, number of copies, owner's right to accept or reject any bids, status of alternates.
- Schedule, including requirements for all drawings and design data submittals, manufacturing schedule, and equipment delivery.
- Acceptable terms of payment, retentions, etc.
- Transportation to and from point of use (or installation) is frequently a consideration, since with very large equipment, it may not be possible to ship by truck and it may be necessary to either barge- or rail-ship the material. In addition, it is important to establish the method of shipment which may be used so that the bidder can include the proper allowance for freight and to establish responsibility for the risk of loss. Thus, a bid could either include a freight allowance, that is, be FOB manufacturer's plant WFA (with freight allowed)

to point of use or be FOB point of use, in which case freight is included. In either case, the risk of loss remaining with the seller and that assumed by the purchaser should be clearly stated.

38.9 Special Considerations

38.9.1 Performance Testing

For routinely manufactured commercial pumps smaller than 6 in (ISO mm), a sample-selection testing on quality control basis is done. The standardized curves of pump performance are subsequently generated. For these pumps, certified tests are not required unless the application is a critical service, such as fire protection or boiler feed.

A certified performance test is required for larger pumps or critical service applications. The head curve is established by the manufacturer while testing the pump at several points on its performance curve. Power curve must be provided with the head curve to ensure that the pump driver is properly sized. Occasionally, the manufacturer cannot test extremely large pumps, for example, condenser cooling water service. The characteristics of these pumps are established by testing a smaller, geometrically similar model and extrapolating the performance tests.

38.9.2 Pump Drivers

Pump drivers such as motors, turbines, engines, etc. can be purchased with the pumps or from a separate supplier. In general, when the size of the driver exceeds 50 hp (40 kW), it is more economical to purchase the driver separately. For smaller pumps, there is no cost advantage for buying the driver separately.

When the driver is being purchased separately, the specification should ask the supplier to determine the characteristics of the driver. This includes determining the speed of the motor, sizing the driver for both accelerating and operating loads, ensuring thrust-bearing loadings (including direction), and selecting and fitting the couplings. It is also important to specify the enclosure type of the driver, insulation standards, and special features required, such as heaters and oversize junction boxes.

When specifying reciprocating pumps or drivers, the bidders should determine the magnitude and direction of the shaking forces, as well as weight and size, to evaluate the foundation requirements. NEMA (National Electric Manufacturers Association) standards provide a 15% service factor (power overload allowance) for certain type of motors based on internal motor temperatures. This margin should be conserved for short-term overloads. For motors larger than 200 hp (150 kW), where heat dissipation is significant, the motor nameplate should specify starting frequency limitations. For motors larger than 1000 hp (750 kW), the purchaser should specify the starting frequency requirements. These motors are typically limited to 3 to 5 starts per hour. The motor manufacturers should be consulted for applications requiring repetitive starting.

Synchronous motors are normally used for large, low-speed pumps. These motors require more complex control equipment but can be adjusted to the specific starting, pull-in (to synchronism), and pull-out (of synchronism) torques of the load. The operating speed of these motors is identical to the synchronous speed, that is, no slip (difference between synchronous and operating speed). In addition, these motors can operate at unit power factor, resulting in reduction in energy cost (most utilities charge more for motors operating at low power factor, e.g., induction motors).

Synchronous motors can also operate at leading power factors (current leading voltage) to improve the average power factor of the other equipment at the plant.

38.9.3 Special Control Requirements

This is an important consideration to ensure that pumps are not damaged by misapplication. Radial and mixed-flow pumps do not require more power at shutoff. They can be started with their discharge valve closed without overloading their drivers. The power requirement of propeller pumps is high. Their shutoff pressure is very high (relative to design). During start-up, their valves are timed to be sufficiently open as the driver comes up to full speed.

Considerations must also be given to pump shutdown and closing of discharge valves. For piping larger than 24 in (600 mm), motor-operated butterfly valves are normally used in the pump discharge instead of check valves. These butterfly valves isolate the pump and, by proper control, can prevent reverse flow. The pump starting requirements should be analyzed carefully, including the motor accelerating capability and the characteristics of the butterfly valve (flow vs. opening).

38.9.4 Drawing and Data Requirements Form

This form indicates the type of drawings and data required for preliminary design purposes and final information (Table 38.3).

38.9.5 Quality Assurance and Quality Control

Quality assurance is the term used to describe the overall programs and principles that are used to control quality. *Quality assurance* is the term used to describe the methods, procedures, and practices that control activities. The principles of quality assurance include

1. Sufficient independence and authority for the QA/QC organization
2. Establishment of procedures to control activities affecting quality
3. Provision of documented evidence of the satisfactory performance of these activities
4. Loop-closing systems to identify, correct, and report on corrective actions taken on nonconforming items

Quality control principles include

1. Control of purchased items
2. Control of manufacturing processes
3. Gage and tool control
4. Test control
5. Control of storage and shipping activities

Reliability requirements are becoming stricter due to the increasing economic cost of equipment failures and outages. Most companies and purchasers are applying formal QA/QC programs. Statistical methods are being used frequently to control quality. This is especially important in repetitive manufacturing. It can economically identify quality trends and allow corrective action before rejecting production results.

																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																					</

TABLE 38.3 Drawings and Data Requirements	
1	General notes
2	General conditions
3	Technical specifications
4	Drawings
5	Bill of materials
6	Quantity takeoff
7	Cost estimate
8	Project schedule
9	Quality control plan
10	Safety plan
11	Environmental impact statement
12	Permitting requirements
13	Insurance requirements
14	bonding requirements
15	Payment schedule
16	Change order process
17	Dispute resolution process
18	Project closeout requirements
19	Warranty requirements
20	As-built drawings
21	Final report

38.10 Bidding and Negotiation

38.10.1 Public and Private Sector

Public corporations are required by law to take competitive bids from a broad range of bidders. Private firms do not have the same legal requirements. However, most of them have a policy to take bids for all substantial purchases. A specification is required

where bids are taken. It ensures that major considerations and characteristics are not overlooked.

38.10.2 Bid List

The development of a bid list is important for providing a checklist of suppliers who can provide the pumps and the quality required on schedule as well as support service during design of the installation and over the life of the pump. It is convenient to limit the number of bidders to five to provide effective price competition without making the selection burdensome.

38.10.3 Evaluation of Bids

The evaluation should consider cost, pump performance, guarantees, economic advantage of alternatives, delivery, maintenance, installation service, etc.

38.10.4 Cost

This is a major consideration in evaluating bids. In many cases where offerings are almost identical, the only real difference may be cost. However, the technical factors should be considered in detail if small differences in performance can yield high capitalized values, over the life of the pump.

38.10.5 Efficiency

The different efficiencies quoted should be evaluated. This is normally done at the warranted point (usually full load). An increasing penalty should be assigned against pumps having decreasing efficiency. The effect of efficiency is given by a direct charge for the difference in pump power required:

$$\text{Efficiency effect} = (\text{difference in required power}) \times \text{power cost} \\ \times (\text{expected lifetime of pump})$$

38.10.6 Economic Life

The economic life of the pump is a consideration. However, it is very difficult to quantify. In general, the speed provides an indication of the economic life of the pump. For example, if a pump operates at 1800 rpm while an alternate pump operates at 3600 rpm, assuming other details are equivalent, it is likely that the first pump will have a longer economic life due to its lower speed. The first pump is also less likely to have premature failures.

38.10.7 Spare Parts

The cost of spare parts should be evaluated. The specification should request a price list of these spare parts.

38.10.8 Guarantee/Warranty

The type of guarantee/warranty is an important consideration. It is recommended to obtain legal advice for establishing an acceptable clause for this subject that would apply to the purchase of the equipment. The clause should be included in the bidding documents with the commercial terms of purchase. A typical clause provides assurance against defects in material and workmanship. It normally states that the supplier will

38.10.9 Sample Bid Evaluation

Table 38.4a to 38.4d shows a sample bid evaluation.

[illegible]

TABLE 38.4a Condensate Pump Bid Evaluallon (Sheet 1 of 4)

		MTM	PROSPER	FLJ	SPEC
PERFORMANCE					
RATED FLOW	GPM	✓	✓	✓	6900 GPM
EFFICIENCY @ 6900 GPM		76.5%	83.0%	73.0%	-
DESIGN FLOW	GPM	✓	✓	✓	8400 GPM
DESIGN TEMP.	°F	✓	✓	✓	112 °F
SPEC. GRAVITY @ 112 °F		0.991	✓	0.991	-
DIFF. HEAD @ 8400 GPM		✓	✓	✓	1080' TDH
PRESS. @ MIN. FLOW	PSIA	710	670 ②	610	675 PSIA
MIN. CONT. FLOW	GPM	800	1700	800	-
FLOW @ RUN CUT	GPM	10000	9400	11500	9400 GPM
NPSH REQ. @ 9400 GPM		15'	21.4'	15'	-
NPSH REQ. @ DESIGN (8400 GPM)		14.5'	17.6'	14.5'	-
BHP @ 6900 GPM	HP	2700	2600 ②	2650	-
BHP @ 9400 GPM	HP	2900	2880	3100	-
SPEED	RPM	1180	1180	1180	900/1200
1 ST CRITICAL SPEED	RPM	N.S.	>9500	N.S.	-
Nº OF STAGES		7	9	6	-
WEIGHT ROT. PART	LBS	N.S.	3700	3100	-
REVERSE FLOW RPM @ 100% HEAD		<125%	1250 RPM	<125%	-
DESIGN					
SUCTION: SIZE / FLANGE RATING		24"/150*	30"/150*	24"/150*	24" PREF.
DISCHARGE: SIZE / FLANGE RATING		18"/300*	18"/300*	18"/300*	16" PREF.
DIAM (O.D.) OF SHELL		42"	44"	42"	-
LENGTH OF SHELL		18'-6" ①	22'-5 1/2"	20'-8"	-
WEIGHT OF PUMP (EACH)	LBS	43,000	46,500	40,000	-
WEIGHT OF MOTOR	LBS	26,000	23,000	26,000	-
SUCTION: DESIGN PRESS.	PSI	15x MAX. SUCT.	50	15x MAX. SUCT.	-
DISCHARGE: DESIGN PRESS.	PSI	15x SHUT-OFF	650	15x SHUT-OFF	-
SUCTION: TEST PRESS.	PSI	15x MAX. SUCT.	75	15x MAX. SUCT.	-
DISCHARGE: TEST PRESS.	PSI	15x SHUT-OFF	975	15x SHUT-OFF	15x MAX. SHUT-OFF
① AS PER LETTER, MARCH 9, 1970; INCREASE IN AVAILABLE SUCTION HEAD BY 8 FT. ENABLES MTM TO SHORTEN LENGTH OF BARREL FROM 21'-6" TO 18'-6" ② ADJUSTED TO PROSPER LETTER MARCH 6, 1970 NOTES: ENCIRCLED ITEMS ARE UNDESIRABLE A CHECK ✓ INDICATES COMPLIANCE WITH SPECS					
ISSUED FOR CLIENT APPROVAL		SUMMARY OF BIDS		JED: XXXX	
REV. DESCRIPTION		CONDENSATE PUMPS		P.O. NO.	
		XYZ POWER COMPANY		M-23	
				SHT. 2 OF 4	
				REV.	
				0	

TABLE 38.4b Condensate Pump Bid Evaluation (Sheet 2 of 4)

		DATE	APPROVALS	MATL	SUPPLY	CHK	DR	ENG	ISSUED FOR CLIENT APPROVAL	REV. DESCRIPTION	
		11-13C ST	PROSPER	FLJ	SPEC						
CASE: MAT. OF SUCTION BELL		11-13C ST	C.I. ①	11-13C ST	11-13C ST						
MAT. OF WEAR RINGS		BRZ	C.I.	BRZ	-						
MATERIAL OF CASE		A-48C.I.	A-48C.I.	A-48C.I.	-						
STD. WALL THICKNESS / MIN.		1" 7/8"	1" 13/16"	1" 7/8"	-						
VANES & WATERWAYS		STD ✓	AS CAST	STD ✓	-						
WEAR RINGS SUCT / DISCH.		- / ✓	✓ / -	- / ✓	-						
CLEARANCE		0.02-0.024"	① NS	0.02-0.024"	-						
IMPELLER: MAT. 1 ST STG. / OTHERS		11-13C ST/BRZ	11-13C ST/BRZ	11-13C ST/BRZ	11-13C ST/BRZ						
TYPE 1 ST STG. / OTHERS		ENCL / SEMI-E	ENCL.	ENCL / SEMI-E	-						
DIAM. 1 ST STG. DESIGN / MAX.		19 5/8" / 20 3/8"	20 3/8" / 20 3/8"	19 5/8" / 20 3/8"	-						
OTHER STG. DESIGN / MAX / MIN.		19 5/8" / 20 3/8"	20 3/8" / 18 5/8"	19 5/8" / 20 3/8"	-						
BALANCE - HYDR. / STAT. / DYN.		✓ / - / ✓	- / ✓ / ✓	✓ / - / ✓	REQ'D.						
WEAR RINGS SUCT. / DISCH.		- / ✓	✓ / -	- / ✓	-						
WR ² LB FT ²		1250	650	1250	-						
HYDR. THRUST MAX. DOWN		40,000	74700 ②	40,000	-						
HYDR. THRUST MAX. UP		NONE	22400 ②	NONE	-						
VELOCITY - EYE DESIGN / MAX.		15 1/2 / 23	17 / 19 FPS	16 / 21	-						
BEARING: MATERIAL		BRZ	BRZ	BRZ	BRZ						
BOTTOM LUBE / L/D		SELF / 2.4	SELF / 1.5 ③	SELF / 2.4	SELF						
GROOVED / TYPE		✓ / SLEEVE	✓ / JOURN.	✓ / SLEEVE	-						
LUBE / L/D		SELF / 2	SELF / 1.0 ③	SELF / 2	SELF						
SERIES GROOVED / TYPE		✓ / SLEEVE	✓ / JOURN.	✓ / SLEEVE	-						
TOP LUBE / L/D		SELF / 2	SELF / 1.0 ③	SELF / 2	SELF						
GROOVED / TYPE		✓ / SLEEVE	✓ / JOURN.	✓ / SLEEVE	-						
THROTTLE LUBE / L/D / GROOV.		SELF / 2 / ✓	SELF / 1.0 ③ / ✓	SELF / 2 / ✓	SELF						
SHAFT: MAT. PUMP ELEMENT		SS 410	SS 416	SS 410	SS 400 SERIES						
MAT. COLUMN		SS 410	C. ST.	SS 410	-						
DIAM. / LONG BEARING SPAN		5" / 48"	5 1/8" / 45"	5" / 48"	-						
COMBINED STRESS MAX.		10,000	14,500 PSI ^{DRG. ONLY}	10,000	-						
SURFACE FINISH & BEAR'G.		32	32 RMS	32	GRD						
SURFACE @ STUFF. BOX		32	125 RMS	32	GRD						
SLEEVE @ STUFF. BOX / THRU GLAND		✓ / ✓	✓ / ✓	✓ / ✓	REQ'D. / REQ'D.						
PRESS @ STUFF BOX		15 PSI > SUCT.	15-20 PSI > SUCT.	15 PSI > SUCT.	-						
SLEEVE MAT.		SS-420	ALLOY- ST.	SS-420	-						

① AS PER PROSPER LETTER MARCH 4, 1970. (SPEC. REQUESTED 11-13C ST SUCTION BELL & 1 1/8" ST'G. IMPELLER MATERIAL)

② HIGH DOWN THRUST.

③ SPEC. RECOMMENDS A MIN. RATIO OF 2. PROSPER DECLINES TO UPGRADE SUBJECT RATIO.

NOTES: ENCIRCLED ITEMS ARE UNDESIRABLE
A CHECK ✓ INDICATES COMPLIANCE WITH SPECS

SUMMARY OF BIDS CONDENSATE PUMPS		JGD No XXXX	REV.
XYZ POWER COMPANY		P.O. NO. M-23 SHT. 3 OF 4	0

TABLE 38.4c Condensate Pump Bid Evaluallon (Sheet 3 of 4)

[illegible]

TABLE 38.4d Condensate Pump Bid Evaluallon (Sheet 4 of 4)

38.11 Bibliography

Karassik, I. J., Krutzsch, W. C., Fraser, W. H., and Messina, J. P., *Pump Handbook*, 2nd ed. McGraw-Hill, New York, 1986.

This page intentionally left blank

CHAPTER 39

Pumping System Calculations

39.1 Analysis of Pumps Installed in Series

A pump application requires a 45 gal/min (2.84 L/s) at 26 ft (7.9 m) head. Figure 39.1 illustrates the characteristic curves of some pumps. A two 100% pumping arrangement (one operating and one standby pump) is being considered. They each have a rating of 0.75 hp (0.56 kW). An alternative option of pumps installed in series is also being considered. Can energy be saved with this option? If so, how much?

Solution

Point 1 in Fig. 39.1 is the required operating head and flow rate. The alternative option consists of installing two 0.25 hp (0.19 kW) in series. The total head developed by this pumping arrangement will be double the head developed by each pump. Therefore, this arrangement can generate the required head of 26 ft (7.9 m).

Figure 39.2 illustrates the characteristic curves of the proposed arrangement. The *seriesed pump curve* is constructed by doubling the value of the single-pump head at any flow rate. When both pumps are operating, the system will operate at the intersection of the system-head curve and the seriesed pump characteristic curve (Fig. 39.2). When a single pump is operating, the system will operate at the intersection of the system-head curve and the single-pump characteristic curve.

Energy can be saved using a seriesed pump arrangement. The amount of saving using this option is

$$0.75 \text{ hp} - 2 \times 0.25 \text{ hp} = 0.25 \text{ hp (0.19 kW)}$$

If the plant operates 8000 hour during the year, the energy saving will be

$$8000 \text{ hour} \times 0.19 \text{ kW} = 1520 \text{ kW} \cdot \text{h or } 1.52 \text{ MW} \cdot \text{h}$$

Many applications require variable load. If this application, requires 32.5 gal/min (2.1 L/s) at a head of 14 ft (4.25 m) for a period of time, the power saving will be doubled during this period because only one pump will be required. Therefore, seriesed pump arrangements are recommended for applications having variable load.

39.2 Analysis of Pumps Installed in Parallel

A pump application requires a flow of 80 gal/min (7.4 L/s) of hot water at 13 ft (3.96 m) system head. This requirement can be handled by two 100% pumps (one operating and one standby). Each of them is driven by 0.5 hp (0.37 kW) motor. The system requires 45 hours of constant operation and 50 hours of partial operation of the pump per week. Can energy be saved using a different pumping arrangement?

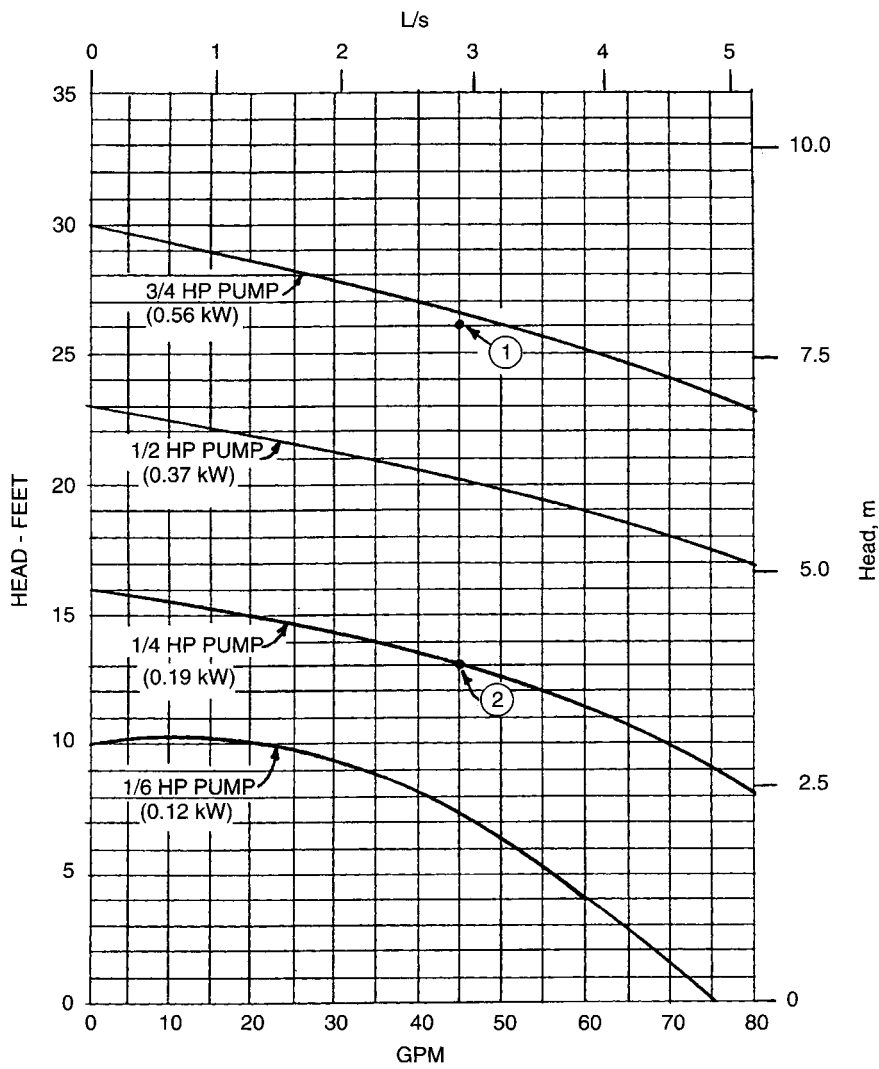


FIGURE 39.1 Pump characteristic curves for use in series installation. (Source: Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006 with permission from McGraw-Hill.)

Solution

Figure 39.3 illustrates the head-flow characteristics of pumps that can be used for this application. Note that this application can be handled by two pumps in a parallel arrangement. Each one will be pumping 40 gal/min (3.7 L/s) at 13-ft (3.96-m) head. Figure 39.4 illustrates the head-flow characteristic curves for a single and dual pumps installed in parallel. The head-flow characteristic curve for the dual pumps installed in parallel is constructed by doubling the flow of the single pump at any given head. Figure 39.5 illustrates the variation of the system-head curve. It should be noted that the system head varies with the square of the flow. If a single pump is operated, it will deliver, as shown in Fig. 39.5, 72 gal/min (6.67 L/s) at 10-ft (3.05-m) head. In many applications requiring variable loads, a single pump operation is adequate to meet the part-load condition.

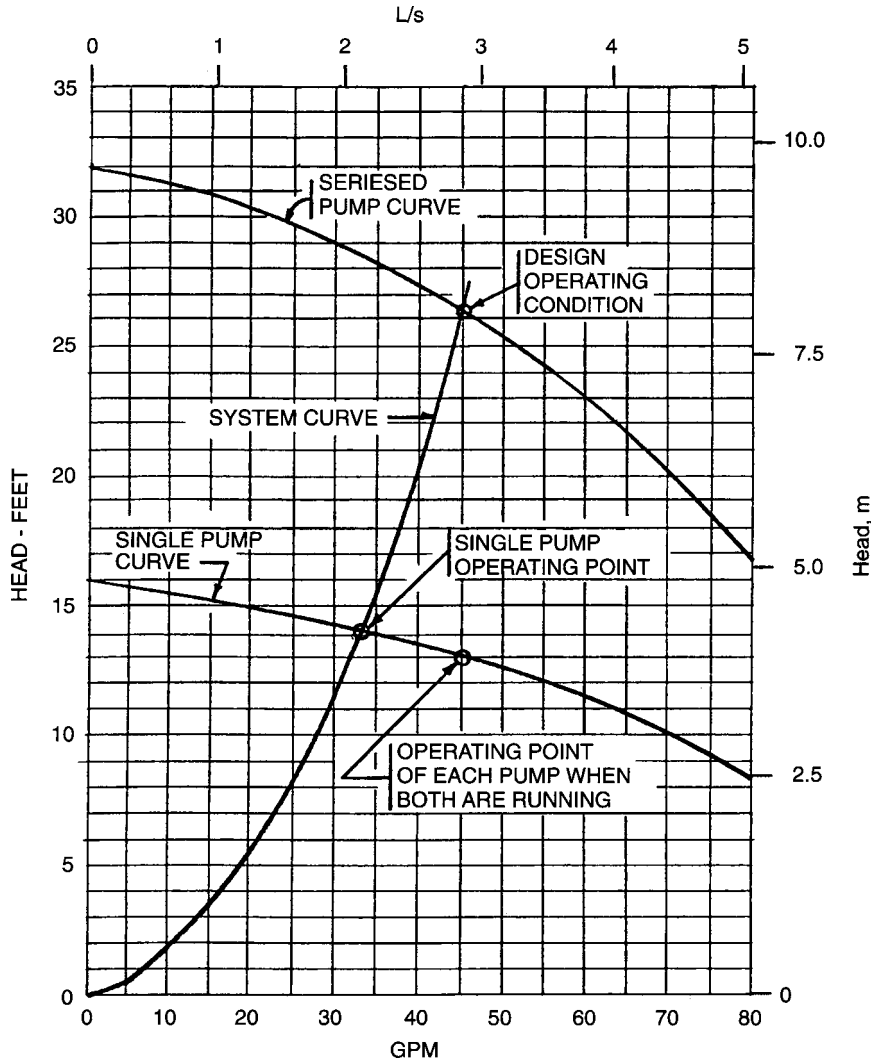


FIGURE 39.2 Seriesed-pump characteristic and system-head curves. (Source: Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006 with permission from McGraw-Hill.)

The characteristic curves of the pump provided by the pump manufacturer specify its power requirement at various loads. Since the cost of two or three smaller pumps is normally less than the cost of a large pump, the parallel pumping arrangement is more desirable than a single large pump for many applications. The economic advantage of the parallel pumping arrangement option becomes significant if the operation of only one of the pumps installed in parallel is adequate to meet the part-load requirements of the system. Check valves should be installed in parallel pumping arrangements to prevent reverse flow. Figure 39.6 illustrates the proper location of check valves in a parallel pumping arrangement.

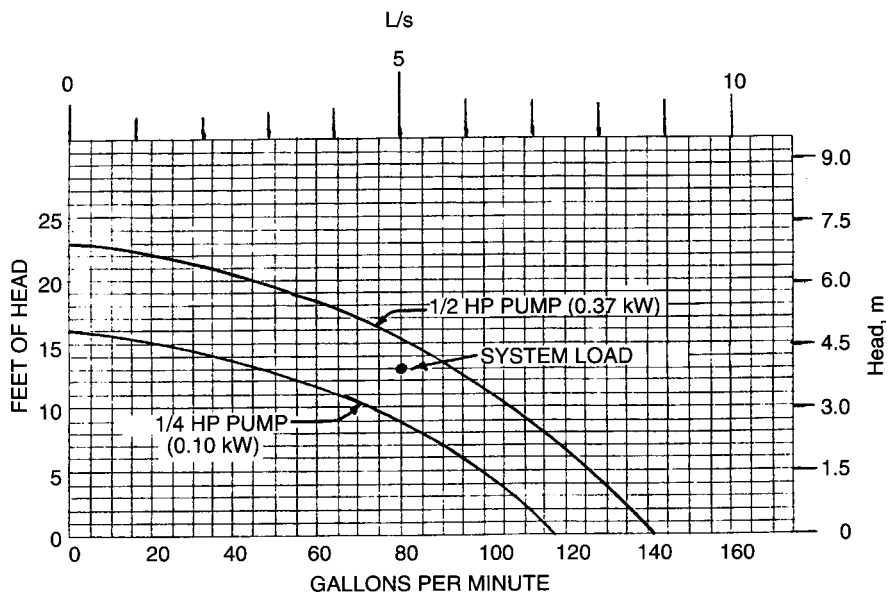


FIGURE 39.3 Typical pump characteristic curves. (Source: Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006 with permission from McGraw-Hill.)

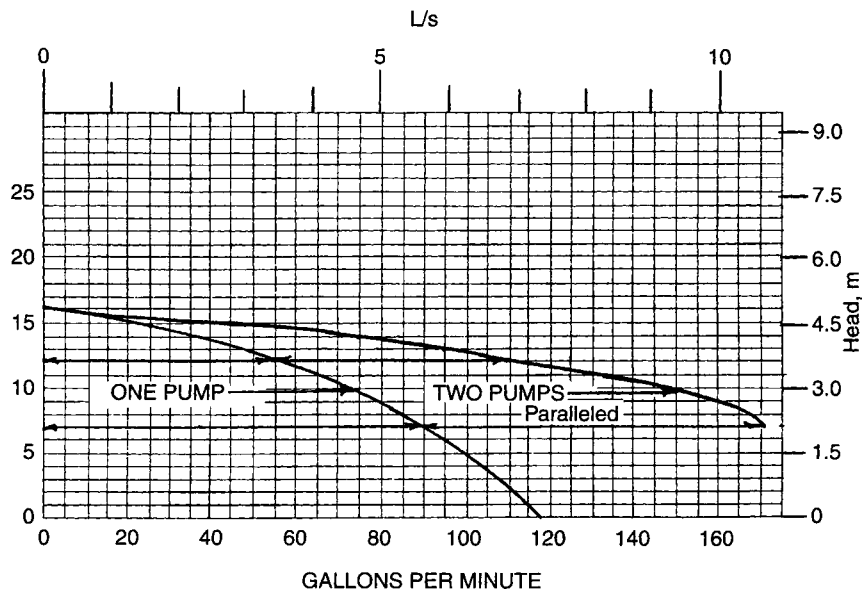


FIGURE 39.4 Single- and dual-parallel pump characteristic curves. (Source: Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006 with permission from McGraw-Hill.)

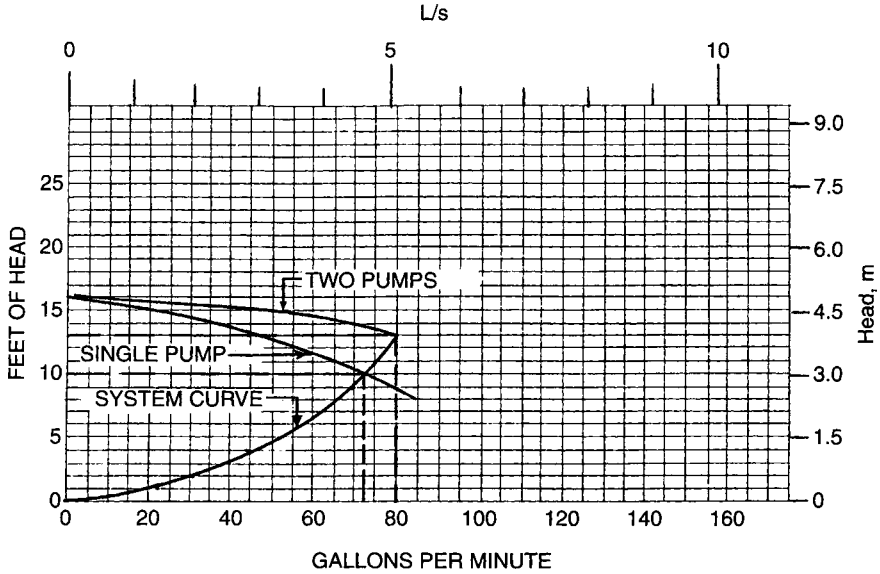


FIGURE 39.5 System-head curve for parallel pumping. (Source: Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006 with permission from McGraw-Hill.)

If one pump can meet the system requirements 85% of the time, the saving in energy is

$$0.85(0.37 \text{ kW} - 0.19 \text{ kW}) \times 95 \text{ hours per week} = 14.54 \text{ kWh/week}$$

If electricity costs 6 cents per kWh, the annual saving is

$$\$0.06 \times 14.54 \text{ kWh/week} \times 52 \text{ week/year} = \$45.35 \text{ per year}$$

Since most industrial pumps have a much higher rating than the ones used in this example, the annual saving in the cost of electricity in these applications will be substantial. The saving will also increase if the operating time of the system is larger.

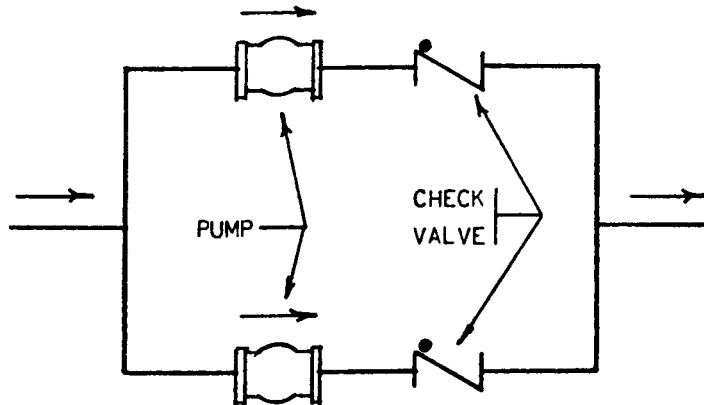


FIGURE 39.6 Check valve locations to prevent reverse flow. (Source: Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006 with permission from McGraw-Hill.)

39.3
Selection of Pump Driver Speed

A single-suction pump is handling 25,000 gal/min (94,750 L/min) at a total head of 80 ft (24.3 m). This pump has a suction lift of 10 ft (3.04 m). Select the rpm of the pump to meet these conditions.

Solution
Figure 39.7 illustrates the Hydraulic Institute specific-speed chart. Using this chart, we find that the specific speed for this application is 2980. From the specific speed equation (Fig. 39.7), we find the following:

$$\text{rpm} = N_s \times H^{0.75} / Q^{0.5}$$

where N_s = specific speed of the pump, rpm
 H = total head of the pump, ft (m)
 Q = pump flow rate, gal/min (L/min)

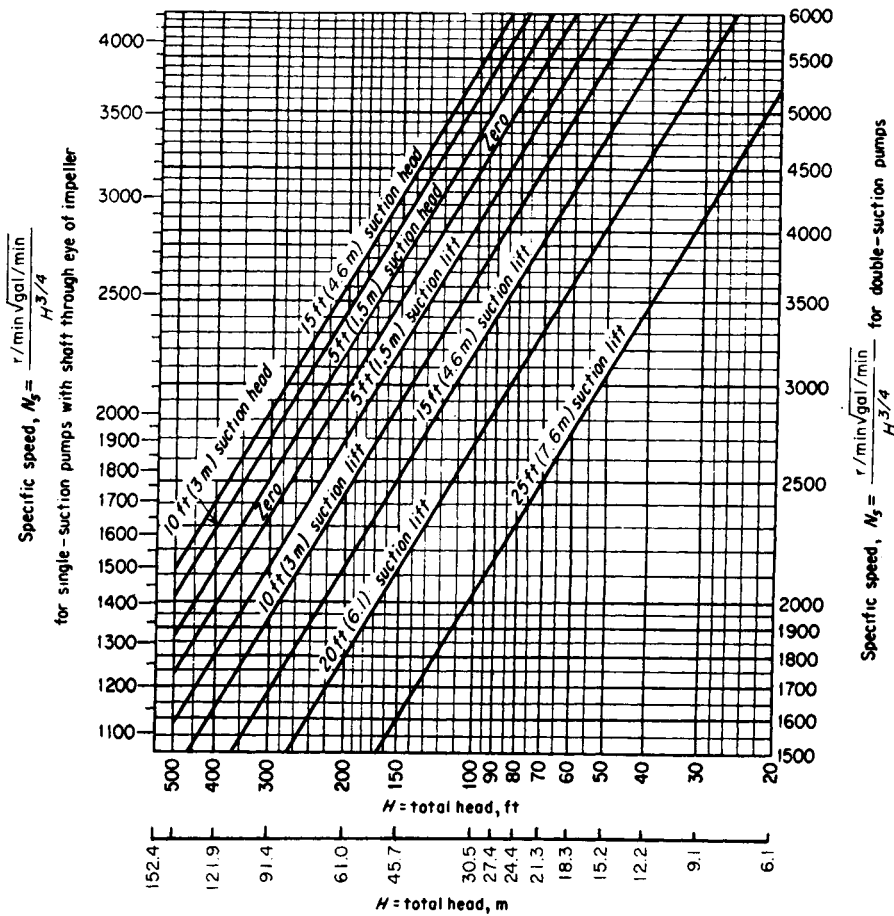


FIGURE 39.7 Upper limits of specific speeds of single-stage, single- and double-suction centrifugal pumps handling clear water at 29.4°C (85° F) at sea level. (Sources: Hydraulic Institute, Parsippany, NJ.)

Thus,

$$\text{rpm} = 2980 \times 80^{0.75} / 25,000^{0.5} = 504.16$$

Hence, a motor having an operational speed around 500 rpm should be selected.

39.4 Affinity Laws for Centrifugal Pumps

A centrifugal pump operates at 3600 rpm, has a flow of 2000 gal/min (126.2 L/s), a head of 100 ft (30.45 m), and a power input of 100 hp (74.6 kW). How would the flow, head and power vary if the operating speed drops to 1800 rpm? How will these variables change if the impeller diameter is reduced from 10 to 8 in (254 to 203.2 mm) while the speed remains constant at 3600 rpm?

Solution

The following are the affinity laws (known also as similarity laws) for a centrifugal pump having a *constant impeller diameter*:

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}; \frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2; \frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3$$

For *constant operating speed*, the affinity laws become

$$\frac{Q_1}{Q_2} = \frac{D_1}{D_2}; \frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2; \frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^3$$

where, in both sets of laws,

Q = flow capacity, gal/min or L/s

N = impeller rpm

D = impeller diameter, in or m

H = total head, ft or m of liquid

P = bhp or kW input

The subscripts 1 and 2 refer to the initial and final condition, respectively.

For this application, when the operational speed drops while the impeller diameter remains unchanged,

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} = \frac{3600}{1800} = 2$$

Thus,

$$Q_2 = \frac{Q_1}{2} = \frac{2000}{2} = 1000 \text{ gal/min}$$

$$Q_2 = 1000 \text{ gal/min (63.1 L/s)}$$

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2 = 2^2 = 4$$

Thus,

$$H_2 = \frac{H_1}{4} = \frac{100}{4} = 25 \text{ ft}$$

$$H_2 = 25 \text{ ft (7.61 m)}$$

$$\frac{P_1}{P_2} = \left(\frac{N_1}{N_2} \right)^3 = 2^3 = 8$$

Thus,

$$P_2 = \frac{P_1}{8} = \frac{100}{8} = 12.5 \text{ hp}$$

$$P_2 = 12.5 \text{ hp (9.33 kW)}$$

These results indicate that in applications requiring variable flow, a variable speed drive should be considered due to the significant reduction of power consumption at low flows.

The second set of laws is used when the impeller diameter is changed while the speed remains constant. For this application,

$$\frac{Q_1}{Q_2} = \frac{D_1}{D_2} = \frac{10}{8} = 1.25$$

Thus,

$$Q_2 = \frac{Q_1}{1.25} = \frac{2000}{1.25} = 1600$$

$$Q_2 = 1600 \text{ gal/min (100.96 L/s)}$$

$$\frac{H_1}{H_2} = \left(\frac{D_1}{D_2} \right)^2 = 1.25^2 = 1.56$$

Thus,

$$H_2 = \frac{H_1}{1.56} = \frac{100}{1.56} = 64$$

$$H_2 = 64 \text{ ft (19.46 m)}$$

$$\frac{P_1}{P_2} = \left(\frac{D_1}{D_2} \right)^3 = 1.25^3 = 1.95$$

Thus,

$$P_2 = \frac{P_1}{1.95} = \frac{100}{1.95} = 51.2$$

$$P_2 = 51.2 \text{ hp (38.2 kW)}$$

The affinity laws are used to change the characteristic curves of a centrifugal pump when the operational speed or impeller diameter is changed.

39.5 Centrifugal Pump Selection Using Similarity or Affinity Laws

A test-model pump operating at its best efficiency point has the following parameters:

Flow = 350 gal/min (22.09 L/s)

Head = 200 ft (60.77 m)

Required net positive suction head (NPSH) = 8 ft (2.43 m)

Power input = 40 hp (29.84 kW)

Operating speed = 3570 rpm

Impeller diameter = 8 in (203.2 mm)

- Determine the performance of the model when the operating speed is 1780 rpm.
- Determine the performance of a full-scale prototype pump having an impeller diameter of 16 in (40.64 cm) when it operates at 1150 rpm.
- Determine the specific speeds and the suction specific speeds of the test-model and prototype pumps.

Solution

Determine the pump performance at the reduced speed

The similarity or affinity laws are

$$Q_p = K_d^3 K_n Q_m$$

$$H_p = K_d^2 K_n^2 H_m$$

$$NPSH_p = K_d^2 K_n^2 NPSH_m$$

$$P_p = K_d^5 K_n^3 P_m$$

where,

$$K_d = \text{size factor} = \frac{\text{prototype dimension}}{\text{model dimension}}$$

The dimension normally used for the size factor is the impeller diameter. The same units should be used for both dimensions.

$$K_n = \text{size factor} = \frac{\text{prototype speed, rpm}}{\text{model speed, rpm}}$$

where Q = flow

H = head of the pump

P = power input to the pump

When the model operating speed drops from 3570 rpm to 1780 rpm while the pump dimensions remain the same ($K_d = 1$),

$$K_n = \frac{1780}{3570} = 0.49$$

Then,

$$Q = 1^3 (0.49)^3 350 = 174.51 \text{ gal/min (11.01 L/s)}$$

$$H = (1.0)^2 (0.49)^2 (200) = 48.02 \text{ ft (14.58 m)}$$

$$NPSH = (1.0)^2 (0.49)^2 (8 \text{ ft}) = 1.92 \text{ ft (0.58 m)}$$

$$P = (1.0)^5 (0.49)^3 (40) = 4.71 \text{ hp (3.51 kW)}$$

The subscripts have been omitted from the equations because the same pump (the test model) is being used.

Determine the performance of the prototype pump when the impeller diameter increases and the operating speed decreases

In this case,

$$K_d = \frac{16}{8} = 2$$

$$K_n = \frac{1150}{3570} = 0.32$$

Then,

$$Q_p = 2^3(0.32)(350) = 901.96 \text{ gal/min (56.93 L/s)}$$

$$H_p = 2^2(0.32)^2(200) = 81.92 \text{ ft (24.9 m)}$$

$$NPSH_p = 2^2(0.32)^2 8 = 3.28 \text{ ft (1.0 m)}$$

$$P_p = 2^5(0.32)^3(40) = 41.94 \text{ hp (31.29 kW)}$$

Calculation of the specific speed and suction specific speed

The specific speed (known also as discharge specific speed) is given by

$$N_s = \frac{N(Q)^{0.5}}{(H)^{0.75}}$$

The suction specific speed is given by

$$S = \frac{N(Q)^{0.5}}{(NPSH)^{0.75}}$$

All the values in these equations are taken at the best efficiency point of the pump. For the test-model pump:

$$N_s = \frac{3570(350)^{0.5}}{200^{0.75}} = 1255.27$$

$$S = \frac{3570(350)^{0.5}}{8^{0.75}} = 14,044.30$$

For the prototype pump:

$$N_s = \frac{1150(901.96)^{0.5}}{(81.92)^{0.75}} = 1268.27$$

$$S = \frac{1150(901.96)^{0.5}}{(3.28)^{0.75}} = 14,170.53$$

These equations can be used for any type of centrifugal pumps where the similarity or affinity laws apply.

39.6 Determination of Centrifugal Pump Capacity and Efficiency

Determine the capacity and efficiency of a single-suction mixed flow centrifugal pump operating at 3560 rpm pumping clear water at 29.4°C (85°F) at sea level if the total head is 200 ft (60.8 m) and 5 ft (1.52 m) suction lift.

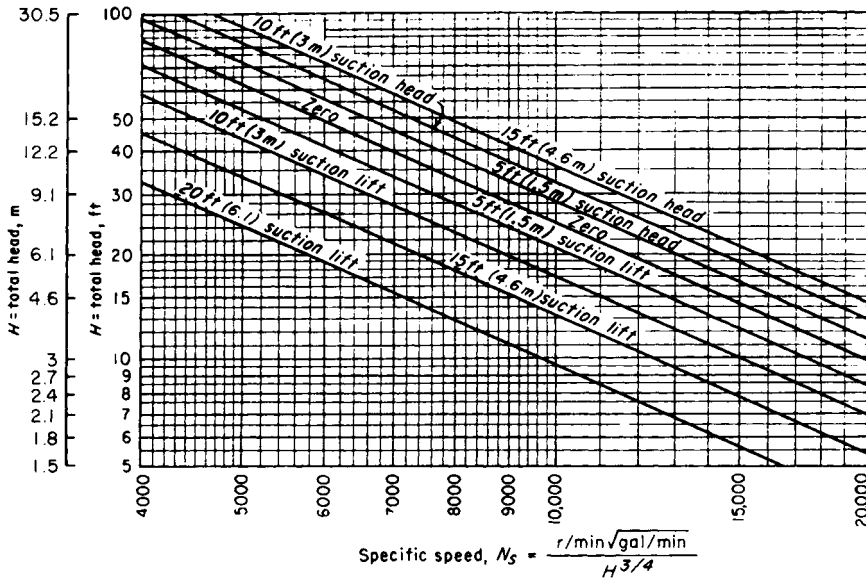


FIGURE 39.8 Upper limits of specific speeds of single-suction mixed-flow and axial-flow pumps. (Source: Hydraulic Institute, Parsippany, NJ.)

Solution

Determine the specific speed of the pump

Figure 39.7 (shown earlier) and Fig. 39.8 illustrate the Hydraulic Institute graphs for the upper limits of specific speed. In this case, the specific speed $N_s = 1905$ (Fig. 39.7)

Calculate the maximum pump capacity

The specific speed of mixed-or axial flow centrifugal pumps is given by

$$N_s = \frac{(\text{flow in gpm})^{0.5} (\text{rpm})}{H_t^{0.75}}$$

where H_t is the total head of the pump in ft of liquid.

From this equation, the flow is given by

$$\begin{aligned} \text{Flow in gpm} &= \left(\frac{N_s H_t^{0.75}}{\text{rpm}} \right)^2 \\ &= \left(\frac{1905 \times 200^{0.75}}{3560} \right)^2 = 809.91 \end{aligned}$$

Flow = 809.91 gpm (51.10 L/s)

Determine the efficiency of the pump

Figure 39.9 illustrates the variation in the shape of the impeller and efficiency with the specific speed. In this case, the efficiency is around 84%. Figure 39.9 shows also the shape of the impeller used in this pump. It is relatively short and has a large discharge area.

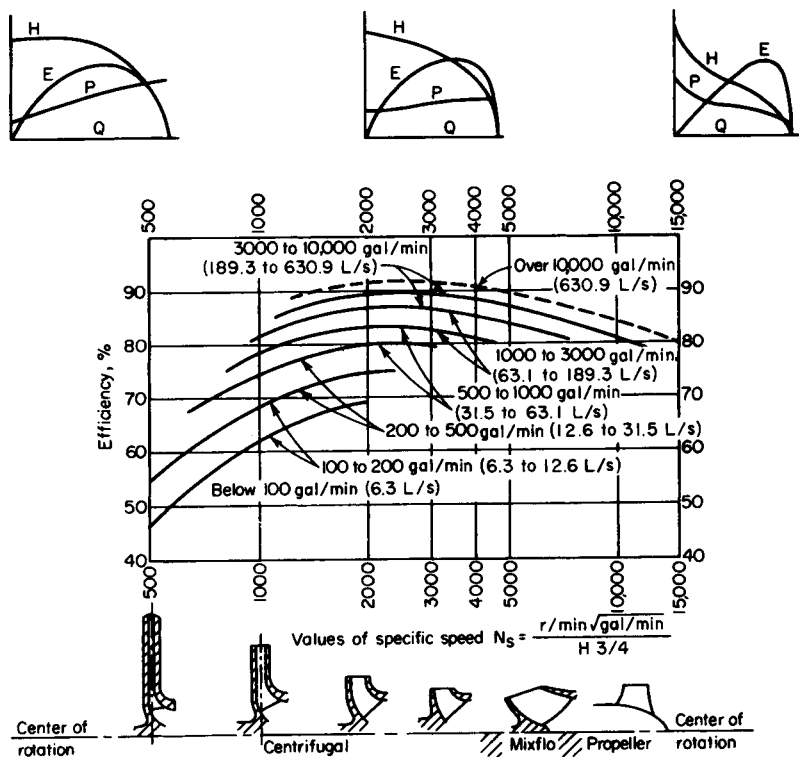


FIGURE 39.9 Approximate relative impeller shapes and efficiency variations for various specific speeds of centrifugal pumps. (Source: Worthington Corporation.)

39.7 Selection of the Best Operating Speed for a Centrifugal Pump

A 60-Hz ac motor drives a single-suction centrifugal pump. The flow through the pump is 8000 gal/min (504.77 L/s) of water at a head of 80 ft (24.4 m). The available net positive suction head is 28 ft (9.76 m) of water. Select the best operating speed for this pump so it can operate at its best efficiency point.

Solution

Calculate the specific speed and suction specific speed

The following are the typical speeds of 60-Hz ac motors depending on the number of poles.

Typical Operating Speed	Number of Poles
880	8
1170	6
1770	4
3560	2

The specific speed is given by

$$N_s = \frac{N(Q)^{0.5}}{(H)^{0.75}} = \frac{N(8000)^{0.5}}{(80)^{0.75}} = 3.34 N$$

The suction specific speed is given by

$$S = \frac{N(Q)^{0.5}}{(NPSH)^{0.75}} = \frac{N(8000)^{0.5}}{(28)^{0.75}} = 7.35 N$$

The following table shows the results at the typical operating speed:

Operating Speed (rpm)	Required Specific Speed	Required Suction Specific Speed
880	2,939.2	6,468.
1170	3,907.8	8,599.5
1770	5,911.8	13,009.5
3560	11,890.4	26,166.0

Select the best operating speed for the pump

Table 39.1 shows the different pump types listed by specific speed and Table 39.2 shows the different ratings of the suction specific speed. Table 39.2 indicates that the suction specific speed rating is poor at an operating speed of 880 or 1170 rpm. However, the suction specific speed is excellent at an operating speed of 1770 rpm. Table 39.1 indicates that a mixed-flow pump will be suitable for this application. The operating speed of 3560 rpm should not be considered because a suction specific speed of 26,166.0 is much higher than the range of conventional pumps.

Specific Speed Range	Type of Pump
Below 2000	Volute, diffuser
2000–5000	Turbine
4000–10,000	Mixed-flow
9000–15,000	Axial-flow

*Peerless Pumps Division, FMC Corporation.

TABLE 39.1 Pump Types Listed by Specific Speed*

Single-Suction Pump	Double-Suction Pump	Rating
Above 11,000	Above 14,000	Excellent
9000–11,000	11,000–14,000	Good
7000–9000	9000–11,000	Average
5000–7000	7000–9000	Poor
Below 5000	Below 7000	Very poor

*Peerless Pumps Division, FMC Corporation.

TABLE 39.2 Suction Specific-Speed Ratings*

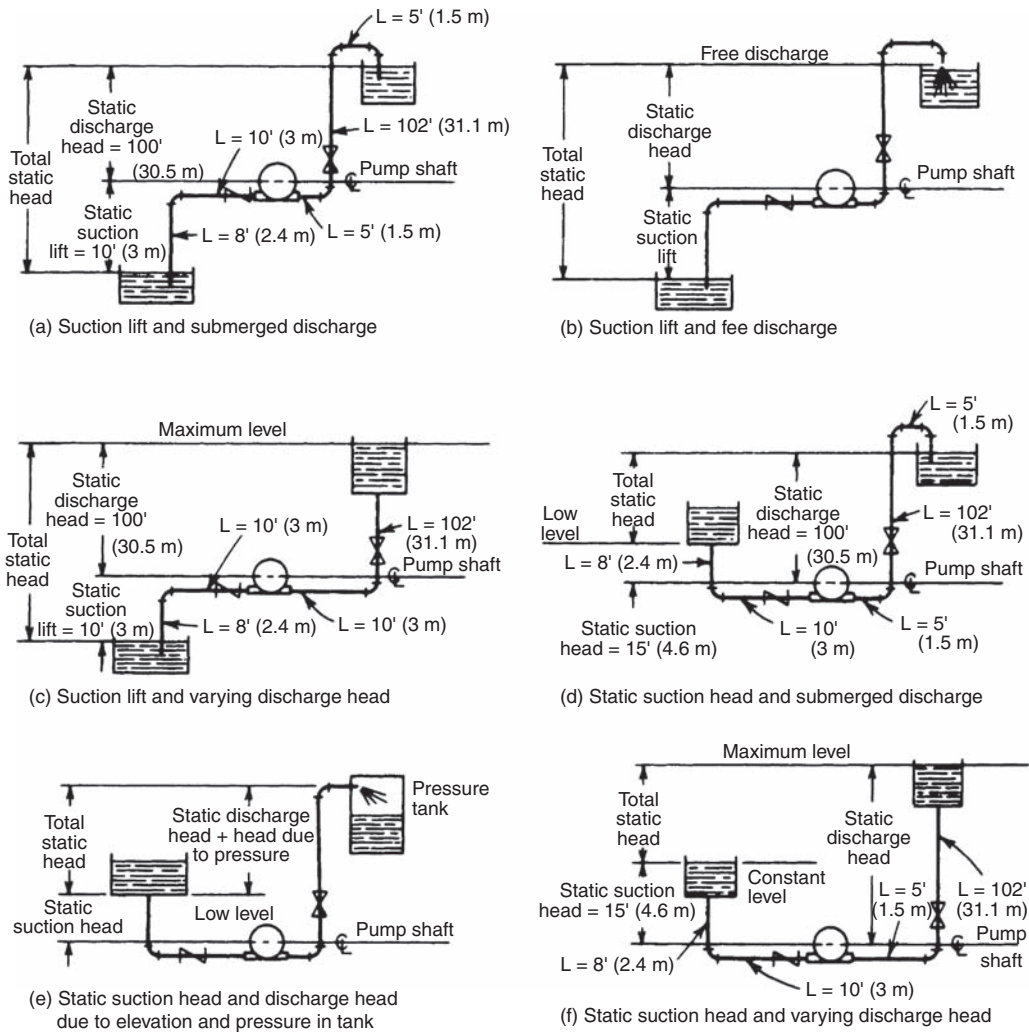


FIGURE 39.10 Typical pump suction and discharge piping arrangements. (Source: Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006 with permission from McGraw-Hill.)

39.8 Calculate the Total Head of the Pump

Figure 39.10 illustrates the typical arrangements for a pump suction and discharge piping. Calculate the total head for each case if the maximum flow through the pump is 1500 gal/min (94.65 L/s) of water through 6 in (15.24 cm) schedule 40 pipe. Calculate the input power required by the pump.

Solution

The total static head is defined as the vertical distance from the surface of the liquid in the supply tank to the free surface of the liquid in the discharge tank, or to the elevation of free discharge point from the discharge pipe. The maximum possible total static head (i.e., the difference between the *lowest* possible level in the supply tank and the *highest* possible level in the discharge tank or pipe should be

used in each case). The *static suction lift* is defined as the vertical distance between the pump centerline and the level of the liquid in the supply tank *below* the pump centerline. If the supply tank is *above* the pump centerline, the vertical distance between the pump centerline and the level of the liquid in the supply tank is called *static suction head*. The *lowest* liquid level in the supply tank should be used when calculating the static suction head.

Calculate the total static head on the pump

In applications where the pump has a suction lift (i.e., Figs. 39.10*a*, *b*, and *c*), the total static head H_{ts} is given by:

$$H_{ts} = \text{static suction lift } (h_{sl}) + \text{static discharge head } (h_{sd})$$

$$H_{ts} = 10 + 100 = 100 \text{ ft (33.5 m)}$$

In these applications, the static discharge head is the distance between the pump centerline and the water level in a tank having an underwater discharge, Fig. 39.10*a*; to the elevation of free discharge point from the pipe outlet, Fig. 39.10*b*; and to the highest water level in the discharge tank, Fig. 39.10*c*. If the liquid in the discharge tank is compressed by a gas blanket, the total static head H_{ts} is given by:

$$H_{ts} = \text{static discharge head} + \text{head equivalent (in ft of liquid) of the internal pressure in the tank}$$

In applications having a pump static suction head (i.e., Fig. 39.10*d*, *e*, and *f*), the total static head is given by:

$$H_{ts} = \text{static discharge head } (h_{sd}) - \text{static suction head } (h_{sh})$$

$$H_{ts} = 100 - 15 = 85 \text{ ft (25.9 m)}$$

The total static heads calculated above refers to the pump head without having any flow through it. The friction losses caused by the liquid flow through the piping system should be calculated to determine the total head of the pump.

Calculate the friction losses in the piping system

In the application shown in Fig. 39.10*a*, the total length of straight pipe is:

$$L_t = 8 + 10 + 5 + 102 + 5 = 130 \text{ ft (39.6 m)}$$

The *equivalent* length of the pipe fittings should be added to the total length of straight pipe. In this application, there are four long-radius elbows, one swing check valve (upstream of the pump), and one globe valve (downstream of the pump). There is also a minor head loss at the pipe inlet and at the pipe outlet. Table 39.3 shows the equivalent length of typical fittings and valves used in a piping system.

The following table shows the equivalent length of the 6 in (15.24 cm) pipe fittings used in the application shown in Fig. 39.10*a*.

Pipe Fitting	Equivalent Length
Long radius elbow	11 ft (3.4 m)
Open swing check valve	40 ft (12.2 m)
Open gate valve	3.5 ft (1.1 m)

Figure 39.11 shows the resistance coefficients for different pipe fittings. Assuming that a basket-type strainer is installed at the suction pipe inlet, the entrance loss, h_e (ft) is given by

$$h_e = \frac{K v^2}{2g}$$

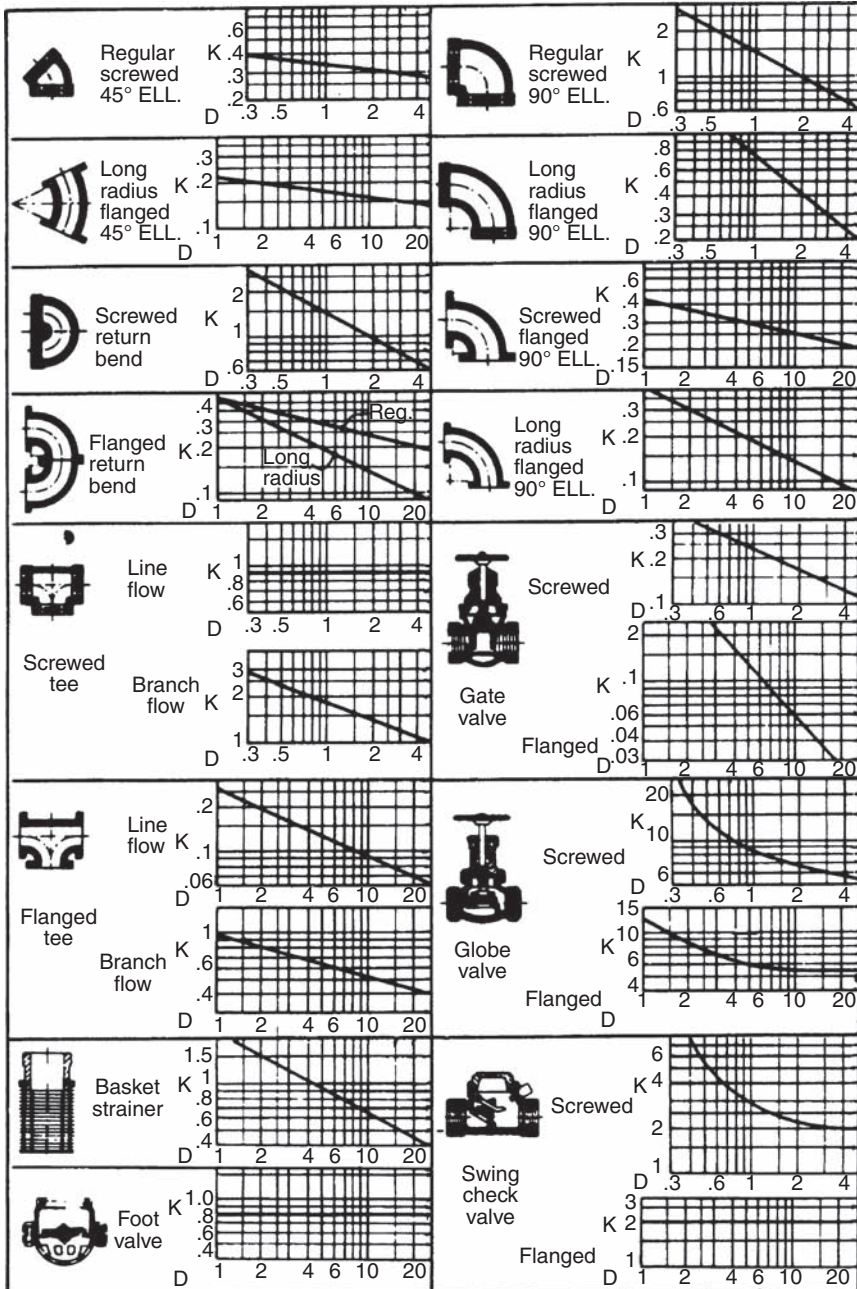
where k = constant obtained from Fig. 39.11, in this case, $K = 0.84$

v = liquid velocity in ft/s

g = gravitational acceleration [32.2 ft/s² (980.67 cm/s²)]

Pipe Size		Standard Ell		Medium-Radius Elbow		Long-Radius Elbow		45° Elbow		Tee		Gate Valve, Open		Globe Valve, Open		Swing Check, Open	
		ft	m	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m
6	152.4	16	4.9	14	4.3	11	3.4	7.7	2.3	33	10.1	3.5	1.1	160	48.8	40	12.2
8	203.2	21	6.4	18	5.5	14	4.3	10	3.0	43	13.1	4.5	1.4	220	67.0	53	16.2
10	254.0	26	7.9	22	6.7	17	5.2	13	3.9	56	17.1	5.7	1.7	290	88.4	67	20.4
12	304.8	32	9.8	26	7.9	20	6.1	15	4.6	66	20.1	6.7	2.0	340	103.6	80	24.4

TABLE 39.3 Resistance of Fittings and Valves (Length of Straight Pipe Giving Equivalent Resistance)



$$h = k \frac{v^2}{2g} \text{ feet of fluid}$$

FIGURE 39.11 Resistance coefficients of pipe fittings. To convert to SI in the equation for h , v^2 would be measured in m/s and feet would be changed to meters. The following values would also be changed from inches to millimeters: 0.3 to 7.6, 0.5 to 12.7, 1 to 25.4, 2 to 50.8, 4 to 101.6, 6 to 152.4, 10 to 254, and 20 to 508. (Source: Hydraulic Institute, Parsippany, NJ.)

The exit loss, h_{ex} (ft) is caused by the sudden enlargement of the flow when it discharges from the pipe into a tank. The final velocity of the flow reaches zero due to the large tank area. The exit loss is given by

$$h_{ex} = \frac{v^2}{2g}$$

The velocity v (ft/s) in the pipe is given by

$$v = \frac{\text{gpm}}{2.448d^2}$$

In this application,

$$v = \frac{1,500}{2.448(6)^2} = 17.02 \text{ ft/s} (5.17 \text{ m/s})$$

Then,

$$h_e = \frac{0.84(17.02)^2}{2 \times 32.2} = 3.78 \text{ ft} (1.15 \text{ m})$$

and

$$h_{ex} = \frac{(17.02)^2}{2 \times 32.2} = 4.5 \text{ ft} (1.37 \text{ m})$$

Therefore, the total length of the piping system is

$$130 + 4(11) + 40 + 3.5 + 3.78 + 4.5 = 225.78 \text{ ft} (68.64 \text{ m})$$

Table 39.4 shows the pipe friction head loss for water flowing through wrought-iron or steel schedule 40 pipe in a good condition. In this case, for 6 in (15.24 cm) and 1500 gal/min (94.65 L/s), the head loss per 100 ft (30.5 m) of pipe is 15 ft (4.56 m) of water.

Since the total length of the piping system is 225.78 ft (68.64 m), the total friction head loss for this application is

$$H_f = (15) \left(\frac{225.78}{100} \right) = 33.87 \text{ ft} (10.3 \text{ m}) \text{ of water}$$

Calculate the total head on the pump

The total head on the pump, H_t , is given by

$$H_t = H_{is} + H_f$$

For the application shown in Fig. 39.10a,

$$H_t = 110 + 33.87 = 143.87 \text{ ft} (43.74 \text{ m})$$

The total head on the pump in the applications shown in Fig. 39.10b and c is the same as the one for the application shown in Fig. 39.10a. The total head on the pump is also known as the "total dynamic head." This is done to differentiate it from the total static head on the pump.

The total head on the pumps shown in Fig. 39.10d, e, and f is calculated in a similar fashion. However, in these applications, the total static head on the pump is given by

$$H_{is} = \text{static discharge head } (h_{sd}) - \text{static suction head } (h_{sh})$$

Diameter		Flow		Velocity		Velocity Head		Friction Loss per 100 ft (30.5 m) of Pipe	
in	mm	gal/min	L/s	ft/s	m/s	ft water	m water	ft water	m water
6	152.4	1000	63.1	11.1	3.4	1.92	0.59	6.17	1.88
6	152.4	2000	126.2	22.2	6.8	7.67	2.3	23.8	7.25
6	152.4	4000	252.4	44.4	13.5	30.7	9.4	93.1	28.4
8	203.2	1000	63.1	6.41	1.9	0.639	0.195	1.56	0.475
8	203.2	2000	126.2	12.8	3.9	2.56	0.78	5.86	1.786
8	203.2	4000	252.4	25.7	7.8	10.2	3.1	22.6	6.888
10	254.0	1000	63.1	3.93	1.2	0.240	0.07	0.497	0.151
10	254.0	3000	189.3	11.8	3.6	2.16	0.658	4.00	1.219
10	254.0	5000	315.5	19.6	5.9	5.99	1.82	10.8	3.292

TABLE 39.4 Pipe Friction Loss for Water (Wrought-Iron or Steel Schedule 40 Pipe in Good Condition)

Calculate the power requirement of the pump

The power requirement of the pump, known also as “brake hp input to the pump,” is given by

$$\text{bhp} = \frac{(\text{gpm})(H_t)(s)}{3960e}$$

where s is specific gravity of the liquid handled, and e is pump hydraulic efficiency, in a decimal form (this term decreases as the liquid velocity increases).

Assuming that $e = 0.65$ in this application, and $s = 1$. Then,

$$\text{bhp} = \frac{(1500)(143.87)(1)}{3960(0.65)} = 83.84 \text{ (62.54 kW)}$$

The procedure used in this section can be used for any type of pump (centrifugal, reciprocating, or rotary).

39.9 Pump Selection Procedure

The following is a detailed procedure for selecting the type, class, capacity, materials, and a drive for a pump that will be used in an industrial application.

39.9.1 Draw the Proposed Piping Layout of the Pumping System

Draw a single-line diagram of the piping system similar to the one shown in Fig. 39.12. Show all the items installed on the system including piping, fittings, pumps, valves, tanks, etc. Mark the *equivalent* pipe length (described in the previous section) on the diagram.

39.9.2 Determine the Required Pump Capacity

The required capacity (flow rate) of the pump should be obtained from the system requirements. A safety factor should be added to the required flow rate. It is normally 10 to 15% of the required flow rate. However, in some applications it can be higher than 50%.

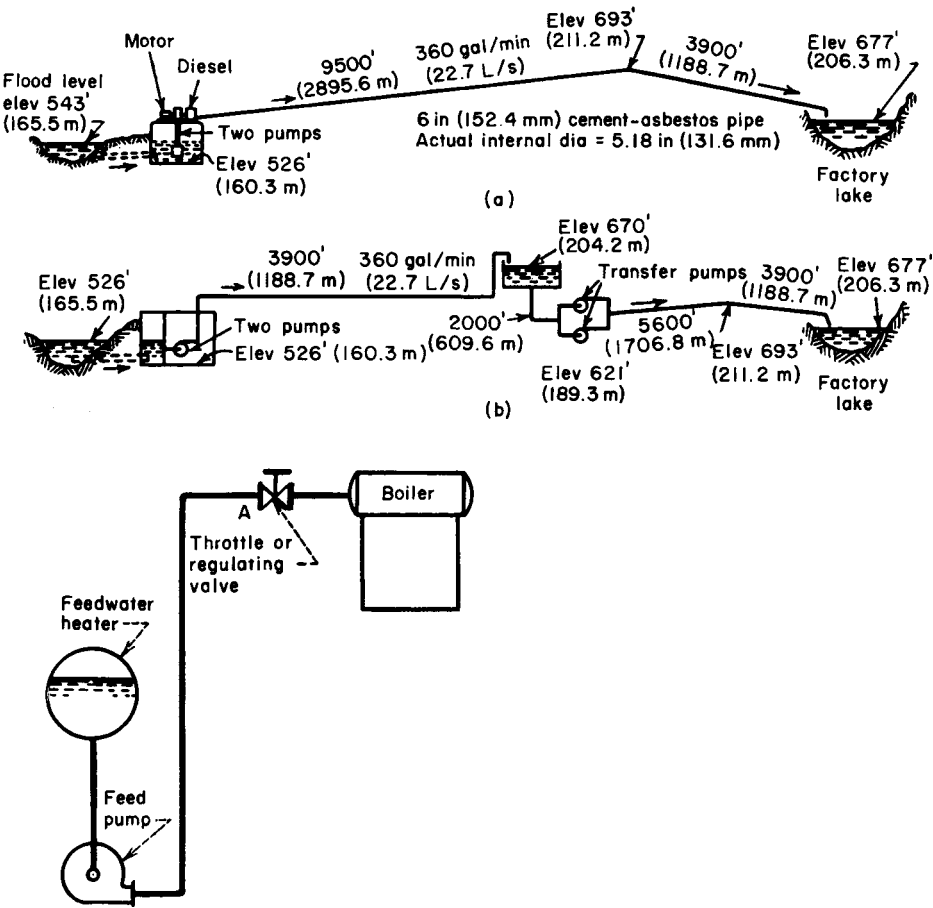


FIGURE 39.12 (a) Single-line diagrams for an industrial pipeline; (b) single-line diagram of a boiler-feed system. (Source: Worthington Corporation.)

39.9.3 Determine the Total Head on the Pump

The calculation procedure that was described in the previous section should be used to calculate the total head on the pump. The results should be expressed in ft (m) of water.

39.9.4 Obtain the Physical and Chemical Data of the Liquid Being Pumped

Obtain the complete physical and chemical data of the liquid being pumped including specific gravity, viscosity, chemical formula, temperature, pressure, vapor pressure, pH, flash point, ignition temperature, solid content, type of solids entrained in the liquid and their size. All the variations (minimum and maximum values) in these parameters as well as any unusual characteristics such as tendency to foam, crystallize, or become gelatinous should also be known.

A pump selection form similar to the one shown in Fig. 39.13 should be filled. These forms are provided by all pump manufacturers at the beginning of the selection process.

Summary of Essential Data Required in Selection of Centrifugal Pumps

1. Number of Units Required
2. Nature of the Liquid to Be Pumped
Is the liquid:
 - a. Fresh or salt water, acid or alkali, oil, gasoline, slurry, or paper stock?
 - b. Cold or hot and if hot, at what temperature? What is the vapor pressure of the liquid at the pumping temperature?
 - c. What is its specific gravity?
 - d. Is it viscous or nonviscous?
 - e. Clear and free from suspended foreign matter or dirty and gritty? If the latter, what is the size and nature of the solids, and are they abrasive? If the liquid is of a pulpy nature, what is the consistency expressed either in percentage or in lb per cu ft of liquid? What is the suspended material?
 - f. What is the chemical analysis, pH value, etc.? What are the expected variations of this analysis? If corrosive, what has been the past experience, both with successful materials and with unsatisfactory materials?
3. Capacity
What is the required capacity as well as the minimum and maximum amount of liquid the pump will ever be called upon to deliver?
4. Suction Conditions
Is there:
 - a. A suction lift?
 - b. Or a suction head?
 - c. What are the length and diameter of the suction pipe?
5. Discharge Conditions
 - a. What is the static head? Is it constant or variable?
 - b. What is the friction head?
 - c. What is the maximum discharge pressure against which the pump must deliver the liquid?
6. Total Head
Variations in items 4 and 5 will cause variations in the total head.
7. Is the service continuous or intermittent?
8. Is the pump to be installed in a horizontal or vertical position? If the latter,
 - a. In a wet pit?
 - b. In a dry pit?
9. What type of power is available to drive the pump and what are the characteristics of this power?
10. What space, weight, or transportation limitations are involved?
11. Location of installation
 - a. Geographical location
 - b. Elevation above sea level
 - c. Indoor or outdoor installation
 - d. Range of ambient temperatures
12. Are there any special requirements or marked preferences with respect to the design, construction, or performance of the pump?

FIGURE 39.13 Typical selection chart for centrifugal pumps. (Source: Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006 with permission from McGraw-Hill.)

39.9.5 Select the Category and Type of Pump

The different categories (known also as classes) and types of pumps are listed in Fig. 39.14. Use Table 39.5 as a guide to select the category and type of pump required for a specific application. For example, a reciprocating pump is chosen for applications requiring the highest head and small flow.

The liquid properties should also be considered when selecting the category and type of a pump. For example, a rotary type pump is selected to handle viscous and

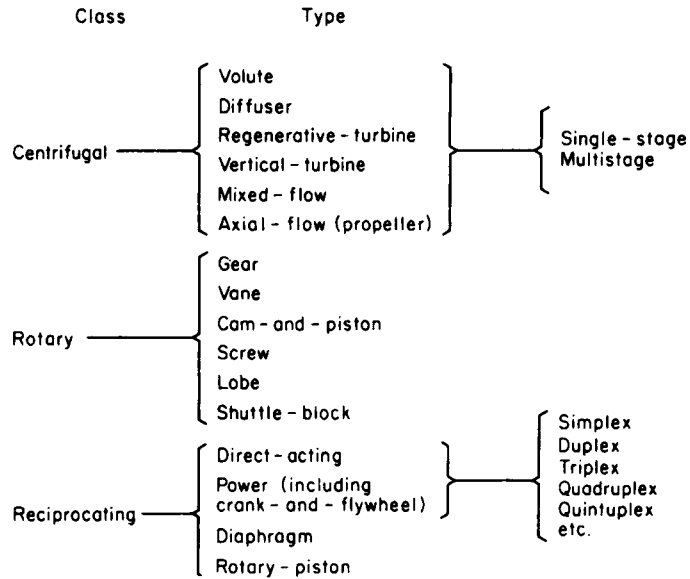


FIGURE 39.14 Modern pump classes and types. (Source: Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006 with permission from McGraw-Hill.)

Pump Variable	Centrifugal		Rotary	Reciprocating		
	Volute and Diffuser	Axial Flow	Screw and Gear	Direct Acting Steam	Double Acting Power	Triplex
Discharge flow	Steady	Steady	Steady	Pulsating	Pulsating	Pulsating
Usual maximum suction lift, ft (m)	15 (4.6)	15 (4.6)	22 (6.7)	22 (6.7)	22 (6.7)	22 (6.7)
Liquids handled	Clean, clear; dirty, abrasive; liquids with high solids content		Viscous; non-abrasive	Clean and clear		
Discharge pressure range	Low to high		Medium	Low to highest produced		
Usual capacity range	Small to largest available		Small to medium	Relatively small		
How increased head affects:						
Capacity	Decrease		None	Decrease	None	None
Power input	Depends on specific speed		Increase	Increase	Increase	Increase
How decreased head affects:						
Capacity	Increase		None	Small increase	None	None
Power input	Depends on specific speed		Decrease	Decrease	Decrease	Decrease

TABLE 39.5 Characteristics of Modern Pumps

nonabrasive liquids. A centrifugal pump is selected to handle abrasive liquids and liquids having high solid content.

The operating factors of the pump should also be considered during the selection process. They include the type of service of the pump (continuous or intermittent), operating speed, and future plans for the application.

Following selection of the category and type of pump, use Table 39.6 (rating table) or Fig. 39.15 (rating chart) to select the speed (rpm) and actual input power required to operate the pump. For example, if the pump capacity is 400 gal/min (25.2 L/s) and the total head is 20 ft, Table 39.6 indicates that the pump speed is 1200 rpm and the actual input required to operate the pump is 3.1 hp (2.31 kW). If the hydraulic requirements fall between two standard pump models, the next larger pump size is selected. For example, if the hydraulic requirements are 350 gal/min (22.05 L/s) and the total head is 20 ft (6.1 m), Table 39.6 indicates that a pump model having a speed of 1200 rpm and actual input power of 3.1 hp (2.31 kW) should be selected.

The pump manufacturer normally provides additional information such as the characteristics when the diameter of the impeller is varied within the same casing (Fig. 39.16) and the variable speed head capacity for a given impeller diameter (Fig. 39.17).

Figure 39.18 shows the performance data (head and capacity ranges) for different types of rotary pumps. Table 39.7 shows the capacity data for different typical horizontal duplex plunger pumps.

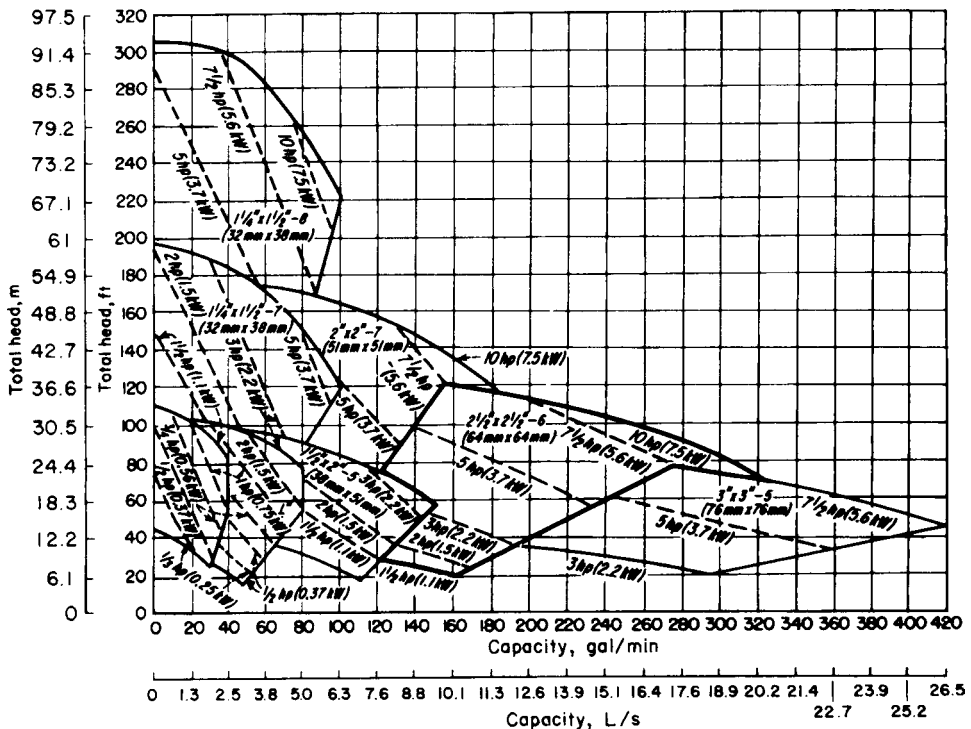


FIGURE 39.15 Composite rating chart for a typical centrifugal pump. (Source: Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006 with permission from McGraw-Hill.)

Size		Total Head			
gal/min	L/s	20 ft, r/min—hp	6.1 m, r/min—kW	25 ft, r/min—hp	7.6 m, r/min—kW
3 CL:					
200	12.6	910—1.3	910—0.97	1010—1.6	1010—1.19
300	18.9	1000—1.9	1000—1.41	1100—2.4	1100—1.79
400	25.2	1200—3.1	1200—2.31	1230—3.7	1230—2.76
500	31.5	—	—	—	—
4 C:					
400	25.2	940—2.4	940—1.79	1040—3	1040—2.24
600	37.9	1080—4	1080—2.98	1170—4.6	1170—3.43
800	50.5	—	—	—	—

Example: 1080—4 indicate pump speed is 1080 r/min; actual input required to operate the pump is 4 hp (2.98 kW).

Source: Condensed from data of Goulds Pumps, Inc.; SI values added by handbook editor.

TABLE 39.6 Typical Centrifugal-Pump Rating Table

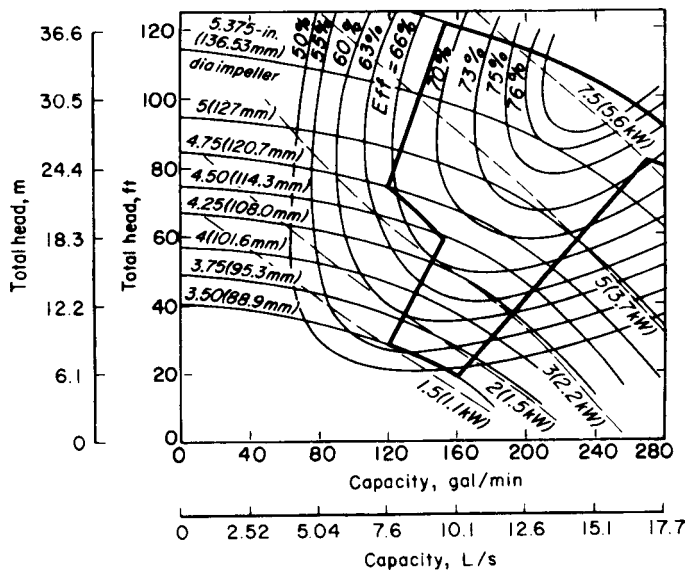


FIGURE 39.16 Pump characteristics when impeller diameter is varied within the same casing. (Source: Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006 with permission from McGraw-Hill.)

39.9.6 Evaluate the Selected Pump

Calculate the specific speed of the pump. Then use Fig. 39.9 to determine the impeller type and the efficiency of the pump.

Confirm using the piping system calculations described earlier that the available net positive suction head is greater than the required net positive suction head of the pump.

FIGURE 39.17 Variable-speed head-capacity curves for a centrifugal pump. (Source: Hicks, T., *Handbook of Mechanical Engineering Calculations*, 2nd ed., McGraw-Hill, New York, 2006 with permission from McGraw-Hill.)

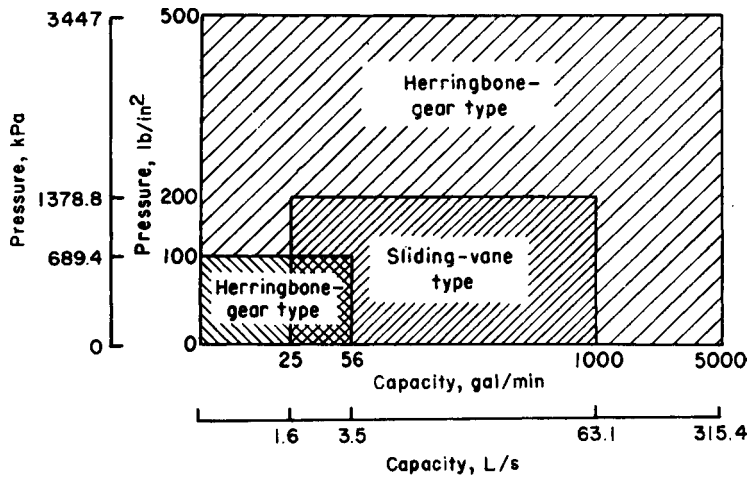
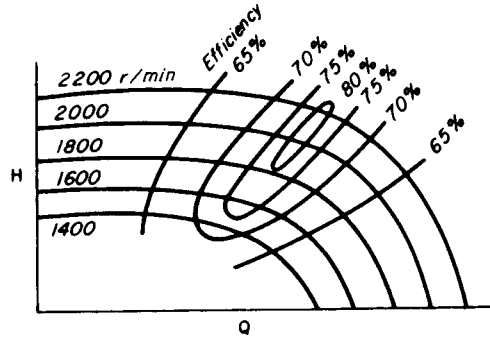


FIGURE 39.18 Capacity ranges of some rotary pumps. (Sources: Worthington Corporation.)

Size		Cold-Water Pressure Service			
		gal/min	L/s	Piston Speed	
in	cm			ft/min	m/min
6 × 3½ × 6	15.2 × 8.9 × 15.2	60	3.8	60	18.3
7½ × 4½ × 10	19.1 × 11.4 × 25.4	124	7.8	75	22.9
9 × 5 × 10	22.9 × 12.7 × 25.4	153	9.7	75	22.9
10 × 6 × 12	25.4 × 15.2 × 30.5	235	14.8	80	24.4
12 × 7 × 12	30.5 × 17.8 × 30.5	320	20.2	80	24.4

TABLE 39.7 Capacities of Typical Horizontal Duplex Plunger Pumps

Size		Boiler-Feed Service					
		gal/min	L/s	Boiler		Piston Speed	
in	cm			hp	kW	ft/min	m/min
6 × 3½ × 6	15.2×8.9×15.2	36	2.3	475	354.4	36	10.9
7½ × 4½ × 10	19.1×11.4×25.4	74	4.7	975	727.4	45	13.7
9×5×10	22.9×12.7×25.4	92	5.8	1210	902.7	45	13.7
10×6×12	25.4×15.2×30.5	141	8.9	1860	1387.6	48	14.6
12×7×12	30.5×17.8×30.5	192	12.1	2530	1887.4	48	14.6

Source: Courtesy of Worthington Corporation.

TABLE 39.7 Capacities of Typical Horizontal Duplex Plunger Pumps (*Continued*)

39.10 Bibliography

Hicks, T. G., *Handbook of Mechanical Engineering Calculations*. McGraw-Hill, New York, 1997.

CHAPTER 40

Bearings

The function of bearings is to keep the shaft or rotor properly aligned with the stationary parts under axial and radial loads.

Bearings that provide radial positioning to rotors are known as *line* or *radial bearings*. Bearings that position the rotor axially are called *thrust bearings*. Thrust bearings usually serve as thrust and radial bearings

40.1 Types of Bearings

The main types of bearings are journal and rolling-contact (ball and roller) bearings. The ball and roller bearings are also known as antifriction bearings. The most common bearings used on centrifugal pumps are the various types of ball bearings. Roller bearings are used less often. However, spherical roller bearings are used frequently for large shaft sizes, for which there is a limited choice of ball bearings.

In horizontal pumps with bearings on each end, *inboard* bearings are located between the casing and the coupling. The bearing on the outside is the *outboard*. The bearings are usually mounted in a housing supported by brackets which are attached or integral to the pump casing. The lubricant is usually contained within the housing. The bearings must be kept within proper temperature limits due to the heat generated within the bearings.

For bearings that are provided with a forced-feed lubrication system, cooling is achieved by circulating the oil through a separate water-to-oil cooler. Otherwise, cooling liquid enters a jacket located inside the housing to remove the heat. There are rigid and self-aligning bearings. A self-aligning bearing will adjust itself automatically to a change in the shaft angular position.

40.1.1 Ball and Roller Bearings

Since the coefficient of rolling friction is less than that of sliding friction, ball bearings offer a significant advantage over sleeve bearings.

In ball bearings, the load is carried on the point of contact between the ball and the race. During normal operation (constant speed), the point of contact does not rub or slide over the race. The heat generated is minimal. Also, the point of contact is constantly changing because the ball rolls in the race. The bearing operation is almost *frictionless*.

In sleeve bearings, the surfaces always rub against each other. Lubrication is used to reduce the friction. The starting friction is only *slightly* larger than the operating friction in rolling-contact bearings. This is a significant advantage of rolling-contact

(ball and roller) bearings. It makes them suitable for machines that are started and stopped frequently under load.

Rolling-contact bearings require little maintenance and lubrication. However, they are noisier than journal bearings and more expensive. The lifetime of rolling-contact bearings is limited due to fatigue failure of the raceways caused by the repeated high stresses as the shaft rotates. However, most failures are caused by lubrication failure or improper mounting. Several types of rolling-contact bearings are capable of handling radial load and axial thrust. Although they are called “rolling-contact bearings,” the balls and rollers do considerable sliding during normal operation due to changes in speed.

40.2 Stresses During Rolling Contact

The stresses during rolling contact are high due to the small area of contact. The sliding motion (frictional force) causes tensile stress near the surface which results in pitting.

Since the contact stresses are usually greater than the yield strength, *residual* compressive stresses are induced in the races and the balls. These stresses cause deformations which increase with repetitions.

Since the area of contact between the balls and the races is small, the stresses in a particular point in a bearing race are very high. Fatigue failure is expected due to the repeated nature of the stresses. Figure 40.1 illustrates a single-row, deep-groove ball bearing.

Rolling bearings have a limited life because any normal loading creates stress higher than the fatigue strength involved.

The bearing life is limited by the number of stress repetitions to cause fatigue failure. For example, doubling the speed of rotation reduces the bearing life by one-half. The centrifugal forces within the bearings become significant as the speed increases. A bearing has a shorter life at very high rpm (revolutions per minute). The load has a significant impact on the bearing life. The bearing life B was found experimentally to

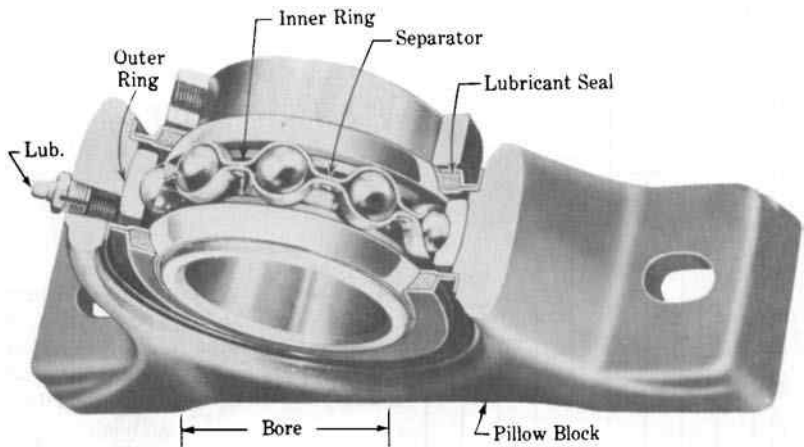


FIGURE 40.1 Single-row, deep-groove ball bearing mounted in bearing block. (Courtesy of Link-Belt Co., Chicago, Ill.)

vary inversely as the k power of the load F ; B is proportional to $1/F^k$, where $k = 3$ for ball bearings and $10/3$ for roller bearings. In equation form

$$\left(\frac{F_1}{F_2}\right)^k = \frac{F_1^k}{F_2^k} = \frac{B_2}{B_1} \quad \text{or} \quad \frac{F_1}{F_2} = \left(\frac{B_2}{B_1}\right)^{1/k}$$

Note that for ball bearings ($k = 3$), the life increases eight times when the load decreases by a half. Corrosion from acid or water reduces the life of bearings. The bearing life is usually measured in millions of revolution. However, some catalogs give the bearing life by the number of hours at various rotating speeds. The signs of fatigue are spalling or flaking of small particles of material from the surface of a ring or a ball (roller) caused by the high shearing stress below the contact surfaces. This mechanism of failure is preceded by noisy operation. Continued operation will eventually lead to fatigue failure.

40.3 Statistical Nature of Bearing Life

The life of rolling bearings is rated on the basis that 90% of a large number of bearings in particular surroundings will survive a specific life. This life is usually designated as B_{10} , which is interpreted as a failure rate of 10% expected within B .

However, other catalogs give the rated load for 50% survival, B_{50} (median life). A commonly accepted relationship is

$$\text{Median life} = (5) \text{ (90\% life); } B_{50} = 5 B_{10}$$

Table 40.1 illustrates the variations in design life depending on the machine and service.

Type of Service	Hours (90% life)
Infrequent use—instruments, demonstration apparatus, sliding doors.....	500
Aircraft engines.....	500–2000
Intermittent use, service interruptions of minor importance—hand tools, hand-driven machines, farm machinery, cranes, household machines.....	4000–8000
Intermittent use, dependable operation important—work moving devices in assembly lines, elevators, cranes, and less frequently used machine tools.....	8000–12,000
8-hour service, not fully utilized—gear drives, electric motors.....	12,000–20,000
8-hour service, fully utilized—machines in general, cranes, blowers, shop shafts.....	20,000–30,000
24-hour service, continuous operation—separators, compressors, pumps, conveyor rollers, mine hoists, electric motors.....	40,000–60,000
24-hour service, dependable operation important—machines in continuous-process plants, such as paper, cellulose; power stations, pumping stations, continuous service machines aboard ships.....	100,000–200,000

TABLE 40.1 Design Life for Rolling Bearings, Hours

40.4 Materials and Finish

The most common material used in bearings is SAE 52100. It is an alloy steel of nominally 1% carbon (C) and 1.5% chromium (Cr), hardened to Rockwell C58-65. Alloys such as nickel and molybdenum are used with chromium.

The bearing life depends heavily on the hardness. The average life of a bearing when the hardness is $R_o = 50$ is about half of that when $R_o = 60$. The operating temperature of SAE 52100 and similar steels is usually held to 300°F. However, the maximum temperature for the installation is 200°F.

40.5 Sizes of Bearings

Table 40.2 shows several series of bearings and their key dimensions that have been standardized.

The bearing is classified as heavier by increasing the outside diameter as illustrated in Fig. 40.2.

40.6 Types of Rolling Bearings

Figure 40.1 has illustrated the *deep-groove ball bearing*. Figure 40.3 illustrates a filling-slot-type ball bearing. The slots or notches permit the assembly of more balls giving the bearing larger radial load capacity. Both bearings are used for radial loads only.

Figure 40.4 illustrates a *self-aligning ball bearing*. It compensates for angular misalignments which arise from the deflection of the shaft or foundation or from errors in mounting. They are recommended for radial loads and moderate thrust in either direction.

Figure 40.5 illustrates an *angular-contact bearing*. There is an angle between the line through the areas carrying the load and the plane of the bearing face. These bearings are used to carry heavy thrust loads.

They are usually used in opposed pairs. They are also suited for *preloading*. Preloading a bearing is placing it under an initial axial load which is independent of the working load to maintain a constant alignment of parts and reducing the axial movement as well as radial deflection under working loads.

Double-row ball bearings (not self-aligning) are similar to the single-row ball bearings. The only exception is that each ring has two grooves. The two rows of balls give a bearing capacity slightly less than twice that of a single row.

Figure 40.6 illustrates a cylindrical roller bearing. The contact is a line instead of a point as the case is in ball bearings. This results in a greater area to carry the load. Hence, for a particular size, the cylindrical roller bearing has a larger radial capacity than a ball bearing.

The main function of the retainer is to keep the roller axes parallel.

A popular type of cylindrical roller bearings is the *needle bearing* shown in Fig. 40.7. This type does not have a retainer to hold the rollers in alignment. Their main advantage is the small diametral dimension. They are used where the speed is relatively low and where there is oscillating motion.

BKG. NO.	Bore		Outside Diameter, mm			Width of Races, mm			Max. Fillet, r		
	mm.	in.	200 Series	300 Series	400 Series	200 Series	300 Series	400 Series	200 Series	300 Series	400 Series
00	10	0.3937	30	35	—	9	11	—	0.024	0.024	—
01	12	0.4724	32	37	—	10	12	—	0.024	0.039	—
02	15	0.5906	35	42	—	11	13	—	0.024	0.039	—
03	17	0.6693	40	47	—	12	14	—	0.024	0.039	—
04	20	0.7874	47	52	—	14	15	—	0.039	0.039	—
05	25	0.9843	52	62	80	15	17	21	0.039	0.039	0.059
06	30	1.1811	62	72	90	16	19	23	0.039	0.039	0.059
07	35	1.3780	72	80	100	17	21	25	0.039	0.059	0.059
08	40	1.5748	80	90	110	18	23	27	0.039	0.059	0.079
09	45	1.7717	85	100	120	19	25	29	0.039	0.059	0.079
10	50	1.9685	90	110	130	20	27	31	0.039	0.079	0.079
11	55	2.1654	100	120	140	21	29	33	0.059	0.079	0.079
12	60	2.3622	110	130	150	22	31	35	0.059	0.079	0.079
13	65	2.5591	120	140	160	23	33	37	0.059	0.079	0.079
14	70	2.7559	125	150	180	24	35	42	0.059	0.079	0.098
15	75	2.9528	130	160	190	25	37	45	0.059	0.079	0.098
16	80	3.1496	140	170	—	26	39	—	0.079	0.079	—
17	85	3.3465	150	180	—	28	41	—	0.079	0.098	—
18	90	3.5433	160	190	—	30	43	—	0.079	0.098	—
19	95	3.7402	170	200	—	32	45	—	0.079	0.098	—
20	100	3.9370	180	215	—	34	47	—	0.079	0.098	—
21	105	4.1339	190	225	—	36	49	—	0.079	0.098	—
22	110	4.3307	200	240	—	38	50	—	0.079	0.098	—

This table does not give all standard dimensions. The maximum fillet radius r is the maximum radius at the shoulder on the shaft, which is cleared by the corner radius on the bearing. Conversion factor: 0.03937 in./mm; 25.4 mm/in.

TABLE 40.2 Dimensions of Rolling Bearings

Figure 40.8 illustrates a *tapered roller bearing*. The rolling elements are arranged as frustums of a cone. All their axes intersect at a point on the axis of the shaft. These bearings are capable of earning significant axial loads.

Figure 40.9 illustrates a *spherangular roller bearing*. This is an angular contact bearing with rollers instead of balls. The bearing is self-aligning because the outer race has a spherical surface. It has also a high load capacity due to its large contact area.

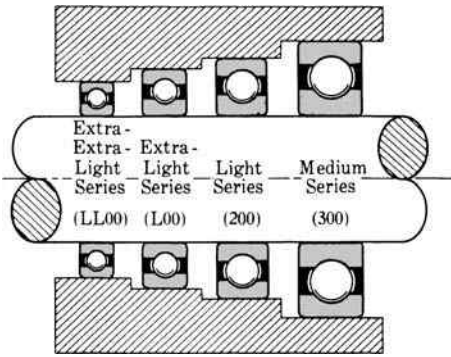


FIGURE 40.2 Various series of ball bearings. All these bearings have the same basic number, as 09 (after new departure).

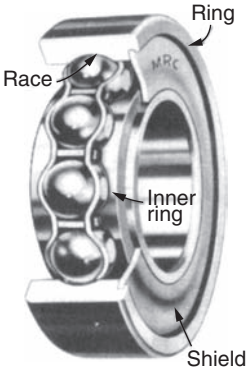


FIGURE 40.3 Single-row ball bearing with shield. The balls in this bearing were loaded with the aid of a filling slot. Notice their spacing. The shield aids in keeping out foreign matter, important in rolling bearings. (Courtesy of Marlin-Rockwell Corp., Jamestown, N.Y.)

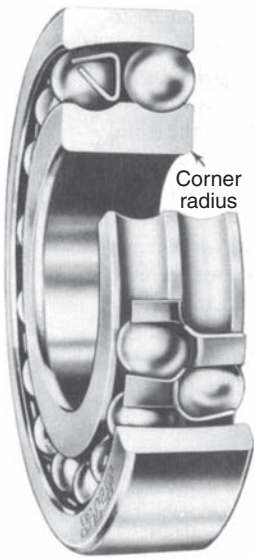


FIGURE 40.4 Self-aligning ball bearing. Self-aligning roller bearings are similar. (Courtesy of SKF Industries, Inc., Philadelphia, Penn.)

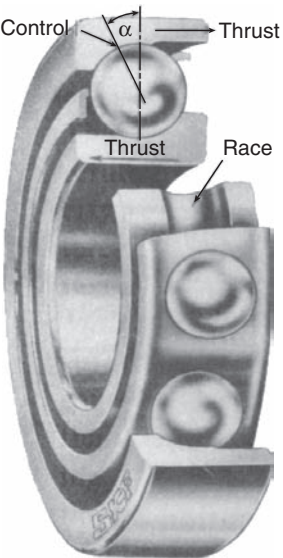


FIGURE 40.5 Angular Contact Bearing. Observe that the action of the thrust is such as to move the surface of contact away from the center-line plane of the balls. Compare the shape of the grooves in this figure with those of Figs. 40.1 and 40.3. (Courtesy of SKF Industries, Inc., Philadelphia, Penn.)

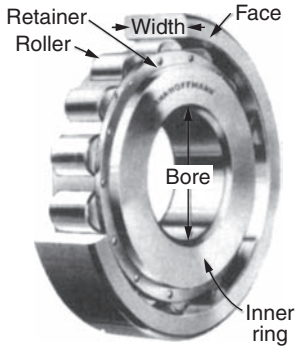


FIGURE 40.6 Cylindrical roller bearing. The rollers run in a groove in the inner ring. (Courtesy of Norma-Hoffman Bearings Corp., Stamford, Conn.)

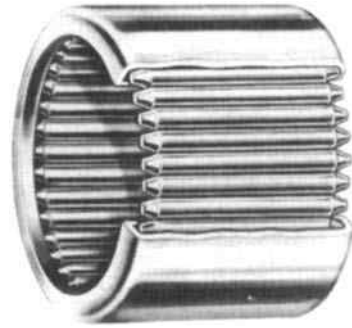


FIGURE 40.7 Needle bearing. (Courtesy of The Torrington Co., Torrington, Conn.)



FIGURE 40.8 Tapered roller bearing. (Courtesy of Timken Roller Bearing Co., Canton, Ohio.)



FIGURE 40.9 Spherangular roller bearing. (Courtesy of Hyatt Roller Bearing Co., Harrison, N.J.)

40.6.1 Thrust Bearings

The rolling type of thrust bearings use the following as rolling elements: balls, short cylindrical rollers, tapered rollers, and spherical rollers that run in spherical races (self-aligning).



FIGURE 40.10 Thrust ball bearing. (Courtesy of Aetna Ball & Roller Bearing Co., Chicago, Ill.)



FIGURE 40.11 Tapered roller thrust bearing. (Courtesy of Timken Roller Bearing Co., Canton, Ohio.)

These bearings can be rigidly supported or one of the races can be supported in a spherical seat to make it self-aligning.

Figures 40.10 and 40.11 illustrate thrust bearings that use balls and tapered rollers as rolling elements, respectively.

CHAPTER 41

Lubrication

41.1 The Viscosity of Lubricants

The viscosity is defined as the resistance of a liquid to flow or deformation. In general, the slower a fluid flows under a given pressure drop, the higher is its viscosity. The ability of a lubricant to maintain an oil film (which reduces friction and wear) between the working parts of machines is determined by viscosity.

Figure 41.1 illustrates two parallel plates separated by a film of oil. The top plate is moving; the bottom plate is stationary. A force is applied to the top plate to keep it moving at a steady speed.

The shear stress, $S_s = \text{force/area}$

The shear rate, $R = \text{speed/thickness}$

The viscosity, $V = S_s / R$ (Newton's equation)

Water and most industrial oils follow this equation (Newtonian fluids). The shear stress increases in direct proportion to shear rate. The viscosity is independent of the shear stress or shear rate at which is measured. Non-Newtonian fluids exhibit nonlinear relationship between the shear stress and the shear rate. Greases and oils below their solidification temperature are non-Newtonian lubricants. The equation of viscosity can be rearranged to show that the force is proportional to the viscosity and the speed is inversely proportional to the viscosity.

41.1.1 Viscosity Units

The physical quantity in the equation above is the dynamic viscosity (also called absolute viscosity). However, the kinematic viscosity is traditionally used when referring to lubricants. The two measures of viscosity are related by density as shown below:

$$\text{Dynamic viscosity} = \text{kinematic viscosity} \times \text{density}$$

The friction in bearings or pressure loss in horizontal pipe flow is not affected by density. Therefore, the *dynamic viscosity* is the controlling measure. However, density has a significant effect on gravitational flow, such as flow from a reservoir through a hose. The driving force is proportional to the density in this situation and the *kinematic viscosity* is the controlling measure.

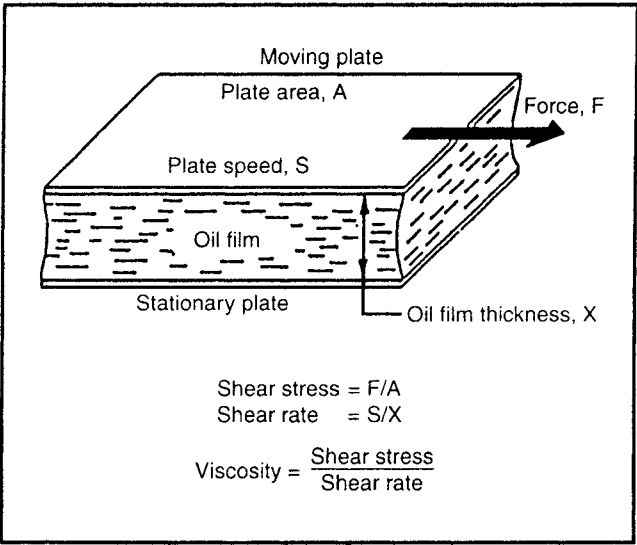


FIGURE 41.1 Schematic diagram showing definition of viscosity.

The lubrication industry has traditionally used the kinematic viscosity. This viscosity is obtained by measuring the flow rate from a reservoir through a small capillary. The unit of kinematic viscosity in the International System (SI) is mm/s. However, the lubrication industry has traditionally used the term centistokes (1 cSt = 1 mm/s). The unit of dynamic viscosity in SI is pascal $\times$ second (pa $\times$ s). However, the lubrication industry has traditionally used the term centipoise (1 cP = 1 mPa $\times$ s).

The following equation is used to convert from kinematic viscosity to dynamic viscosity:

$$\text{Viscosity (cP)} = \text{viscosity (cSt)} \times \text{density (g/mL)}$$

The density is given at the temperature of the viscosity measurements.

The Saybolt Universal seconds (SUS) is an older measure of the viscosity, which is still used occasionally in the U.S. Figure 41.2 illustrates the relationship between the kinematic viscosity and the SUS.

A reference point for viscosity is water at room temperature that has a viscosity of one cSt approximately. The viscosity of a typical engine oil at 280°F is 5 to 6 cSt. The viscosity of the same oil at -25°C is between 3000 and 3500 cP (temperature has a significant effect on viscosity).

41.1.2 Significance of Viscosity

Viscosity is the most important property of lubricating oils. In bearings, oil prevents wear between mating surfaces by keeping them separated, minimize friction between rubbing surfaces, and remove heat generated by surface contact. Viscosity is important in all these functions.

The role of viscosity is illustrated in plain journal bearings. When the viscosity is high, the bearing load is completely separated by the *oil film*. The journal does not contact the

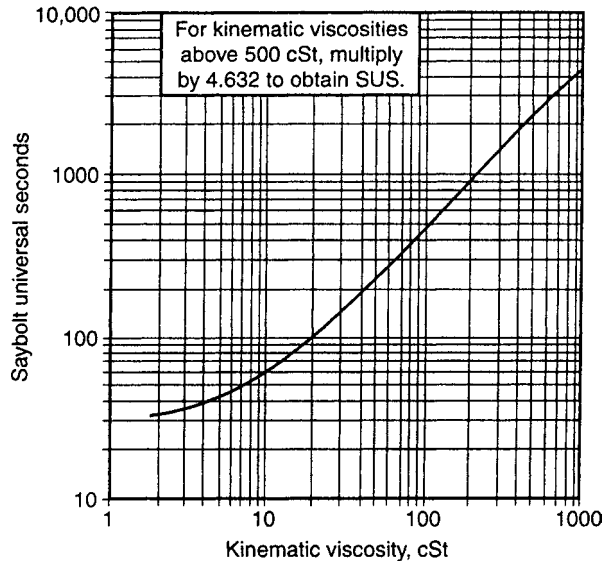


FIGURE 41.2 Relationship between kinematic viscosity (cSt) and Saybolt Universal seconds (SUS).

bearing and the friction is low. When the viscosity is low, the metal-to-metal contact and the friction is high. In this case, the oil viscosity has a minor effect. The antiwear properties of the oil play a major role. The antiwear properties of the oil play a minor role when the viscosity is high enough to support the bearing load completely. The friction between the moving parts is reduced by lubricants even if the oil film does not carry the load. The effect of the lubricant viscosity is to “cushion” the asperity-to-asperity (microscopic hills) contact between moving surfaces to reduce the local stresses. The contact between the parts in roller bearings and gear teeth is along a line (one gear tooth against another). A very thin film is trapped between the surfaces due to the high contact pressure. The pressure in these contact zones increase to several hundred thousand psi, which results in significant increase in viscosity. The viscosity of the oil becomes high enough to keep the moving surfaces separated. This mechanism of lubrication is called *elastohydraulic lubrication*. The oil viscosity at these high pressures has a major role in determining the film thickness in such contacts.

The oil viscosity should be carefully selected to ensure proper lubrication. If the viscosity is low, high wear rates and frequent failures will result. If the viscosity is too high for the application, significant amount of heat would be generated. This may lead to failures due to high temperature. Most manufacturers specify the viscosity required for their machines.

41.1.3 Flow Through Pipes

In laminar flows through pipes, viscosity plays an important role. In this case, the pressure loss is proportional to the fluid viscosity. The flow rate is inversely proportional to the viscosity. When the flow through pipes becomes turbulent, the fluid friction increases by a factor of six. The viscosity has a minor effect on pressure loss while the pipe roughness has a major effect.

41.2 Variation of Viscosity with Temperature and Pressure

41.2.1 Temperature Effect

The viscosity of all fluids is inversely proportional to temperature. The oil viscosity is reported at two standard temperatures (40°C and 100°C).

MacCoull's equation describes the variation of viscosity with temperature of simple lubricants:

$$\log (V_k + 0.7) = A + B \log T$$

where

V_k = kinematic viscosity

T = absolute temperature

A, B = constants for a particular oil

"log" indicates the natural logarithm

MacCoull's equation is not accurate if the oil has additives or, if the viscosity is lower than 1 cSt.

41.2.2 Viscosity Index

The *viscosity index* (VI) characterizes the change of viscosity with temperature. A high VI indicates a small change in viscosity with temperature. A low VI indicates a large change.

41.2.3 Effect of Pressure on Viscosity

The viscosity of all fluids increases with pressure.

41.3 Non-Newtonian Fluids

In Newtonian fluids, the shear stress (force/area) is proportional to the applied shear rate (speed thickness of oil film). Thus, the viscosity which is the ratio of these properties, is constant.

Non-Newtonian fluids exhibit a nonlinear relationship between the shear stress and the shear rate. The viscosity of some non-Newtonian fluids changes with time after a flow starts. Non-Newtonian fluids are characterized by a graph of shear stress versus shear rate.

The three classes of non-Newtonian fluids in the lubrication industry are: greases, VI-improved oils, and oils at low temperature (near the pour points).

41.3.1 Greases

The viscosity of greases is time dependent. The viscosity is relatively high when the flow starts. Then, it decreases as the flow progresses. The yield stress of a grease is the minimum stress required to cause a flow.

41.3.2 VI-Improved Oils

VI improvers are often included in oils to extend the useful range of operating temperature to permit low viscosities for low-temperature start-up, while still providing adequate viscosities at operating temperatures. Organic polymers are the active ingredients in VI improvers. They are extremely large molecules containing many thousands

of carbon atoms. The concentration of polymer in VI-improved oils is about 1%. However, this is sufficient to cause major effects on viscosity. The viscosity of VI-improved oil at 210°F and 0°F is about 1.8 and 1.2 times the viscosity of the base oil, respectively.

The viscosity of polymer-thickened oils (VI improved oils) decreases while the oil is in service. This loss is called *permanent viscosity loss* (PVL). The *shear stability* is the resistance of the oil to this loss.

In engine service, the oil viscosity decreases 7 to 14% of its 100°C kinematic viscosity after 500 to 1500 miles of service.

41.3.3 Oils at Low Temperatures

The viscosity of the oil increases rapidly as the temperature decreases. If the lubricant is used below its pour point, the machinery may experience severe problems. The working parts of the machinery may be covered fast enough by oil. This leads to high wear and could result in total failure.

The pour point is usually lowered by adding certain additives called *pour point depressants*.

41.4 Variation of Lubricant Viscosity with Use

41.4.1 Oxidation Reactions

The viscosity of the oil increases when it is placed in service. This phenomenon is significant when the oil is subjected to excessive heat. Some of the oil molecules react with oxygen to form oxygenated molecules. This increases the oil viscosity and darkens its color.

In service, heat can result in a serious problem. Some lubricants contain antioxidant additives to control oxidative thickening.

41.4.2 Physical Reactions

The mineral oil is composed of a broad range of molecular types and sizes. The lower viscosity molecules tend to evaporate in hot environment, while the heavier, higher viscosity molecules remain. This results in an overall increase in viscosity.

41.5 Housing and Lubrication

Cleanliness has a major effect on the life of a rolling bearing. For this reason, These bearings must be protected from airborne particles and expected sources of dirt. Figure 41.3 illustrates typical bearing housing with various seals.

Interference fits for the rings must be tight to prevent relative motion. The bearings are usually mounted against a shoulder which makes it possible to move them off without transmitting the force through the rolling elements.

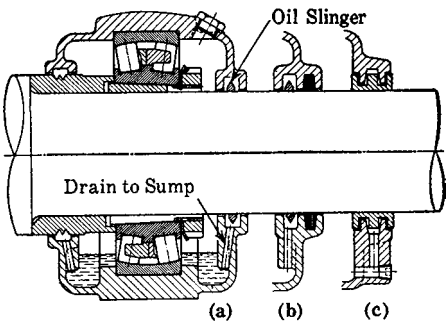
Grease is usually used at low and medium speeds because it provides a better seal against dirt. It is suitable for temperatures from -70 to 210°F. It is preferable not to pack tightly the bearing and the housing with grease. Two-third full is usually enough.

Oil is used at higher speeds. The oil level should be kept below the center of the lowest rolling element as shown in Fig. 41.3. Otherwise, excessive churning of the oil at high speed will cause overheating.

For a particular method of lubrication, the coefficient of friction is almost constant. Hence, the temperature rise under constant load is almost proportional to the speed.

Some bearings are cooled by circulating and cooling the oil due to excessive generation of heat. Some prelubricated bearings having integral seals to run for the life of the

FIGURE 41.3 Typical housing, with various seals. Sometimes, an oil slinger only is sufficient, as at (a). At (b), a slinger and a seal; (c) labyrinth seal. (Courtesy of SKF Industries, Inc. Philadelphia, Penn.)



bearing (or machine) without any required maintenance. Figure 41.4 illustrates a typical ball bearing with seals. The seals are usually on one or both sides.

Sealed prelubricated bearings require attention if they are not operated for a long period of time. The shaft should be rotated occasionally (once every three months) to move the lubricant and maintain a film coating of the balls.

The self-aligning ball bearings have been operated satisfactorily at high speeds. They also have a long life and very minimal thrust capacity.

The self-aligning spherical roller bearings are used for larger shafts and for applications having considerable thrust component.

The single-row deep-groove ball bearing is used most commonly on centrifugal pumps having a relatively small diameter. It is capable of handling radial as well as axial loads. This type of bearing has sometimes seals to exclude dirt and retain lubricant.

The angular-contact bearing is good for heavy thrust loads. The single-row is good for thrust in one direction only (commonly used on vertical pumps). The double-row type is capable of handling thrust in both directions (Fig. 41.5).

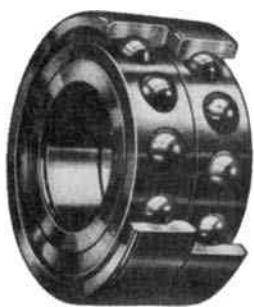


FIGURE 41.4 Ball bearing with seals. This type is relubricated by means of a hypodermic-like needle through the small holes on the side. (Courtesy of New Departure, Bristol, Conn.)

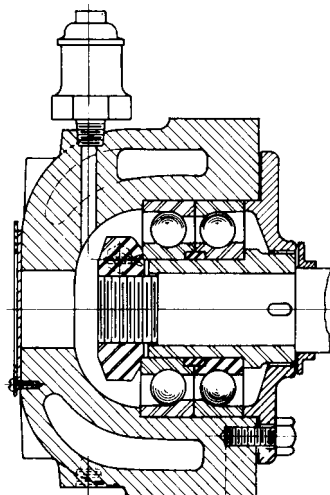


FIGURE 41.5 Double-row angular-contact ball thrust bearing that is grease-lubricated and water-cooled.

41.6 Lubrication of Antifriction Bearings

In general, the choice of lubricant is dictated by the application and the cost. For example, water is the preferred lubricant for vertical wet-pit condenser circulating pumps (Fig. 41.6). If oil or grease was used and the lubricant leaked, the condenser operation might be seriously affected because the lubricant will coat the tubes. Most ball bearings used for centrifugal pumps are grease-lubricated bearings.

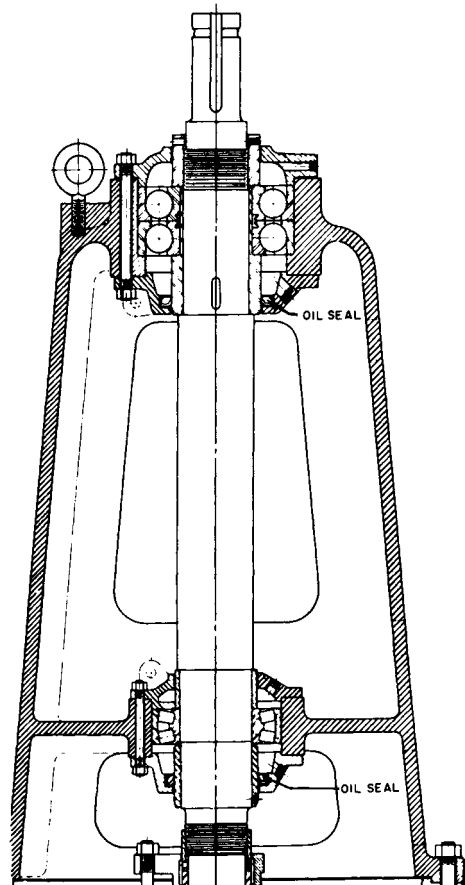
Figure 41.6 illustrates the seals required within the housing of some bearings to prevent the lubricant from escaping.

The proper grease circulation within the bearing and the housing is prevented if the bearing is overfilled with grease. Only one-third of the void spaces in the housing should be filled. Any additional amount of grease will cause the bearing to overheat and the grease will flow out of the seal to relieve the pressure. If the grease cannot escape through the seal or through the relief cock, the bearing will fail early.

The oil is supplied to the bearing from the bearing housing reservoir by oil rings as shown in Fig. 41.7. A constant-level oiler is used sometimes as shown in Fig. 41.8.

It is advantageous to have the capability to interchange the lubricant. Some pump lines are built with bearing housing that is capable of using oil or grease lubrication with minimal modifications (Fig. 41.9).

FIGURE 41.6 Ball bearing with seal in vertical pump. The seal guards against escape of grease.



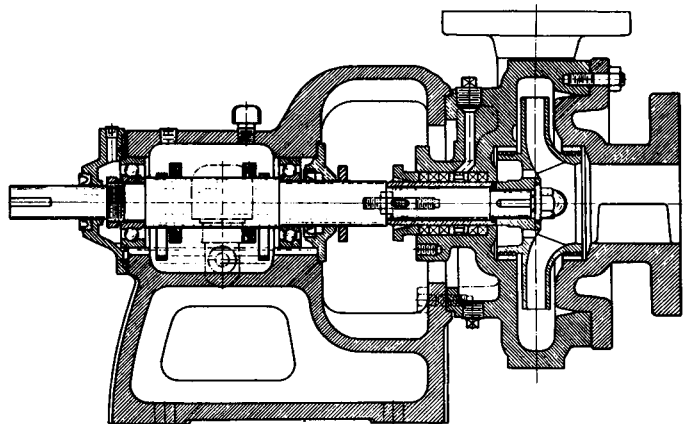


FIGURE 41.7 Ball bearing pump with oil rings.

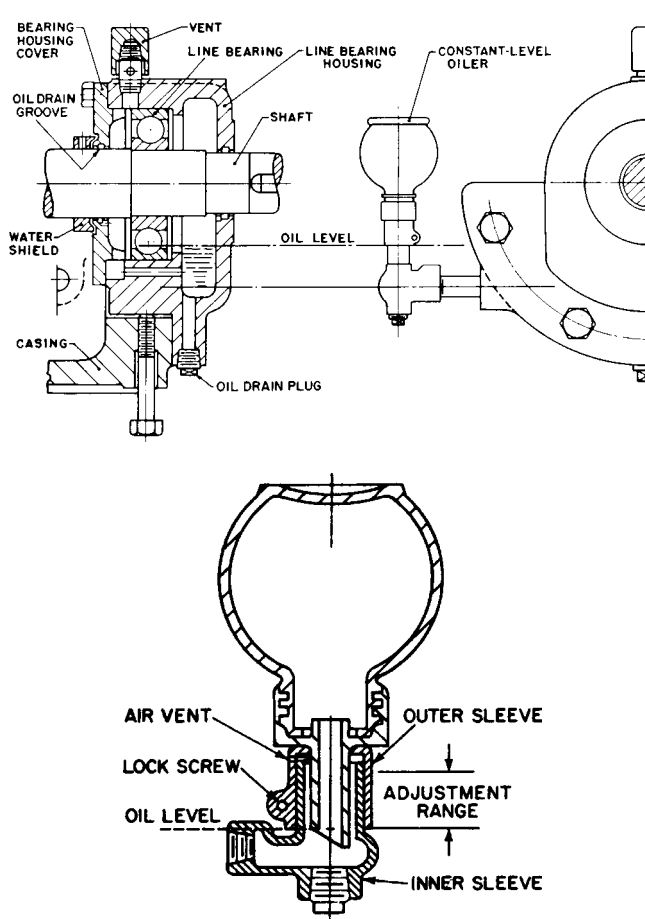


FIGURE 41.8 Constant-level oiler.

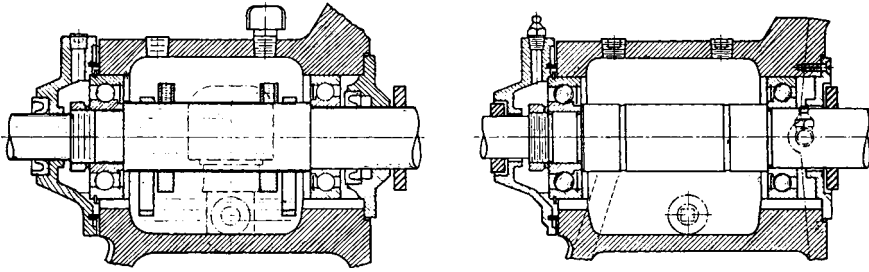


FIGURE 41.9 Ball bearings arranged (a) with oil rings in the housing and (b) for grease lubrication.

41.7 Bibliography

- Alexander, D. L., "The Viscosity of Lubricants," *Texaco Magazine Lubrication*, Vol. 78, 3, 1992.
- Faires, V. M., *Design of Machine Elements*, 4th ed. Collier-MacMillan, Toronto, 1965.
- Karassik, I. J., Krutzsch, W. C., Fraser, W. H., and Messina, J. P., *Pump Handbook*, 2nd ed. McGraw-Hill, New York, 1986.

This page intentionally left blank

Used Oil Analysis— A Vital Part of Maintenance

42.1 Proper Lube Oil Sampling Technique

All sample containers must be clean and dry. A used oil analysis program is started by sending a sample of the new, unused oil to the analysis laboratory. This sample will be used as a reference standard. The analysis is based on deviation of properties relative to the new, unused oil. The used oil sample should be collected when the oil is warm and well-mixed. If a machine is down, it should be started to circulate the oil through the system.

The sample line should be purged to remove stagnant oil and debris (the sample must not be contaminated). The sample should be taken from the line before the filter. The sample should be representative of the system. If the sample is taken from a sump or a gear box, it should be taken from the middle every time. Handpumps can be used to withdraw a sample from a sump or crankcase into a bottle. The sample must properly identify the oil and equipment type, collection date, time since last oil change, and oil makeup rate. The sample should be taken to the laboratory as soon as possible.

42.1.1 Test Description and Significance

Table 42.1 lists the test schedule for used industrial oils.

42.1.2 Visual and Sensory Inspections

An experienced observer can determine if the oil has contamination or deterioration. Engine oils in short service or those exhibiting little degradation will have a bland or an additive odor similar to that of unused oil. Oils with longer service periods under favorable operating conditions have a normal “used” odor.

In examination of compressor, turbine, and hydraulic systems, appearance and odor are particularly useful. These oils are bright, clear, and have bland odor under favorable operating conditions. The oil becomes “hazy” in the presence of a small amount of suspended water. The oil will have a cloudy appearance when it has a larger amount of water (Fig. 42.1). The reference line behind the oil to the left is sharply defined. This confirms that the oil is dry and clear. The oil is “hazy” in the middle bottle containing 200 parts per million (ppm). The reference line behind the oil to the right can hardly be seen because the oil is so cloudy (containing 400 ppm of water). Larger quantities of

Property	ASTM Method	Compressor Oil	Gear Oil	Turbine Oil	Hydraulic Oil
Appearance	—	X	X	X	X
Odor	—	(a)	X	X	X
Water					
(Crackle)	—	—	X	—	X
(Karl Fischer)	D 1744	(b)	X	(b)	(b)
Viscosity					
@40°C	D 445	X	—	X	X
@100°C	D 445	—	X	—	—
Toluene insolubles	D 893	—	X	—	—
Emission spectrometry	—	X	X	X	X
Infrared spectrometry	—	X	—	X	X
Total acid number	D 664	(c)	(c)	(c)	(c)
Particle count	F 661	(d)	—	(d)	(d)

- (a) Use caution when examining oils from ammonia or other noxious gas systems.
- (b) Determine if sample is hazy or if water content is critical.
- (c) Determine when sensory or infrared methods indicate need.
- (d) Determine if cleanliness is a major criterion or to meet equipment manufacturer recommendations.

TABLE 42.1 Basic Test Schedule for Used Industrial Oils

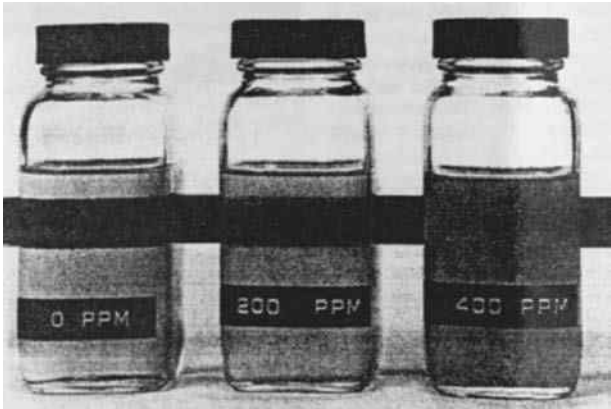


FIGURE 42.1 Various degree of water contamination in a clear petroleum product. (Source: Reprinted with permission from Chevron.)

water coalesce and form free water droplets which will settle at the bottom of the container. When contaminants increase in oil, its water separation properties gradually deteriorate. When there is contamination or oxidation in the oil, its color becomes significantly darker. The oil has characteristic odors under these conditions. Severe oxidation is characterized by a sharp or burnt odor. Some chemical contaminants

have specific odors typical of fuel, chlorinated solvents, or sour gas (i.e., a gas containing sulphur compounds). An oil analyst confirms these conditions by requesting additional tests.

42.1.3 Chemical and Physical Tests

Water Content

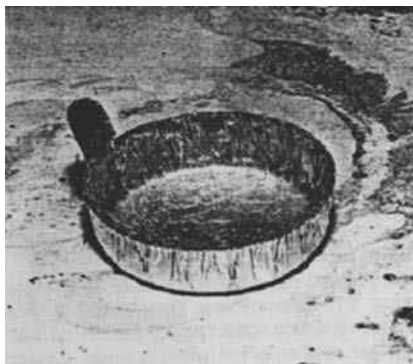
The presence of water in oil is confirmed by a crackle test. A few drops of oil are placed in a small aluminum dish and heated rapidly over a small flame or on a laboratory hot plate (Fig. 42.2). The test can be conducted by using a hot electric soldering iron which is immersed in oil (Fig. 42.3). If the water concentration is 0.01% or less, an audible crackling sound will be heard. When a crackle test confirms the presence of water, a quantitative test is conducted by distillation (ASTM D95) shown in Fig. 42.4. The test involves dissolving a specific quantity of oil in a water-immersible hydrocarbon solvent such as xylene, and heated in a distillation flask. The water present in the oil distills solvent. A condenser is used to cool the vapors. The water settles in the bottom of the trap where its volume can be measured while the solvent flows back to the distillation flask.

Figure 42.3 illustrates an automated instrument used for the Karl Fischer method (ASTM D1744). The advantage of this method is the ability of determining the amounts of free and dissolved water content in oil. However, it is normally used for relatively clean oils. This method is especially useful for refrigeration and hydraulic system lubricants which cannot have water content due to the potential damage to internal components. This is essential in refrigeration applications where free water droplets can freeze and plug the lubricant lines. The presence of water confirms lubricant contamination through leaking seals, coolant seepage, improper storage, or incorrect application of the lubricant. The water can cause rust, sludge, and deterioration in the lubricant properties. The source of water should be identified and eliminated as soon as possible.

Viscosity

Viscosity is a measurement of the lubricant's resistance to flow at a specific temperature. It is the most important property of the lubricant. The flow characteristics of the lubricant must be adequate to ensure that all the parts of the system are properly

FIGURE 42.2 Crackle test for water using a small aluminum in a hot plate. (Source: Reprinted with permission from Chevron.)



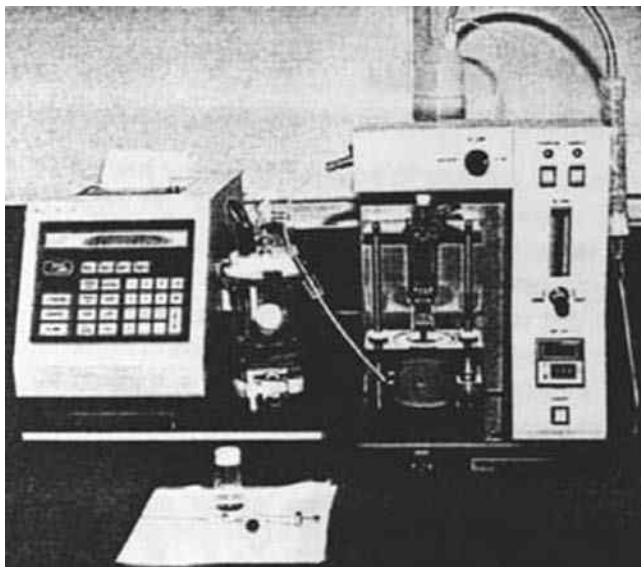


FIGURE 42.3 An automated apparatus for determination of water in petroleum products using the Karl Fischer method. (Source: Reprinted with permission from Chevron.)

lubricated. The viscosity varies with the type of the lubricant, degree of oxidation, and contamination in the lubricant. As the viscosity of the oil increases, it becomes harder to pipe it or take it off lubricated surfaces. The film-forming properties of the lubricant depend heavily on viscosity. However, if the lubricant's viscosity is very high, it cannot flow fast enough at low temperatures to perform its function. Unnecessary heat would be generated in this case, resulting in damage to the equipment. The lubricant should flow easily to perform its other functions such as carrying foreign material to system filters for removal, cooling, and transmitting force in hydraulic applications. The viscosity of the lubricant will increase with oxidation, foreign material contamination, degradation of additives, or evaporation of light base oil components. An increase of 10% in viscosity of the lubricant would be a cause of concern for an operating machine. A decrease in viscosity is expected with fuel dilution, and thermal or shear degradation of high-viscosity lubricant components such as viscosity index improvers. Sudden changes in viscosity provide indication of equipment malfunction or a change in operating conditions.

Emission Spectrographic Analysis

Emission spectrometers determine the presence of inorganic contaminants and metallo-organic oil additive elements. A film of oil is carried in this instrument on a rotating graphite wheel to a narrow gap between the wheel and a sharpened graphite rod electrode. The film is subjected to a high voltage arc. The energy of the arc excites the metallic elements in the oil. Each element emits a characteristic spectrum of light. The individual wear metal elements are characterized by selected lines of this spectrum. A diffraction grating is used to separate or isolate the selected lines permitting their light intensities to be measured by photomultiplier tubes. A computer is used to convert the

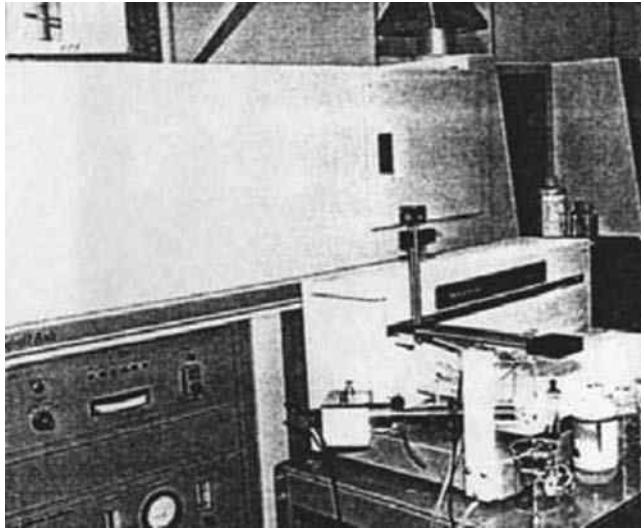


FIGURE 42.4 An inductively coupled plasma spectrometer. (Source: Reprinted with permission from Chevron.)

intensities of the light to element concentration. The results are printed for examination by the analyst.

Other types of spectrometers are inductively-coupled plasma (ICP) or direct-coupled plasma (DCP). Figures 42.4 and 42.5 show the advantages of these instruments is the ability of utilizing automatic samples for the production of the sample into the instrument.

Emission spectrographic analysis is a powerful tool for determining the levels of wear metal (trace amounts). The significance of wear metals in used oils depends on the

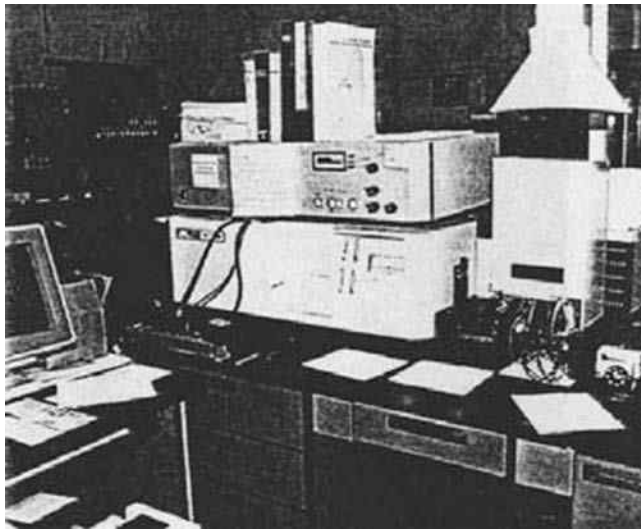


FIGURE 42.5 A direct-coupled plasma spectrometer. (Source: Reprinted with permission from Chevron.)

make and model of the equipment and type of service (including working environment, drain frequency, filter change interval, etc.). The concentration of metals increases with time. A sudden increase in concentration of metallic elements such as copper, lead, or iron indicates an increase in wear rate and possibly abnormal operating conditions. Immediate attention is required (maintenance or resampling are needed).

Trends should be developed over a period of time to determine normal operating conditions for wear metals. A judgment should not be made on a single sample analysis. The emission spectrograph provides the levels of metallo-organic additive in the used oil. This provides information about the depletion of additive, oil make-up, and suitability of the oil being used.

The following is the significance of common metals found:

- Copper: indicates wear in rocker arm bushings, wrist pin bushings, thrust washers, and other bronze or brass components.
- Iron: indicates wear from rings, liners, crankshaft, gears, pistons, cylinder walls, or valve train. It can appear as rust after storage.
- Lead: indicates wear from babbitt or copper/lead bearings.
- Tin: indicates bearing wear when babbitt overlays are used.
- Silicon: indicates the presence of sand, dust, or dirt entering the system. Major abrasion can occur causing wear in ring, liner, bearing, and crankshaft.
- Aluminum: indicates bearing or piston wear. Attention is needed even for a slight increase because rapid wear generates larger particles which may not get detected by emission spectrography.

Particles larger than 10 to 15 μm cannot be detected by a typical spectroscopy analysis. Particles smaller than 40 μm cannot be discerned by the naked eye. This is the reason for having emission spectrographic analysis done in conjunction with other analysis techniques such as ferrography (wear particle analysis) or vibration analysis.

Infrared Analysis

Infrared (IR) spectrometry is another powerful method for detecting low levels of organic contaminants, water, and oil degradation products. It provides a simple and rapid technique for establishing the following:

1. The general lubricant type (paraffinic or naphthenic)
2. The presence and often the quantity of certain contaminants such as alcohols, polar solvents, and free water (but not normally dissolved moisture)
3. The depletion of additives such as antioxidants
4. The presence of lubricant degradation products resulting from oxidation and nitration

Normally, a double-beam IR (DIR) spectrometer is used in the differential mode. The used oil is placed in the sampling cell of the instrument and an unused oil is placed in the reference cell. The instrument traces a curve representing the difference between the sample and the reference. It defines clearly the spectral regions characteristic of the organic contaminant or the oil degradation products. A newer technique for DIR spectrometry uses a Fourier transform infrared (FTIR) analyzer. It involves placing the sample in a

single-beam cell. The spectrum is compared with the one of the reference oils stored in the instrument computer. The method is particularly useful in analyzing oils from natural gas-fueled engines that have high combustion temperature. These engines promote the fixation of nitrogen by combining nitrogen and oxygen from air to form NO or NO₂ (termed NO_x), especially in naturally aspirated four-cycle gas engines when fuel/air ratios are lean.

The FT/IR technique calculates the “nitration number” which measures the nitrogen compounds in the oil resulting from fixation of nitrogen. A sudden increase in the nitration number relative to the “oxidation number” results in the formation of acidic materials. This increases the rate of oxidation and thickening of the oil, which results in the formation of varnish and sludge deposits. A buildup in the concentration of nitration is an indication that the engine needs tuning.

The “oxidation number” is also obtained from FT/IR. It is a measure of the amount of oxidation of the oil. This is an important parameter that determines the service life of a lubricating oil. The rate of oxidation increases significantly with high-temperature operation and contamination with water and glycol antifreeze. An increase in the level of oxidation (as measured by the oxidation number) requires attention to prevent significant increase in viscosity, organic acid formation, filter plugging sticky rings, or piston deposits and lacquering.

FT/IR spectroscopy also indicates contamination due to the presence of water, glycol antifreeze, fuel soot, or gasoline and diesel fuel, depending on the type of lubricant. When any of these contaminants is detected, the analyst will perform other tests to quantify the amount of contamination present.

Total Base Number

The total base number (TBN) indicates the total alkalinity of a lubricant. It determines the ability of the oil to neutralize harmful acidic by-products. The TBN is measured by perchloric acid titration (ASTM Method D2896) or by potentiometric titration with hydrochloric acid (ASTM Method D4739). The latter method replaced another potentiometric titration with hydrochloric acid (ASTM D664 for TBN) because it is more accurate and is in agreement with ASTM D2896.

The main use of TBN is for monitoring the remaining alkalinity of overbased detergent additives present in crankcase engine oils.

Total Acid Number

The total acid number (TAN) is measured by ASTM Method D664. It is used for industrial lubricants such as turbine, hydraulic, and refrigeration oils to monitor acid buildup in service. The TAN is used as a guide to track the oxidative degradation of an oil. When the TAN reaches a predetermined level, the oil should be replaced. A sudden increase in TAN indicates abnormal operating conditions that should be investigated.

Particle Count

Many systems such as hydraulic and turbine applications circulate lubricants through narrow passageways that must be kept clean and open to allow the lubricant to flow. These systems require fine filtration of the oil in service. The particle count analysis should be included in the used oil monitoring program to confirm the proper cleanliness of the lubricant for the service. The level of oil cleanliness for an application depends on its precision of operation, required reliability, and the system clearances. Figure 42.6 shows a modern-day particle counters. It uses a photometric cell to determine the classification of

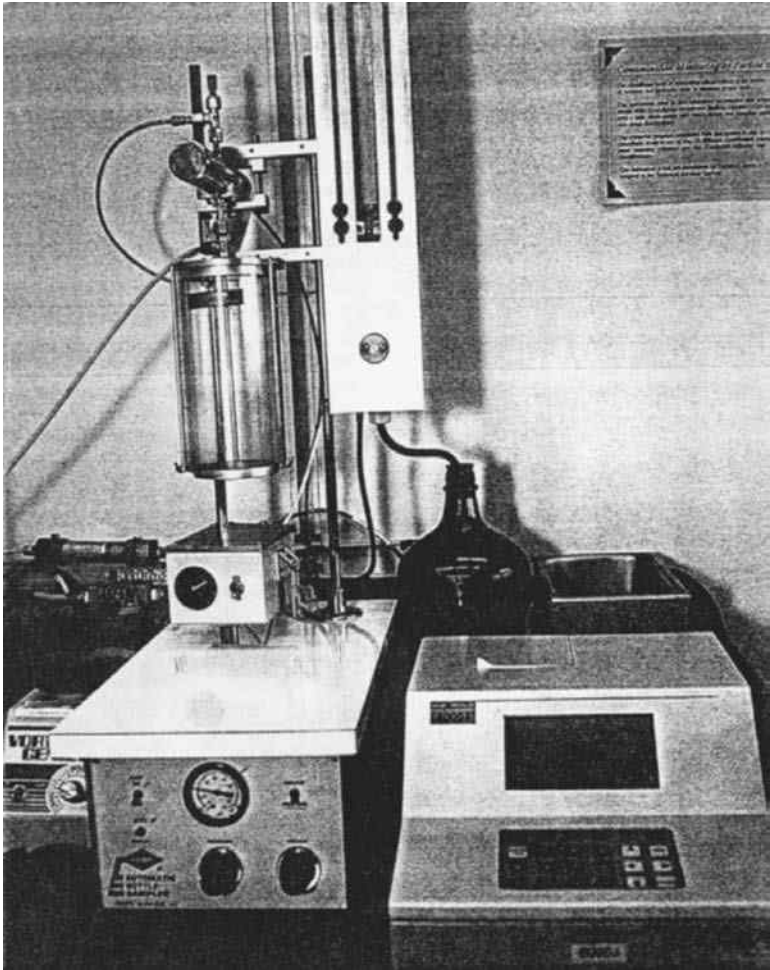


FIGURE 42.6 The modern particle counter uses a computer to classify cleanliness levels.
(Source: Reprinted with permission from Chevron.)

particles in an oil sample. The cleanliness of the lubricating oil is defined by particle size ranges of 5 to 10, 10 to 25, 25 to 50, 50 to 100, and over 100 μm . The particle count is measured in the size ranges of concern for the system being monitored. Proper sampling technique should be followed to avoid false results. The samples should also be mixed well prior to analysis.

The international standard method for classifying the size distribution of solid particles in the lubricant is the ISO Solid Contaminant Code. It typically reports particle size ranges of larger than 5 μm and 15 μm . Particles larger than 5 μm are generally caused by foreign contamination, while particles larger than 15 μm are wear particles in the system.

42.2 Summary

The used oil analysis program describes the condition of the oil in service. It is used to determine the suitability of the lubricant for continuing service and optimum intervals for preventive maintenance activities. Decisions regarding maintenance requirements and oil change intervals should be made, based on trends not on the results of one sample. The first step in the program is determining which equipment should be monitored and which tests will give the most useful information for the application. If improper sampling or testing procedures are used, the results can be misleading. The used oil analysis program is a useful technique if implemented properly. It has significant contribution to the overall success of the maintenance program.

42.3 Bibliography

Marshall, E., "Used Oil Analysis—A Vital Part of Maintenance," Texaco Technical Publication on lubrication. Vol. 79, No. 2, 1993.

This page intentionally left blank

Vibration Analysis

Vibration is defined as small oscillations about some equilibrium point. The main characteristics of vibration are amplitude and frequency.

Figure 43.1 illustrates the movement of a piston driven by a crankshaft moving up and down in a cylinder. The position of the center line of the piston above the center line of the cylinder (equilibrium position) is assumed positive. When the piston is at the top dead center, the piston center line will be at the maximum positive distance above the cylinder center line. The plot of the piston versus time is given by a sine curve. The period of the sine wave is 1 second (the frequency is 1 Hz). The maximum peak (or amplitude) of the motion is D .

The energy produced by the motion is related to the area under the curve (Fig. 43.1c). The RMS value of the amplitude is equal to $0.707 \times$ peak value of the amplitude. It represents the steady value of the amplitude that gives the same energy as the sine curve.

43.1 The Application of Sine Waves to Vibration

Any vibration occurring can be described by a sine wave. The crankshaft/piston arrangement shown in Fig. 43.1 can be described by the spring mass system shown in Fig. 43.2.

Figure 43.3 illustrates the variation of velocity with time. The velocity curve is actually a cosine wave. It leads the displacement curve by 90° . The maximum acceleration occurs at the top dead center. The largest force exerted on the connecting rod is the force required to change the direction of the piston. Figure 43.4 illustrates the variation of acceleration with time. The acceleration curve is 180° out of phase with the displacement curve and 90° out of phase with the velocity curve. The amplitude of displacement is proportional to the amplitude of acceleration (with a negative sign). Since the accelerometer sensors (measuring acceleration) are inexpensive, and easy to use, the amplitude of displacement is often in units of acceleration, known as "G's" of acceleration. It is convenient to express vibration acceleration in terms of gravitational acceleration "g."

$$\begin{aligned} \text{Acceleration (g)} &= \frac{\text{Acceleration (in/s}^2\text{)}}{386} \\ &= \frac{\text{Acceleration (mm/s}^2\text{)}}{9815} \end{aligned}$$

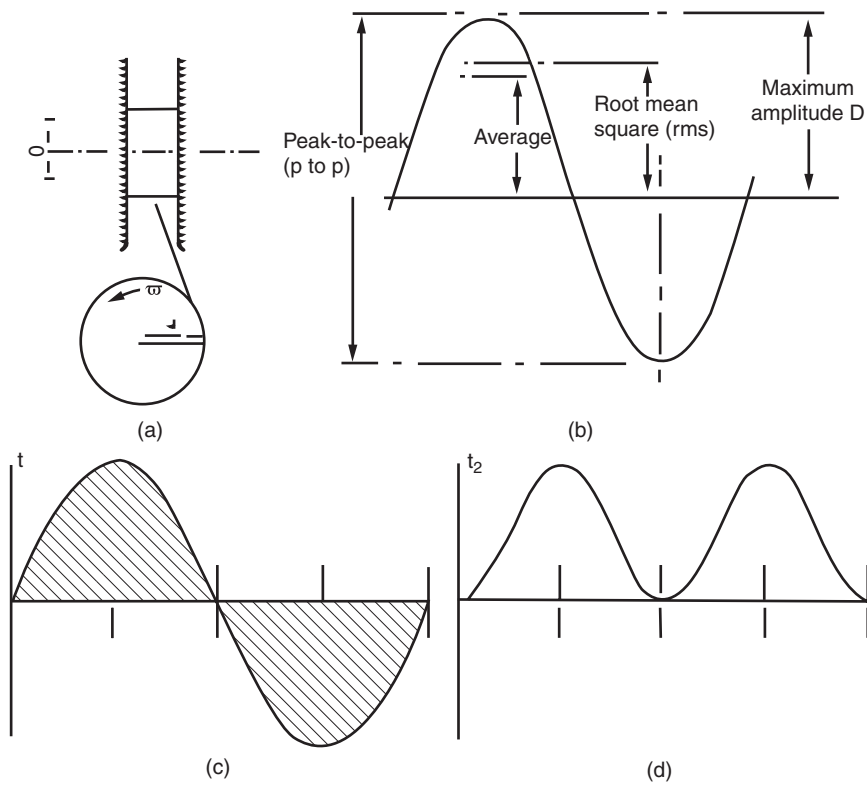


FIGURE 43.1 Vibration of a piston driven by a crankshaft.

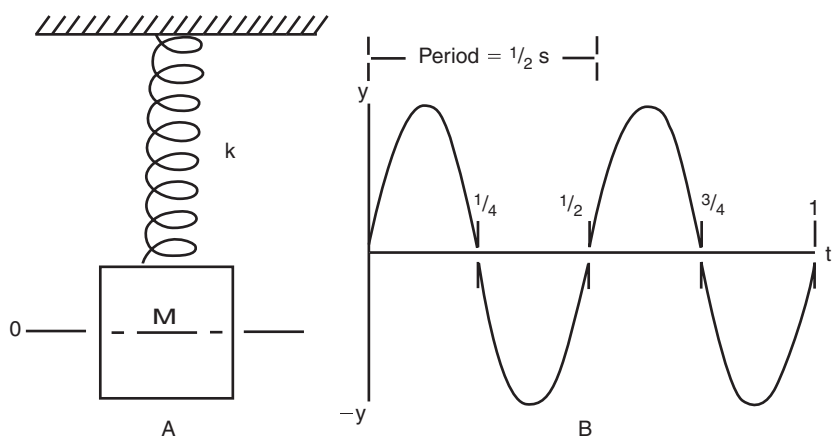
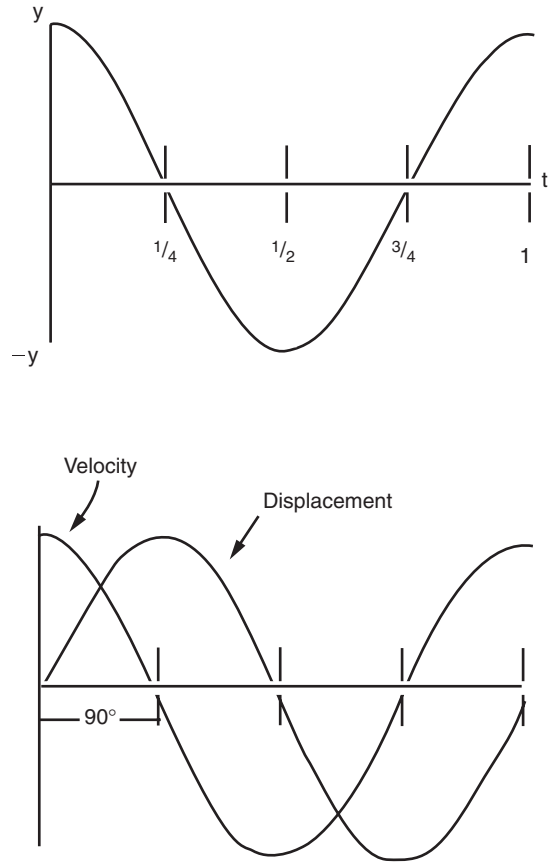


FIGURE 43.2 Vibration of a spring mass system.

FIGURE 43.3 Variation of vibration velocity with time.



43.1.1 Multimass Systems

Any real machine can be assumed to be made of many springs and masses that are vibrating at different frequencies. The total vibration level of the machine is the sum of the motions of all the parts of the machine (Fig. 43.5).

43.1.2 Resonance

Every system has one or more “natural frequencies” where it “likes to vibrate” at. If a system is excited by a small force at a natural frequency, it will exhibit very large magnitude of vibration. This wild vibration at the system natural frequency is known as “resonance.” A system is more likely to fail when it is in resonance due to the high internal forces imposed on the springs by the masses.

The values of the natural frequencies of a system can be calculated if its masses, spring constants, and damping constants are known. By applying a set of known forcing frequencies to the system, the natural frequency can be measured by determining the frequencies at which the system exhibits large magnitudes of vibration (a two channel real-time spectrum analyzer is usually used to obtain these measurements). The natural frequencies of the *machine-support-piping* system should be determined when analyzing the vibration of a machine. This is necessary to ensure the forcing frequencies are not near the natural frequencies.

FIGURE 43.4 Variation of vibration acceleration with time.

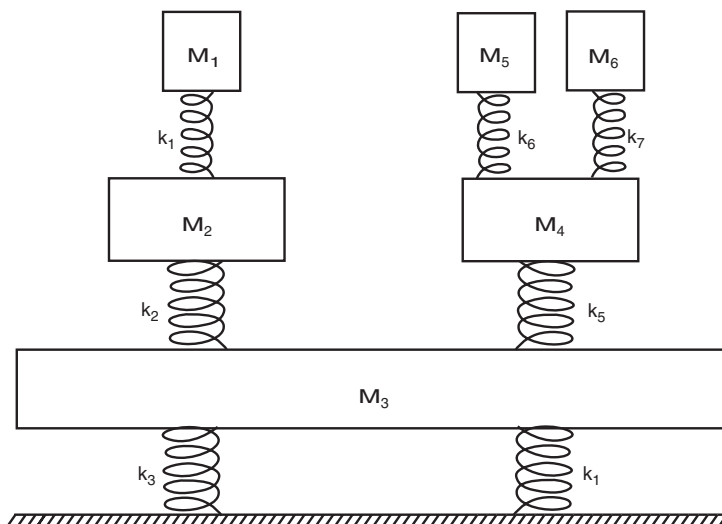
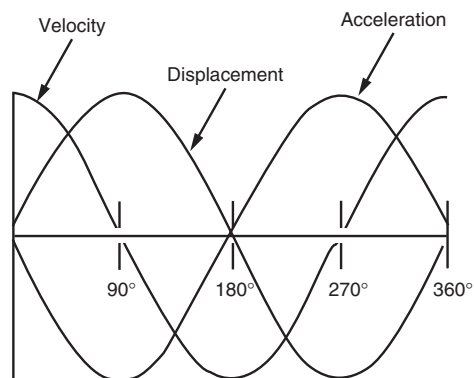
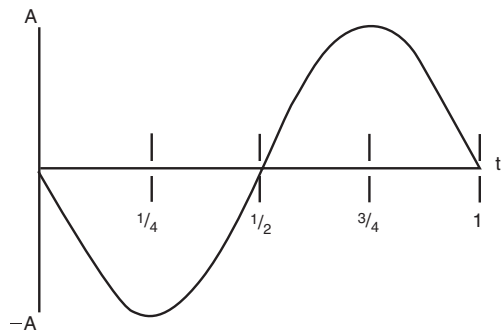


FIGURE 43.5 A real machine is assumed to be made of many springs and masses vibrating at different frequencies.

43.1.3 Logarithms and Decibels

A logarithmic scale is often used for vibration velocity and acceleration values due to the very wide range of vibration levels that are encountered with industrial machinery.

Since the measurements of both sound and vibration vary from very low to very high levels, the decibel (dB) is often used. The definition of decibel is:

$$\text{Decibel dB} = 20 \log \frac{\text{amplitude}}{\text{ref. amplitude}}$$

The standard reference for vibration is 10^{-3} cm/s^2 .

For sound, the standard reference is a pressure of 0.0002 microbar. By definition, doubling the amplitude is equal to an increase of 6 dB. Two machines which each generate 100 dB when operating alone, if they operate in phase, they would generate a maximum total noise of 106 dB (not 200 dB).

43.1.4 The Use of Filtering

Filters are used to observe only the frequencies which are of immediate interest.

43.1.5 Vibration Instrumentation

The most common transducers, in vibration monitoring are

- Displacement transducer
- Velocity transducer
- Acceleration transducer

Displacement Transducer (Proximity Probe)

The displacement transducer is commonly known as proximity probe. An electromagnet is placed in the tip of the proximity probe. It generates an electrical eddy current proportional to the gap between the probe and the metallic object to be measured.

The proximity probe (unlike velocity and acceleration transducers) must be mounted on a stationary support relative to the observed object. The proximity probes are useful for monitoring shaft orbits because they do not have to contact the moving surfaces.

Velocity Transducer

The velocity transducer is a movable permanent magnetic core with an enclosed coil. The relative motion between the magnet and the coil creates a voltage change which is proportional to velocity. Figure 43.6 illustrates two velocity transducers. The displacement can be obtained by integrating the output of the transducer.

Acceleration Transducer

Most accelerometers use the characteristic of piezoelectric crystals to measure acceleration. By applying a force to two faces of the crystal, a voltage is generated by the crystal proportional to the force. Figure 43.7 illustrates a piezoelectric accelerometer. The acceleration levels can be integrated to obtain velocity and integrated again to obtain displacement. Accelerometers are used at higher frequencies more than velocity transducers or proximity probes.

FIGURE 43.6 Velocity transducers. (a) LVT (contracting); (b) seismic (mounted on object to be measured).

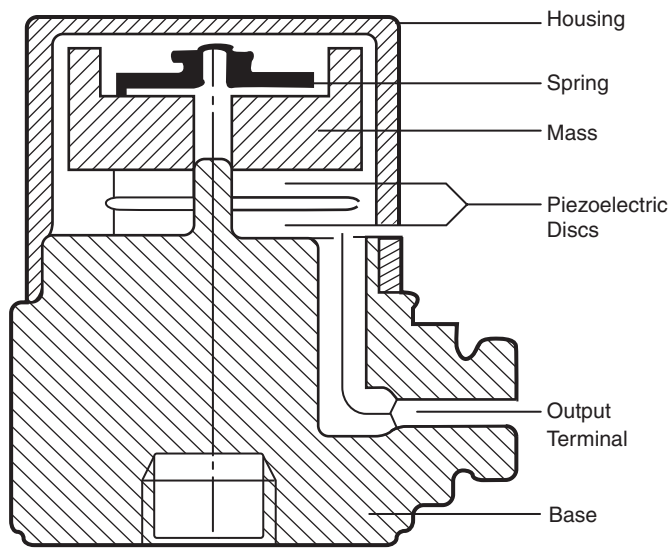
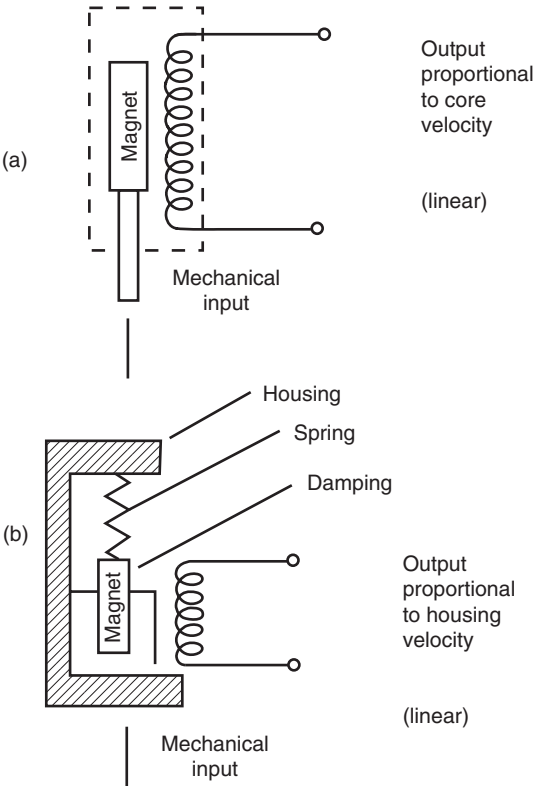


FIGURE 43.7 Piezoelectric accelerometer.

43.1.6 Transducer Selection

The amount of energy dissipated by a vibrating machine is proportional to the displacement, velocity, and acceleration. They all must be considered. If the machine is exhibiting high acceleration at the high frequency, the associated displacement is probably *very low*. If low acceleration was observed at low frequency, the associated displacement could be *very high*.

Figure 43.8 illustrates the relationship between acceleration, velocity, and displacement measurements. If the velocity of the vibrating machine remains constant over a frequency range, the acceleration will increase with frequency whereas the displacement will decrease with frequency. Therefore, the accelerometer would indicate large values at high frequencies and very low values at low frequencies. A displacement transducer would indicate large values at low frequencies and very low values at high frequencies.

In general, displacement transducers are useful zero to several hundred hertz (10,000 rpm). Velocity transducers are useful from 5 to 2000 Hz (300 to 120,000 rpm). Accelerometers are useful from 1 Hz (60 rpm) to more than 50 kHz (3,000,000 rpm).

The accelerometer is the most versatile transducer available. However, it may not have adequate sensitivity at low frequencies. Also, the accelerometer may be over sensitive at high frequency.

These problems can be corrected by selecting the proper transducers and by the addition of external filters to eliminate the high-frequency response of the accelerometer and enhance the sensitivity at low frequency.

Time Domain

Figure 43.9 shows vibration due to an unbalanced rotor in the time domain. The amplitude of vibration is proportional to the amount of imbalance and the cycle repeats itself once per revolution. Analysis in the time domain becomes more difficult when there is more than one vibration component present.

Figure 43.10 illustrates a situation where two sine wave frequencies are present. The individual components of this combination are difficult to derive from a time domain display. The impulsive signal from bearings and gear defects can be detected in the time

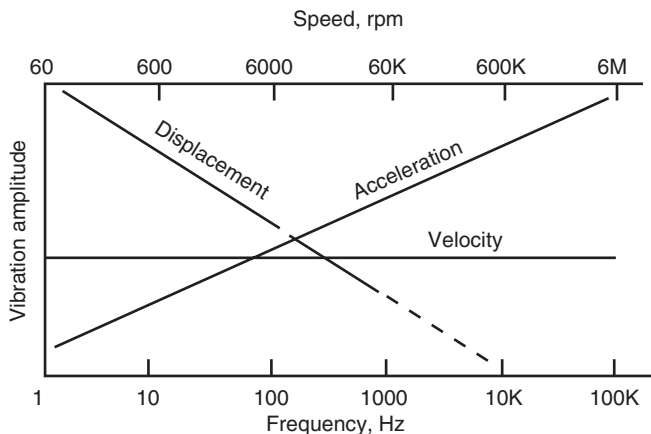


FIGURE 43.8 Transducer selection guide (useful ranges).

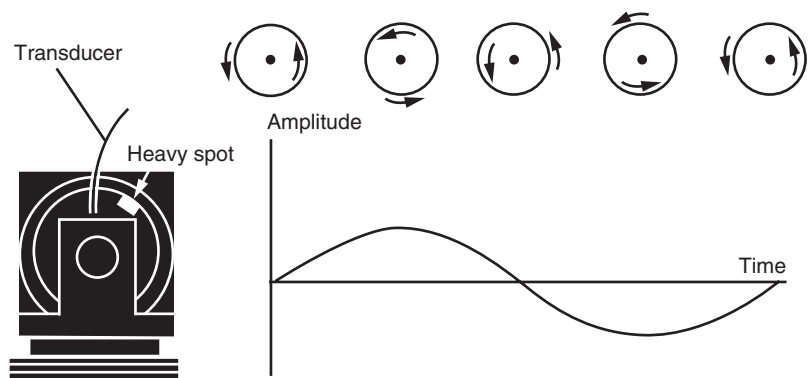


FIGURE 43.9 Vibration due to an unbalanced rotor in the time domain.

domain. It is also useful for analyzing the phase relationships of the vibration signals. However, it is difficult to determine the individual components of complex signals. The frequency domain is used to analyze these components.

Frequency Domain

Figure 43.11 illustrates a three-dimensional graph of the signal shown in Fig. 43.10. The frequency domain permits separation of the components in the wave form.

The frequency domain illustrates the same time domain that was shown in Fig. 43.10. It is the summation of two sign waves that cannot be recognized in the time domain.

Each frequency component appears as a vertical line in the frequency domain. The height represents the amplitude of the vibration. The representation of the signal in the frequency domain is called the *spectrum* of the signal. The frequency domain is powerful because any real signal can be generated by adding up sign waves (Fourier’s law). Therefore, the frequency domain can separate the sign wave components of any vibration signal.

The frequency spectrum of a vibration signal defines the vibration completely. There is no loss of information by converting to the frequency domain provided that the phase information is included.

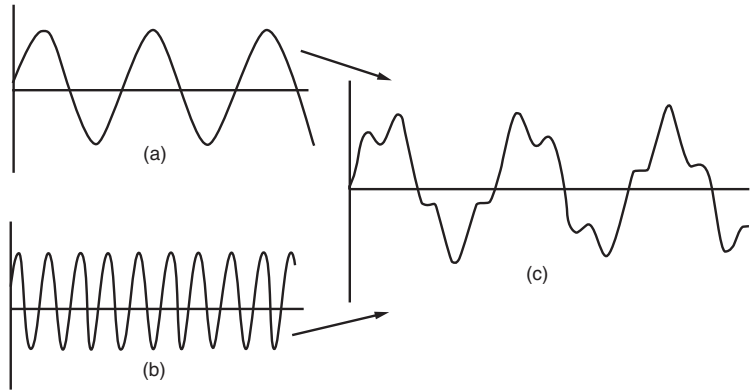


FIGURE 43.10 Vibration resulting from two different causes.

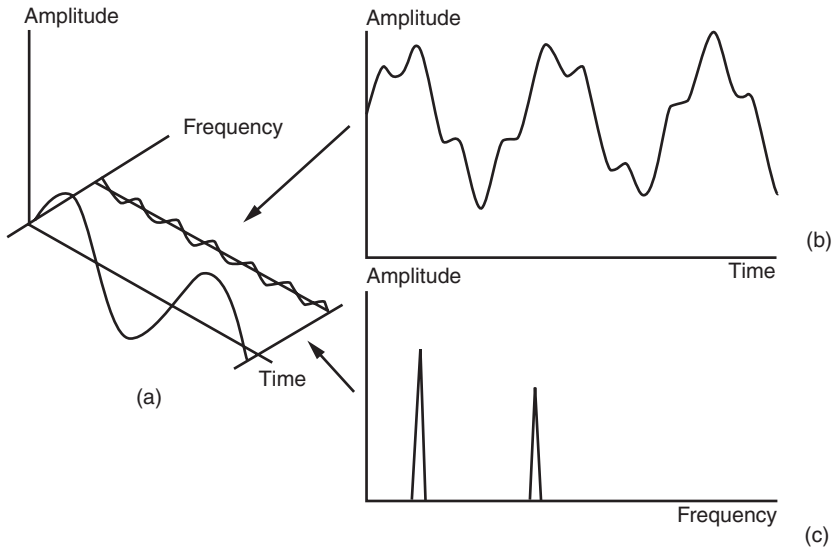


FIGURE 43.11 Vibration in the frequency domain.

43.1.7 Machinery Example

Figure 43.12 illustrates the usefulness of the frequency domain analysis for a vibrating machine. The internal sources of vibration are rotor imbalance, a ball bearing defect, and reduction gear meshing.

43.1.8 Vibration Analysis

The first step in vibration analysis is the understanding of the machinery (expensive diagnostic equipment is useless if the machine is not well understood). A study of the machine dynamics should precede taking vibration readings.

43.1.9 Vibration Causes

A wide variety of faults that results from wear, damage, or poor installation may lead to vibration in machinery. Vibration may also be caused by an inherent and unavoidable mechanical features such as the reaction of turbine blades passing an opening. However, serious vibration problems occur when one of the forcing frequencies approaches one of the natural frequencies of vibration. This is known as resonance. It amplifies the effect of the forcing frequency.

43.1.10 Forcing Frequency Causes

The first step in performing vibration analyses is to investigate the forcing frequencies which result from any of the following causes:

Unbalance

Unbalance is characterized by sinusoidal vibration at the machine's running speed ($1\times$). It occurs when the center of mass of a component does not coincide with its center of rotation.

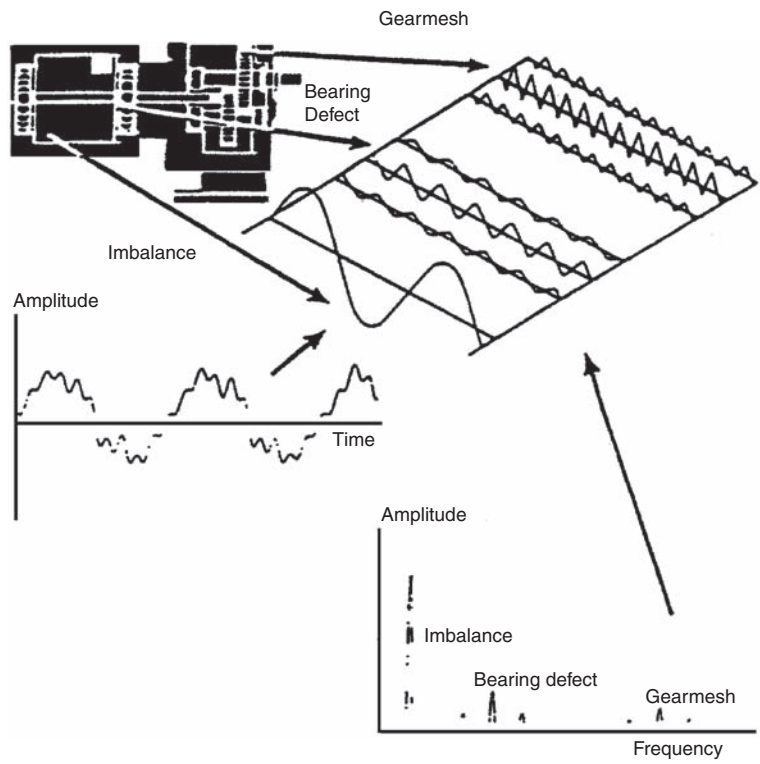


FIGURE 43.12 Identification of the vibration causes using frequency domain analysis.

The majority of vibration problems are caused by unbalance. The features of vibration from unbalance are

1. It is a single-frequency vibration which has a constant amplitude in all radial directions.
2. It increases with speed and does not contain harmonics (higher-frequency vibrations).

Misalignment

Vibration is generated when there is a misalignment between shafts of connected machines. It can be mistaken for unbalance. Its distinguishing feature is that its principle frequency is at $1, 2, 3 \times$ shaft rpm. Also, it contains a large axial component. Special cases of misalignment are improperly seated bearings and bent shafts.

Mechanical Looseness

Mechanical looseness vibrations at $1 \times$ rpm. However, it always contains higher harmonics. Mechanical looseness can often be located by taking several velocity readings at different points on the machine. The measured vibrations will be the highest in the vicinity and direction of looseness.

Bearing Defects

The most common cause of failures in small machinery is due to rolling-element bearings. The vibration is usually caused by a defect in the inner race, outer race, or rolling elements. The vibrations are characterized by high frequency and low energy.

Formulaes are used to determine the exact frequency generated by a fault in a race if the dimensions of the inner and outer races and the rolling elements are known.

Gear Defects

High-frequency vibrations are also generated by gear-meshing. They are usually low-energy signals that are easy to recognize but difficult to interpret.

The vibration frequency is equal to the rotational speed of the gear times the number of teeth. The vibration amplitude varies with the load.

Oil Whirl

Oil whirl in bearings is caused by instability of the rotor which is supported by the liquid film. Changes in oil viscosity, pressure, or external preloads produce conditions that prevents the film from supporting the shaft and bearing wipe occurs.

Blade or Vane Problems

Blade or vane passing generates single-frequency vibration or a large number of harmonies close to the blade passing frequency. The amplitude varies considerably with load. A cracked or missing blade usually increases the number of harmonics without changing the fundamental frequency.

Electric Motor Defects

Motor vibrations are caused by mechanical or electrical defects. The frequency generated by a broken rotor bar (mechanical defect) is distinguished from an electrical defect by the amount of motor slip.

A simple technique is used to differentiate mechanical from electrical problems in motors. If the vibration remains unchanged when the motor is turned off, the deficiency is mechanical in nature.

Uneven Loading

Vibrations are generated by uneven loading (such as a belt drive on an eccentric pulley). The vibration is generated at running speed. It could be mistaken for unbalance.

The vibration is distinguished by being unidirectional and usually varies with the load.

Drive Shaft Torsion

Torsional vibration is becoming a problem with the advent of variable-frequency drives, which may contain many electrical harmonics. Torsional vibrations do not generate externally measurable effect on the machine. They cannot be detected by conventional vibration sensors.

Resonant Frequency

If the running speed is close to one of the natural frequencies, the small amount of unbalance causes unacceptable vibration. Balancing the machine would not eliminate the vibration (because unbalance will return). The solution in this case is not to remove the forcing frequency, but to change the vibration characteristics of the machine.

This can be done by any of the following:

1. Increase the stiffness of the machine to raise the natural frequency.
2. Add mass to lower the natural frequency.

3. Add damping such as shock-absorbers to reduce the machine’s response to the forcing frequency.

If the forcing frequency is below resonance, damping is the most effective way to reduce the response. If the forcing frequency is higher than resonance, the mass should be increased.

43.1.11 Vibration Severity

Figure 43.13 illustrates the severity of vibration depending on displacement, frequency, and velocity.

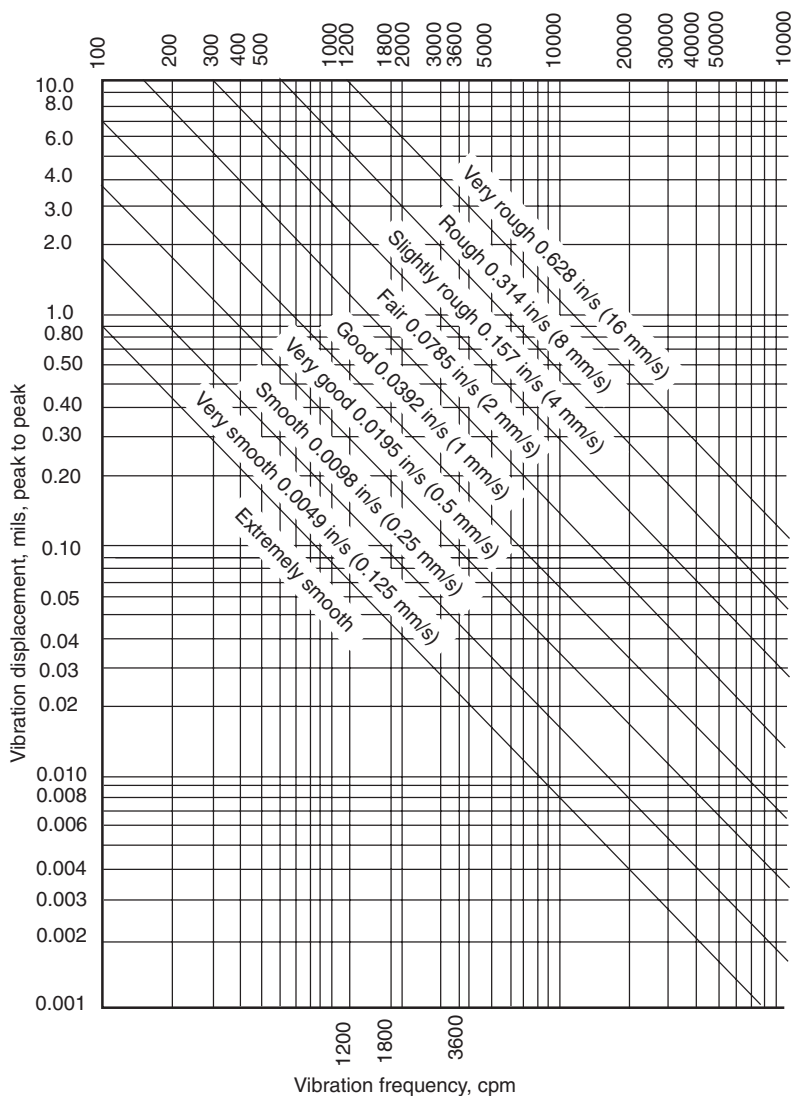


FIGURE 43.13 General machinery. Vibration severity chart. (Values shown are for filtered readings taken on machine structure or bearing cap. Vibration velocity, in/s peak.)

43.2 Appendix: A Case History (Condensate Pump Misalignment)

43.2.1 Problem

Machinery diagnostic program baseline data was collected for a power station's condensate pump. The data showed that levels recorded at the inboard pump axial and horizontal positions exceeded alarm values defined for these points.

43.2.2 Test Data and Observations

Velocity spectra from the inboard pump bearing showed the majority of the vibration was at $2 \times \text{rpm}$ in the axial direction and at $1 \times \text{rpm}$ in the horizontal direction. Misalignment between the motor and pump was suspected, and a coupling inspection was performed. The coupling was found to be out of alignment, as the following readings indicate: parallel side to side was out .009 in; parallel top to bottom was out .018 in; and the motor was .018 in; lower than the pump. Two of the four upper bearing housing bolts were also discovered to be loose.

43.2.3 Corrective Actions

The coupling was realigned to the following tolerances: .002 in parallel side to side and .006 in parallel top to bottom. The motor also was raised to a position .002 in lower than the pump. New bolts were installed on the upper inboard pump bearing housing.

43.2.4 Final Results

A new set of baseline vibration spectra was collected for this equipment at a later date. The data showed alignment greatly reduced the vibration levels. Inboard pump axial vibration dropped from 0.42 in/s to .09 in/s. The inboard pump horizontal vibration likewise was reduced from .18 in/s to .12 in/s. Similar reductions were observed at other motor and pump measurement points.

43.2.5 Conclusion

Had this misalignment condition continued unabated, it is very likely that the pump and motor bearings would have been damaged. This, in turn, could have led to a catastrophic failure of the condensate pump, consequently derating the unit. It took 2 men 12 hours to align the coupling. If the bearing needed replacement, it would have taken 3 or 4 men about 48 hours to complete the job. In summary, correcting this condition before the bearings or coupling were damaged saved the station many hours of manpower, and the expenses of new parts and lost production.

Cause	Amplitude	Frequency	Phase	Remarks
Unbalance (worn or lost parts, bent shaft)	Proportional to unbalance; largest in radial direction	1 × rpm	Single reference mark	One of the most common causes of vibration; appears after maintenance or after long periods of use
Bent shaft	Steady	1 × rpm; 2 × rpm also if bent at coupling	180° out phase axially	
Thermal bow	Varies during operation	1 × rpm	Single, reference mark	Increasing vibrations during variations in load or during start-up from cold condition
Misalignment of couplings or bearings	Large in axial direction; 50% or more of radial vibration	1 × rpm usual; 2 and 3 × rpm sometimes	Single, double, or triple	One of the most common causes of vibration; best found by appearance of large axial vibration; appears after maintenance or after long periods of use
Mechanical looseness		1, 2, and 3 × rpm	Two reference marks; slightly erratic	
Soft foot	Variable with load	1 × rpm; sometimes 2 × rpm		Check mountings for variations in vibration amplitude
Bed bearings of antifriction type	Unsteady—use velocity, or acceleration, or spike energy measurements, if possible	Very high; several times rpm	Erratic	Bearing responsible; most likely the one nearest point of largest high-frequency vibration
Eccentric journals	Usually not large	1 × rpm	Single mark	If on gears, largest vibration in line with gear centers; if on motor or generator, vibration disappears when power is turned off; if pump or blower, attempt to balance

Electrical	Disappears when power is turned off	1 × rpm or 1 or 2 × line frequency	Single or rotating double mark	If vibration amplitude drops off instantly when power is turned off, cause is electrical
Rubs		0 to 50% of 1×, 1×, higher harmonics	Erratic	Many frequencies over entire frequency spectrum
Oil or steam whirl	May change rapidly	40 to 50% of 1 × rpm	Unsteady	Small changes in radial steam forces, or bearing loading may result in large vibration changes; may excite critical
Aerodynamic hydraulic forces		1 × rpm or number of blades on fan or impeller × rpm and harmonics		
Bed gears or gear noise	Low—use velocity or acceleration measure if possible	Very high; number of gear teeth times rpm	Erratic	
Structural resonance	Steady or variable	1 × rpm but often odd nonsynchronous frequency	Erratic	May be flow excited in vertical pump; if problem starts on old machine, check for soft feet or cracked supports
Bad drive belts	Erratic or pulsing	1, 2, 3, and 4 × rpm of belts	One or two depending on frequency; usually unsteady	Strobe light best tool to freeze faulty belt
Reciprocating forces		1, 2, and higher orders × rpm		Inherent in reciprocating machines; can only be reduced by design changes of isolation
Cracked shaft	Variable during transients	1, 2 × rpm	Change in 1 × phase often occurs	2 × excitation of critical during coastdown; <i>note</i> other faults can also cause change in 2 × rpm

TABLE 43.1 Vibration Diagnostics Chart

This page intentionally left blank

Index

Note: Page numbers referencing figures are followed by an *f*; page numbers referencing tables are followed by a *t*.

A

Abradable seals:

- advantages, 393, 394, 395
- centrifugal compressor, 393, 393*f*, 394
- rubbed, 393*f*

Absorption chilling system, 123

ACC. *See* Advanced combined cycles

Acceleration transducers:

- accelerometers overview, 695
- piezoelectric accelerometer, 695, 696*f*

Advanced combined cycles (ACC), 11

Aeroderivative engines, 1

Aeroderivative gas turbine engine, 419, 420*f*

Aerosol:

- distribution in gases, 361, 362*f*
- formation in gases, 361, 362*f*

Air chambers, 613–614

Air compressors. *See also* Reciprocating air compressor maintenance

- brake horsepower input, 480*t*
- delivery variation with inlet air temperature, 481, 483*t*

Air distribution system:

- central, 481, 482*f*
- selection, 481
- unit, 481, 482*f*

Air filters, 289*f*

- air suction lines and, 289
- considerations and maintenance, 289

Air intake screens, compressor, 235

Air receivers, 233

- location and capacity, 289–290
- overview, 289–290
- pump-up time calculation, 483–484

Air receivers (*Cont.*):

- sizes, 290
- sizing calculation, 482–483

Amagat's law, 180

American Petroleum Institute (API):

- Standard 610, 622
- Standard 619, 326

Amonton's law, 180

Anti-surge control system, 408. *See also* Surge prevention systems

API. *See* American Petroleum Institute

ASME Boiler and Vessel Code, 622, 625–626

Availability factor, 46

Avogadro's Law, 181

Axial compressors. *See also* Axial flow compressors

- gas turbine, 419, 420
- inlet overhung compressor with thrust-reducing seals and magnetic bearings, 465–466, 466*f*, 467*f*
- types, 220*f*

Axial flow compressors, 232*f*. *See also* Dynamic compressors

- with adjustable stator blades performance maps, 371*f*

aircraft test bed, 368*f*

applications, 369–370

compressor auxiliaries, 419

enthalpy-entropy diagram for, 419, 421*f*

horizontally split, 217–218, 218*f*

overview, 232, 368–370

performance maps with speed variation, 370*f*

typical, 369*f*

typical performance maps, 370*f*, 371*f*

typical rotor, 419, 421*f*

Axial-flow pumps:
 propeller pump, 489, 490f
 specific speed upper limits of mixed-flow
 and, 647f

Axial thrust. *See* Impeller axial thrust

B

Balance line, 467
 Balancing disks, 502–503, 503f
 Balancing drums, 501–502, 502f, 503f
 Ball bearings. *See also* Bearings
 double-row, 666, 676, 677f
 double-row angular-contact thrust, 676, 677f
 filling-slot-type, 666, 668f
 grease lubrication, 679, 679f
 with oil rings in housing, 679f
 overview, 663–664
 pump with oil rings, 678f
 with seal in vertical pump, 677f
 with seals, 676f, 677f
 self-aligning, 663, 668f
 single-row, deep-groove, 664f
 single-row with shield, 668f
 thrust, 670f, 676, 677f
 various series of, 668f
 Barrier water, 334
 BDFM. *See* Brushless Double-Fed Machine
 Bearings. *See also* Ball bearings; Steam turbine
 bearings
 angular contact, 666, 668f
 antifriction, 663, 677f, 678f, 679, 679f
 bearing defects as forcing frequency cause,
 701
 cap data, velocity unfiltered, 601
 centrifugal pump, 504, 505f
 crankshaft, 302
 cylindrical roller, 666, 669f
 design life for rolling bearings, 665f
 diaphragm compressor, 302
 inboard, 663
 journal bearings, 34f, 50, 58, 59, 59f, 425, 663
 life statistical nature, 665, 665t
 line, 663
 materials and finish, 666
 needle, 666, 669f
 outboard, 663
 overview, 663
 preloading, 666
 radial, 332f, 663
 reciprocating compressor, 259–260

Bearings (*Cont.*):

rigid, 663
 roller, 663–664, 666, 667, 669f
 rolling bearing types, 664f, 666–667,
 668f, 669f
 rolling bearings dimensions, 667f
 rolling contact, 663, 664
 rolling contact stresses, 664–665, 664f
 rotary screw compressor, 332, 332f
 self-adjusting multi-segment thrust
 bearings, 332, 332f
 self-aligned spherical roller, 505
 self-aligning, 663, 666, 668f
 sizes, 666, 667t, 668f
 sleeve, 332, 663–664
 spherangular roller, 667, 669f
 straight lobe compressor, 353
 tapered roller, 667, 669f, 670f
 tapered thrust, 670f
 thrust, 34f, 50, 58, 59f, 332, 332f, 425, 663,
 669–670, 670f, 676, 677f
 tilted-pad journal, 34f, 50, 58, 59, 59f
 types, 663–664, 664f, 666–667, 668f, 669f
 vibration analysis of, 590, 590t, 591t, 592t,
 595t, 599, 601
 Bearing housing. *See also* Seal housing system
 lubrication and, 675–676, 676f, 677f, 678f,
 679, 679f
 typical, 675, 676f
 using oil or grease lubrication, 679, 679f
 Bedplates. *See* Pump bedplates
 Bellows compressors, 227, 227f, 228f
 Bellows-type metering pumps:
 operating limits and applications, 569t
 overview, 567, 568f, 569–570
 Bidding and negotiation. *See* Pump bidding
 and negotiation
 Black start, 7–8
 Boilers. *See also* Fire-tube boilers; Once-through
 boilers; Steam generators; Water-tube
 boilers
 closed feedwater heaters, 106, 106f
 condensers, 106
 controlled- or forced-circulation, 72
 heat exchangers, 99–103, 101t, 102f, 103f,
 104f, 105f
 heat transfer, 93–99, 94f, 95f, 96f, 99f
 natural-circulation, 72
 shell-type, 64
 Boiling point, 185

- Boundary layer:
 description, 415
 separation, 416–417, 416f
 wake, 417
- Boyle's law (isothermal law), 180
- Brushless Double-Fed Machine (BDFM), 167
 alternative referred per-phase equivalent circuit, 170, 170f
 conclusions, 174
 converter rating, 172–173, 173f
 core model, 170, 170f
 equivalent circuit, 168–170, 168f, 169f
 equivalent circuit definition of parameters, 170, 170f
 equivalent circuit parameter extraction, 171
 generator operation, 171–172, 171f, 172f
 ideal model, 170, 170f
 inverter ratings, 172–173, 173f
 machine control, 174
 overview, 167
 reactive power, 171–172, 171f, 172f, 173
 referred per-phase equivalent circuit, 169, 169f
 synchronous mode of operation, 168, 168f
 system configuration, 168–169, 168f
- Bushing seals:
 liquid, 446–448, 448f
 restricted, 449–450, 450f
- C**
- Cam bell diagram:
 critical speed map as, 42
 for turbine-generator vibration analysis, 42
- Cam pumps:
 cross-section of, 567f
 overview, 564
 with plunger valve, 564, 567f
- Canned motor pumps:
 with built-in valve for servo-thrust control, 580f
 design and applications, 579–582, 580f
 overview, 579–582, 580f
- Capital, working, 141
- Capital cost, of combined-cycle and cogeneration plants, 128, 134f
- Capital Cost Allowance (CCA):
 Canadian Class 43.1 write-down, 125
 Class 34, 125
- Cavitation. *See also* Pump cavitation
 overview, 508
 pitting, 509
- CC. *See* Combined-cycle
- CCA. *See* Capital Cost Allowance
- Centrifugal compressors, 212f, 231f. *See also*
 Dynamic compressors; Multistage centrifugal compressors
 applications, 381
 cleaning, 413, 415
 compressor drive, 375–376
 gas turbine, 419
 head capacity curve, 375f, 381
 instrumentation, 413, 414f, 415f
 liquid entrainment, 412–413
 nonoverloading characteristics of, 373
 operation principles of, 210, 212–213
 overview, 220f, 231–232, 365, 368, 381
 power requirement calculations, 475–476
 prime movers, 375
 rotor balancing, 406–407, 407f
 rotor dynamics, 406
 single-stage integral gear high-speed, 213, 215f
 single-stage (low-ratio) overhung turbo, 213, 214f
 single-stage with integral step-up gearing, 365, 366f
 static pressure tap, 413, 415f
 test setup, 414f
 two-phase with central double-flow rotor, 430, 432f
 vertically or radially split (barrel) with one compression phase, 430, 433f
 vertically or radially split (barrel) with two compression phases, 430, 433f
- Centrifugal compressor calculations:
 compression ratio *vs.* number of impellers, 377, 377f
 for discharge temperature, 376, 379
 equations, 376–379, 377f
 for head, 376, 377, 378
 hydraulic efficiency, 378
 for operating speed, 376
 power, 378–379
 for shaft horsepower, 376, 377
- Centrifugal compressor casing configurations:
 back-to-back design, 387f
 basic single-stage compressor, 383f
 double-flow compressor, 385f
 fabricated compressor casing, 382f
 high-speed single-stage compressor, 384f
 multistage compressor, 387f

Centrifugal compressor casing configurations

(Cont.):

- other, 386f
- overview, 381
- straight-through multistage compressor
 - with balancing piston, 385f

Centrifugal compressor classification types:

- with bell casings, 433, 434f
- horizontally split casing, 213, 215f, 365, 367f, 430, 431f, 432f
- pipeline, 433, 434f
- SR, 434–435, 435f
- vertically or radially or split (barrel) casings, 214, 216f, 365, 367f, 368, 430, 433f

Centrifugal compressor construction features:

- balance piston seal (center seal), 395, 395f
- coating effects on ethylene feed gas compressor, 390f
- collector nozzle, 382, 387f
- compeller thrust schematic, 395f
- compressor coatings, 389–390, 390f
- cooling and intercooling, 384f, 386f
- diaphragms, 383–384, 388–391, 389f, 390f
- discharge nozzle, 382, 387f
- flow path of gas from tip to return channel, 388f
- impeller, 381, 382, 383, 387f, 388f, 393, 394, 394f
- impeller thrust, 396
- inlet guide vanes, 381, 382–383, 387f, 388f
- inlet nozzle, 381, 382, 387f, 388f
- interstage seals, 391–395, 391f, 392f, 393f, 394f
- labyrinth seals, 391–392, 391f, 393–394, 395
- overview of major elements, 381–383, 387f
- radial diffuser, 381, 383, 387f, 388f
- return channel, 382, 383, 387f, 388f
- side stream nozzles, 386f
- wash system, 390–391
- water injection, 389–391

Centrifugal compressor diaphragms:

- diaphragm bundle, 425
- diffuser walls, 383, 384, 388
- discharge volute, 383, 389f
- overview, 383–384, 388–391, 389f, 427, 428, 428f
- polymer buildup, 384, 388
- return bend, 383
- return channel, 383

Centrifugal compressor head curve slope:

- backward leaning blade profile, 398, 398f, 399, 399f
- basic head curve shapes, 398–399, 398f
- flow rate change effect on impeller, 397, 398f
- forward leaning blade profile, 398, 398f, 399, 399f
- gas velocity radial and tangential components, 397, 397f
- gas velocity relative to impeller blade, 397, 397f
- head curve aspects, 396, 396f
- overview, 398–399
- radial blade profile, 398, 398f, 399, 399f
- velocity/pressure development for radial inlet impeller, 396f

Centrifugal compressor interstage seals:

- abradable seals, 393, 393f, 394, 395
- clearance and impeller thrust, 393, 394f
- compressor leakages and, 394–395
- compressor tripping and, 392–393
- degradation of, 392
- labyrinth seals, 391–392, 391f, 392f, 393–394
- overview, 391–395
- polytropic efficiency and, 391, 392, 393, 394–395
- rotating fins, 393–394
- seal clearance, 394
- thrust loading and, 394

Centrifugal compressor off-design operation:

- equivalent tip speed and, 406
- inlet condition variation effects, 404, 404f
- overview, 404–406
- speed change effect on head curve, 405, 405f
- variable speed drive applications, 405, 405f
- volume ratio effects, 405, 405f

Centrifugal compressor performance:

- head curve slope, 396, 397–399, 397f, 398f, 399f
- off-design operation, 404–406, 404f, 405f
- overview, 396, 396f
- stonewall (choke), 396, 399–401, 400f
- surge, 396, 401–404, 401f, 402f, 403f

Centrifugal compressor stonewall (choke):

- boundary layer separation and, 401
- flow vectors for impeller design condition, 400f
- limitation in impeller throat, 400f
- overview, 399–401, 400f

Centrifugal compressor surge:

- damage of, 404
- diffuser vanes and, 402, 403f
- flow through diffuser, 401, 401f
- identification, 412
- overview, 396, 401–404, 403f
- prevention systems, 408–411, 409f, 410f
- rotating stall, 402, 402f

Centrifugal pumps:

- affinity laws, 643–644
- axially split casing horizontal-shaft double-suction volute pump, 492f
- bearings, 504, 505f
- bedplates, 504, 505f, 506f
- best operating speed selection, 648–649, 649t
- bottom-suction axially split casing single stage pump, 492f
- capacity and efficiency determination, 646–647, 647f, 648f
- casings and diffusers, 489, 492
- cavitation, 509
- composite rating chart, 659f
- couplings, 504
- data sheet, 624t
- double-casing multistage with radially split inner casing pump, 493f
- double-volute casing pump, 492f, 495f
- double volute of multistage pump, 495, 496f
- efficiency chart, 486, 488f
- four-stage pump with opposed impellers, 500, 501f
- horizontal single-stage double-suction volute pump, 493f
- hydraulic balancing devices, 501–504, 502f, 503f
- hydraulic balancing disks, 502–503, 503f
- hydraulic balancing drums, 501–502, 502f, 503f
- hydrostatic pressure tests, 495–496
- main parts, 486
- minimum flow requirement, 506
- multistage pump with single-suction impellers, 500, 501f
- operation theory, 487, 488f, 489, 489f
- overview, 485–486, 487f, 488f
- parts, 494t
- radial thrust, 492, 494f, 495, 495f, 496f
- rating table, 660f
- recommended maintenance, 510
- rotating element, 486

Centrifugal pumps (Cont.):

- selection using similarity or affinity laws, 644–646
 - single-stage end-suction volute pump, 488f
 - specific speed upper limits of, 642f
 - stationary element, 486
 - suction specific-speed ratings, 649t
 - typical, 485, 487f
 - uniform or near-uniform single-volute casing pressures, 492, 494f, 495, 495f
 - velocity variation, 488f, 489
 - vertical-shaft end-suction pump with double-volute casing, 490f
 - water equality, 504
- Centrifugal pump bowl section:
- of horizontal double-suction volute pump, 489, 493f
 - of single-stage axial-flow propeller pump, 489, 490f
 - of vertical dry-pit single-suction volute pump, 489, 490f
 - of vertical wet-pit diffuser pump, 490f
- Centrifugal pump classifications. *See also*
- Canned motor pumps
 - axially split, 489, 492f
 - bottom suction, 489, 492f
 - diffuser-type, 487, 489f
 - double suction, 489, 492f, 493f, 511f
 - end suction, 488f, 489, 490f, 491f
 - horizontal, 489, 492f, 493f, 505f
 - multistage (two or more impellers), 489, 493f, 500, 501f
 - overview, 489
 - radially split, 489, 493f
 - side suction, 489, 492f
 - single stage (one impeller), 488f, 489, 492f, 493f
 - top suction, 489, 493f
 - vertical, 489, 490f
 - volute, 487, 488f, 489, 490f, 492, 492f, 493f, 494f, 495, 495f, 496f
 - Centrifugal pump impellers, 496, 497f, 498f, 499f
 - axial thrust, 500, 500f, 501f
 - axial thrust in multistage pumps, 500, 501f
 - four-stage pump with opposed, 500, 501f
 - multistage (two or more impellers), 489, 493f, 500, 501f
 - multistage pump with single-suction, 500, 501f
 - shape and efficiency for pump speeds, 648f
 - single stage (one impeller), 488f, 489, 492f, 493f

- Centrifugal pump performance characteristics:
 - efficiency, 486, 488*f*, 506–507, 507*f*
 - head, 506
 - power, 506
 - problem, 506–507, 507*f*
- Centrifugal pump seal-less pump motors.
 - See Seal-less pump motors
- Centrifugal pump troubleshooting, 586
 - mechanical troubles, 587*t*
 - suction troubles, 587*t*
 - symptoms, 587*t*
 - system troubles, 587*t*
- Centrifugal separators, 359, 361*t*
- CFS. *See* Conventional filter separators
- Charles' law, 180
- Check valves. *See also* Diaphragm pump check valves
 - ball check valves, 574, 575*f*
 - diaphragm compressor, 305
 - flap check valves, 573–574, 573*f*, 574*f*
 - rapid-closing, 612
 - slow-closing, 612
 - water hammer and pumps with, 610–612
- CHF. *See* Critical heat flux
- Clearance pockets:
 - manual controlled variable-volume, 250*f*
 - manual fixed-clearance pocket valve, 248*f*
 - overview, 244–246, 249*f*
 - positive displacement compressor capacity control and, 244–246
 - reciprocating compressor capacity control and, 244–246
- Clearance volume, 222
- Coke fuel:
 - other coking processes, 348
 - overview, 348
 - properties and usage, 348
- Coke gas plants, water-injected two-stage screw compressors, 334–335, 335*f*, 336*f*, 337
- Combined-cycle (CC), 119
- Combined-cycle and cogeneration plants
 - economics:
 - base case, 127–128
 - Canadian CCO Class 43.1 write-down economic consideration, 140
 - capital cost, 128, 134*f*
 - combined-cycle configuration, 128, 129*t*, 130*f*, 131*f*, 132*f*, 133*f*
 - conclusions, 140–141
 - Combined-cycle and cogeneration plants
 - economics (*Cont.*):
 - definitions of terms used in tables for, 141
 - economic growth and, 126, 127*t*
 - electricity purchase rate, 140
 - evaluation of different combined-cycle configurations, 135–136, 135*t*–136*t*, 137*t*, 138, 138*t*, 139*t*, 140
 - financial analysis, 127
 - financial analysis of different plant configurations, 127–165
 - natural gas prices and, 125–126, 126*t*
 - operating and maintenance cost, 128–129, 134, 134*f*
 - overview, 125
 - reference plant analysis results, 137*t*
 - reference plant economic assumptions, 135*t*–136*t*
 - revised plant analysis results, 139*t*
 - technical and financial comparison of five combined-cycle plants, 138*t*
- Combined-cycle plants. *See also* Combined-cycle and cogeneration plants economics; Single-shaft combined-cycle plants; STAG combined-cycle plants
 - description and illustration, 119, 120*f*
 - environmental impact, 114–115
 - generation economics, 111
 - multi-shaft, 111
 - overview, 111–112
- Combined-cycle plants configurations, financial analysis:
 - income statements, 144*t*, 150*t*, 156*t*, 165*t*
 - project assumptions, 142*t*, 148*t*, 154*t*, 160*t*
 - project cash flows, 145*t*, 151*t*, 157*t*, 162*t*
 - project results, 143*t*, 149*t*, 155*t*, 161*t*
 - schedules, 146*t*–147*t*, 152*t*–153*t*, 158*t*–159*t*, 163*t*–164*t*
- Combined-cycle plants power enhancement options:
 - absorption chilling system, 123
 - inlet-air chilling option, 122–123, 122*f*
 - inlet-air pre-cooling option, 119–120
 - performance of all, 119, 121*t*
 - selection of best, 119–120, 120*f*, 121–124, 121*t*, 122*f*, 124*f*
 - steam and water injection options, 123–124, 124*f*
 - supplementary firing in HRSG option, 124

- Combined-cycle power systems, steam turbine selection for, 11–30
- Combined-cycle units. *See* Single-shaft combined-cycle units
- Combustion chambers (combustors), 4
- Compressed-air system:
 - air leakage annual cost, 474f
 - calculation of air leaks, 473–475
 - main and branch pipes, 474f
 - plant and compressor, 474f
- Compressibility:
 - correction, 195–196
 - gas mixture, 190, 191
 - generalized factor chart, 184, 184f
- Compression:
 - efficiency and cylinder clearance, 201
 - ratio, 175
 - stroke, 176, 176f
- Compression cycles:
 - isothermal compression, 191, 192
 - near-adiabatic (isentropic) compression, 191, 192
 - P-V diagram of theoretical, 191f
 - reciprocating compressor, 210
- Compression principles. *See* Gas laws and compression principles
- Compressors. *See also* Dynamic compressors; Positive displacement compressors
 - air intake screens, 235
 - components arrangement, 223f
 - definition, 219
 - filters, 235
 - fouling, 423–424
 - intercoolers and aftercoolers, 234–235
 - limitations of single-stage, 196
 - multiple-stage, 196–197, 197f
 - oil system problems, 453–454
 - P-V diagram for two-stage air, 196, 197f
 - polytropic efficiency, 193–194, 193f
 - power requirement, 194–195
 - preventive maintenance and housekeeping, 235–236
 - starting new, 290–291
 - volumetric flow rate, 197–198
 - water cooling requirements, 481, 482t
- Compressor air filters. *See* Air filters
- Compressor air receivers. *See* Air receivers
- Compressor applications:
 - positive displacement compressor, 204
 - ranges, 203, 204f, 205f, 206t
 - various, 206t
- Compressor categories. *See also* Dynamic compressors; Positive displacement compressors
 - classification of various types, 220f
 - overview, 203, 219, 220f
- Compressor control. *See also* Positive displacement compressor capacity control
 - constant speed control, 233
 - dual control, 233
 - overview, 233
 - start-stop control, 233
- Compressor cooling:
 - closed cooling water system, 271f
 - large compressor cylinder forced, 269, 270f
 - overview, 269, 270–272
 - reciprocating compressor, 269, 270f, 271–272, 271f, 272f
 - thermosyphon or static-filled cooling system, 271–272, 272f
 - typical cooling system, 271, 271f
- Compressor cylinders. *See* Cylinders
- Compressor foundation:
 - concrete, 288
 - considerations and maintenance, 287–289, 288f
 - proper, 288f
 - vibration and, 287, 289
- Compressor frames:
 - cast iron compressor frame assembly, 264, 264f
 - reciprocating compressor, 264, 264f
- Compressor maintenance. *See* Reciprocating air compressor maintenance
- Compressor off-design performance. *See also* Centrifugal compressor off-design operation
 - compressor range, 423
 - high rotational speeds, 422–423
 - local instabilities, 423
 - low rotational speeds, 422–423
 - overview, 419, 422, 422f
 - pressure ratio *vs.* mass flow rate ratio, 422, 422f
 - rotating stall, 422
- Compressor operation:
 - compression ratio, 175
 - compression stroke, 176, 176f
 - compressor cylinder full of gas, 175–176, 176f
 - expansion stroke, 177–178, 177f
 - gas laws and compression principles, 175–179, 176f, 177f, 178f, 219–220

- Compressor operation (*Cont.*):
 intake or suction stroke, 178, 178f
 overview, 219
 P-V diagram for two-stage compressor, 178–179, 178f
 piston completion of delivery stroke, 176–177, 177f
 reciprocating compressor, 175–176, 176f
- Compressor performance. *See also* Centrifugal compressor performance; Compressor off-design performance; Dynamic compressor performance; Positive displacement compressor performance
 energy available for recovery, 222–223, 223f
 inlet conditions, 222
 measurement, 220–222
 overview and typical characteristics of, 222
- Compressor pistons. *See* Pistons
- Compressor range, 423
- Compressor seals. *See also* Abradable seals; Bushing seals; Centrifugal compressor interstage seals; Labyrinth seals; Liquid seals
 carbon ring seals, 332, 333f
 contact seals, 448–449, 449f
 diaphragm compressor, 303, 307
 floating ring seals, 333f, 334
 gas seals, 445–446, 445f
 rotary screw compressor, 332, 333f, 334
 slide ring seals, 333f
 stationary single seal, 333f, 334
- Compressor seal systems:
 atmospheric draining system, 443
 contact seals, 448–449, 449f
 gas seals, 445–446, 445f
 liquid bushing seals, 446–448, 448f
 liquid seals, 446, 447f
 overview, 441, 442f
 restricted bushing seals, 449–450, 450f
 seal housing system, 442–443, 442f
 seal leakage system, 443–445, 445f
 seal liquid leakage system, 452
 seal oil system, 441, 442f
 seal supply systems, 450–452, 451f
 supply system, 441
- Compressor selection calculations, 476, 478–479
 air compressor brake horsepower input, 480t
 air system requirement calculations, 477t
 compressor drive selection, 479, 481
 pneumatic tools air needs, 478t
 reciprocating compressors characteristics, 481t
- Compressor service check chart:
 broken valves and springs, 300t
 control trouble, 300t
 high intercooler pressure, 299t
 high oil pressure, 298t
 incorrect delivery of mechanical lubricator, 298t
 incorrect operation of suction-valve unloaders, 300t
 knocks, 299t
 low intercooler pressure, 299t
 low oil pressure, 298t
 overheated cylinder, 299t
 reciprocating air compressor, 298t–300t
 scored cylinder, liner, or piston, 299t
 water in cylinders, 299t
- Compressor system calculations:
 air distribution system selection, 481, 482f
 air leakage annual cost, 474–475
 air receiver pump-up time, 483–484
 air receiver sizing, 482–483
 centrifugal compressor power requirement, 475–476
 components sizing, 482–484
 compressed-air system air leaks, 473–475, 474f
 compressor delivery variation with inlet air temperature, 481, 483t
 compressor drive selection, 479, 481
 compressor selection, 476, 477t, 478–479, 478t, 480t, 481t
 compressor water cooling requirements, 481, 482t
- Compressor unloading. *See* Unloading
- Compressor valves. *See also* Valves
 different designs of, 294f
 maintenance, 293–296
 oils, 294–295
 reciprocating compressor, 246–248, 250f, 274–275, 274f, 275f, 276f, 277, 277f, 278f, 279f, 280, 280f, 293–296, 294f
 reciprocating compressor performance and, 246–248, 250f
 troubleshooting, 295
- Condensers, 106
- Conduction:
 through cylindrical walls, 95–96, 95f
 definition, 93
 steady-state, 93–95, 94f

Connecting rods:
 component parts, 262f
 reciprocating compressor, 262–263, 262f
 typical, 262f

Constant-level oiler, 678f, 679

Convection:
 forced, 93
 heat transfer mode, 93

Convection heat-transfer coefficients:
 overview, 97
 streamlined forced-convection flow inside tubes, 98
 turbulent forced-convection flow across *N* onbaffled tube banks with circular tubes, 98, 99f
 turbulent forced-convection flow inside long circular tubes, 98

Conventional filter separators (CFS):
 filter quality, 345
 overview, 343, 343f
 removal efficiencies, 344

Cooling. *See also* Compressor cooling
 inlet-air pre-cooling, 119–120

Crankshaft design:
 optimization of unbalanced forces, 256, 259f
 reciprocating compressor, 256–257, 259f, 260f, 261f
 10-throw compressor, 256, 259f
 three-throw angles with dummy crossheads, 256, 260f
 three-throw crankshaft oil passages and counterweights, 256, 261f
 two-throw, 256, 260f

Critical heat flux (CHF), 72

Critical pressure, 187

Critical speed map, 42

Critical temperature, 187

Crossheads:
 components, 263f
 crankshaft design with dummy, 256, 260f
 reciprocating compressor, 263–264, 263f
 sliding crosshead piston compressors, 225–226, 225f

Cylinders:
 cast iron or nodular iron, 266f
 clearance and compression efficiency, 201
 clearance and volumetric efficiency, 198–201, 199f, 200f, 201f

Cylinders (*Cont.*):
 conventional and tandem, 268, 269f
 double-acting, 265, 265f, 268, 269
 fabricated carton or stainless steel, 266f
 forged steel, 266f, 267f
 full of gas, 175–176, 176f
 lubrication, 282
 manufacturing materials, 264
 non-lubricated, 293
 reciprocating compressor, 240, 264–265, 265f, 266f, 267f, 268–269, 268f, 269f, 270f, 282
 selection in reciprocating compressor sizing, 240
 single-acting, 268
 tandem, 265, 268f
 truncated or step, 265, 268f

Cylinder liners:
 clamped, 270f
 double-acting, 269
 materials, 269
 steel, 269

D

Dalton's law, 180

Degree of saturation, 187

Departure from nucleate boiling (DNB), 72

Depreciation, 141

Dew point, 185

Diaphragms. *See also* Centrifugal compressor diaphragms
 nozzle boxes and, 56, 57f

Diaphragm compressors:
 Burton Corblin two-stage, 313f, 314f
 cleaning and testing, 310–311
 construction materials, 308–309
 corrosive environments of, 306
 goal, 313
 installation and maintenance, 312–313, 313f, 314f
 internal pressures during compression, 303, 304f
 limitations, 312
 operating at different pressures, 313f, 314f
 overview, 226, 226f, 301
 piping, 309
 specification, 315
 theory of operation, 301–304, 304f
 U.S. standards, 315

Diaphragm compressor accessories:

- accumulators, 309
- gas bypass line, 310
- hydraulic system protection instrumentation, 310
- monitoring instrument, 309–310
- relief valve and rupture disc, 310
- screwed joints, 309
- socked welds, 309
- suction filter, 309
- variable speed drives, 310

Diaphragm compressor applications:

- automotive air bag filling, 312
- in different countries, 315
- overview, 311–312
- petrochemical industry, 312

Diaphragm compressor components:

- auxiliary lubricating pump, 305
- bolts, 307
- check valves, 305
- compensating circuit, 302
- compensating pump, 303, 305
- compression head, 301, 303
- connecting rod journals, 302
- crankcase, 301, 302
- crankshaft bearings, 302
- crosshead, 301
- crosshead sliding surfaces, 302
- diaphragm set, 303
- diaphragms, 307
- discharge valve, 303, 304
- eccentric, 305
- electric motor, 301, 305
- head assembly, 306, 306*f*
- head integrity or leak detection system, 307–308, 307*f*
- hydraulic oil, 302, 303
- hydraulic pressure limiter, 303, 304, 305
- hydraulic system, 301
- oil head, 303, 304*f*
- overview, 301, 302*f*
- piston rings, 301
- sealing elements, 308
- seals, 303, 307
- shaft-driven gear pump, 305
- suction valves, 303
- typical drive arrangement, 301
- valve retainer and bolting, 303
- valves, 308
- wrist pins, 302

Diaphragm compressor design:

- basic crankcase, 304
- configuration, 304–305
- foundation, 312–313
- model number, 304
- oil filtration, 305
- oil head and gas head maximum pressure rating, 304
- oil selection, 306
- strokes, 304

Diaphragm pumps. *See also* Bellows-type metering pumps; Mechanically driven diaphragm pumps; Pneumatically powered diaphragm pumps

- advantages, 577, 578
- construction materials, 576–577
- hydraulically actuated, 573
- liquid handled by, 577
- mechanically actuated, 560, 561*f*
- metering pump flow characteristics, 562*f*
- metering pumps, 560, 562*f*, 569–570, 569*t*
- operating limits and applications, 569*t*
- operation, 560, 561*f*
- overview, 555, 560, 571
- pump controls, 577
- pumping dry powders, 577

Diaphragm pump check valves:

- ball, 574, 575*f*, 576, 576*f*
- flap, 573–574, 573*f*, 574*f*, 576, 576*f*
- overview, 576, 576*f*
- poppet, 576, 576*f*

Diaphragm pump limitations:

- air-operated, 577
- overview, 577
- pressure limitations, 577

Dirty gas, 342

Displacement pumps, 485, 571. *See also* Positive displacement pumps

Displacement transducer (proximity probe), 695

Distance pieces:

- extended-length, 284, 284*f*
- overview, 282, 284, 286
- packings and, 284, 284*f*, 285*f*
- piston rods and, 284, 284*f*
- reciprocating compressor, 282, 284, 284*f*, 285*f*, 286
- standard, 282, 284*f*
- two-compartment, 284, 285*f*, 286

DNB. *See* Departure from nucleate boiling

- Double-feed generators, 167. *See also* Brushless Double-Fed Machine
- Dry air, 187
- Dry gas seal failure prevention by gas conditioning:
 need for training, 460
 overview, 459
 sealing problem risk minimization, 460–461
- Dry seals:
 assembly cartridge, 456, 456f
 background, 453–454
 bidirectional dry seal faces, 458, 458f
 dry gas seal failure prevention by gas conditioning, 459–461
 groove shape and seal leakage rates, 458–459, 458f
 monitor and control system, 456–457, 456f
 operating experience, 457
 operating principles, 454–457, 454f, 455f, 456f
 overview, 453
 problems and solution, 457
 rings, 454, 454f
 single dry seal configuration, 455f
 tandem dry seals, 455, 455f
 upgrade developments, 458–459, 458f
- Dry seal support system factors:
 commissioning procedures, 460
 control system design, 460–461
 gas composition, 460
 interface between compressor, seals and control system, 461
 plant specifications, 461
- Dynamic compressors:
 applications, 210, 231
 common characteristics, 231t
 overview, 203, 210, 220f, 231, 365
 principles of, 365
 technology, 365, 366f, 367f, 368–370, 368f, 369f, 370f, 371f
- Dynamic compressor performance, 231–232
- Dynamic compressor performance equations:
 for brake horsepower, 373
 centrifugal compressor calculations, 376–379, 377f
 centrifugal compressor head capacity curve, 375f
 for discharge temperature, 373
 head *versus* flow without speed change, 373, 375f
 for operating speed, 373
- Dynamic compressor performance equations (Cont.):
 for speedy change, 373, 374f, 375f
 for stability, 373
 for suction valve throttling, 375f
 for variable inlet guide, 374f
 for variable speed, 374f
- Dynamic compressor performance limitations:
 anti-surge control systems and, 438–440, 439f
 overview, 435, 435f
 pressure and flow during surge cycles and, 436, 437f
 single-parameter surge protection system and, 438, 439, 439f
 standard two parameter or biased surge protection system, 439, 440f
 stonewall, 438
 surge control line (SCL) and, 436f, 438
 surge control zone and, 436f
 surge limit, 435f, 436–438, 436f, 437f
 surge limit line (SLL) and, 436, 436f
 surge prevention and, 438–440, 439f
 typical performance curve and, 435, 435f
- Dynamic compressor surge:
 anti-surge control systems, 438–440, 439f
 control zone, 436f
 cycles, 436, 437f
 limit, 435f, 436–438, 436f, 437f
 prevention, 438–440, 439f
 protection system, 438, 439, 439f, 440f
- Dynamic compressor types, 220f. *See also* Axial flow compressors; Centrifugal compressors

E

- Economic growth:
 average annual growth projections
 1982–2020, 126, 127t
 combined-cycle and cogeneration plants
 economics and, 126
- EHC. *See* Electrohydraulic control
- EHC systems, maintenance, 60
- Elastohydraulic lubrication, 673
- Electric generator, 70-MW, 1
- Electrohydraulic control (EHC), 60
- Emission spectrographic analysis:
 common metals found using, 686
 spectrometers used in, 684–685, 685f
 of used oil, 684–686, 685f
- External irreversibility (losses), 193

F

- Feedwater heaters:
 - closed, 106
 - modern high-pressure, 106*f*
 - tube bundle for H.P., 105*f*
- Filter separators:
 - conventional filter separators, 343, 343*f*
 - filter quality, 344–345
 - filtration configuration evaluation, 346
 - life-cycle-cost calculations, 346–347
 - removal efficiencies, 344
 - selection of gas filtration equipment, 345
 - selection of modern reverse-flow, 343–347
 - self-cleaning coalescers, 343–344, 344*f*
 - total ownership cost, 347
- Filter separator considerations:
 - filter element cost, 346
 - flow velocity, 346
 - pressure drop, 346
 - vessel life, 346
- Filter vane separators, 359, 361*t*
- Filters, compressor, 235
- Financing cost, 141
- Finger-type unloaders (valve depressors), 242, 242*f*, 243*f*, 244
- Fire-tube boilers:
 - early horizontal-tube, 65*f*
 - fire box, 64
 - overview, 64, 66
 - scotch marine, 64, 66, 66*f*
 - stead steam generator, 65*f*
- Forced convection, 93
- Forced outage rate, 46
- Forcing frequency causes:
 - bearing defects, 701
 - blade or vane problems, 701
 - drive shaft torsion, 701
 - electric motor defects, 701
 - gear defects, 701
 - mechanical looseness, 700
 - misalignment, 700
 - oil whirl, 701
 - resonant frequency, 701–702
 - unbalance, 699–700
 - uneven loading, 701
 - vibration analysis of, 699–702
- Free (standard) air flow rate, 220
- Free power turbine, 1, 4

G

- Gas. *See* Natural gas prices; Non-perfect gases; Perfect gases
- Gas compressors. *See* Compressors
- Gas conditioning units (GCUs), 459
- Gas film technology, 454
- Gas generator, 1
 - Olympus, 2*f*
 - starting, 4
- Gas laws and compression principles:
 - compressibility correction, 195–196
 - compression cycles, 191–193, 192*f*
 - compressor operation, 175–179, 176*f*, 177*f*, 178*f*
 - compressor polytropic efficiency, 193–194, 193*f*
 - compressor power requirement, 194–195
 - compressor volumetric flow rate, 197–198
 - critical conditions, 187
 - cylinder clearance and compression efficiency, 201
 - cylinder clearance and volumetric efficiency, 198–201, 199*f*, 200*f*, 201*f*
 - gas mixtures, 187
 - gas mole, 187
 - gas mole percent, 188
 - gas partial pressure, 188–189
 - gas property relationships, 182, 183*t*, 184, 184*f*, 185*t*
 - gas specific gravity, 188
 - general gas law equation, 219–220
 - molecular weight of gas mixtures, 188, 188*t*
 - multiple staging, 196–197
 - overview, 175, 219–220
 - partial vapor pressure, 186–187
 - perfect (ideal) gas laws, 179–181, 181*t*
 - pseudo-critical conditions and compressibility, 190–191, 190*t*
 - specific heats, 189–190, 190*t*
 - symbols, 175, 201–202
 - thermodynamics first law, 179
 - thermodynamics second law, 179
 - vapor pressure, 185
 - volume percent of constituents, 188
 - weight-basis item, 191
- Gas/liquid separation. *See* Liquid/gas separation

Gas mixtures:
 gas properties weight-basis item, 191
 mole, 187
 mole percent, 188
 molecular weight of, 188, 188*t*
 overview, 187
 partial pressure, 188–189
 pseudo-critical conditions and compressibility, 190–191, 190*t*
 specific gravity, 188

Gas property relationships:
 non-perfect gas, 182, 184, 184*f*, 185*t*
 overview, 182, 183*t*, 184, 184*f*, 185*t*
 perfect gas, 182, 183*t*

Gas turbine (GT):
 applications, 1
 axial compressors, 419, 420
 centrifugal compressors, 419
 free power turbine, 1, 4
 natural-gas-fired, 120*f*
 protection, 6–7
 routine tests, 8–9
 shutdown, 6
 working cycle, 4, 5*f*

Gas turbine engine:
 aeroderivative, 419, 420*f*
 routine tests, 8–9

Gas turbine performance degradation:
 compressor fouling, 423–424
 non-recoverable, 424
 overview, 423
 recoverable, 423–424

Gas turbine plants. *See also* Combined-cycle plants
 current design of, 3*f*
 requirements, 1

Gas turbine starting, 4
 Black start, 7–8

Gas volume (velocity), 4

Gasification, 107. *See also* Integrated gasification combined cycles

GCUs. *See* Gas conditioning units

Gear pumps:
 external, 562–563, 563*f*
 operation, 563*f*
 overview, 562–563
 performance of external, 563*f*

Generalized compressibility factor chart, 184, 184*f*

Gravity separators (knockout drums), 359, 361*t*

Greases:
 ball bearing lubricated by, 679, 679*f*
 bearing housing lubrication by, 679, 679*f*
 as non-Newtonian fluids, 674
 viscosity, 674

GT. *See* Gas turbine

H

Halon, 6

Heat exchangers:
 film coefficients, 101*t*
 overview, 99–101
 possible temperature patterns, 102*f*
 shell-and-tube, 101–103, 103*f*, 104*f*, 105*f*

Heat recovery system generator (HRSG), 11

Heat transfer:
 boiling liquids and condensing vapors, 99
 combination heat-transfer effects, 96–97, 96*f*
 conduction through cylindrical walls, 95–96, 95*f*
 convection heat-transfer coefficients, 97–99, 99*f*
 steady-state conduction, 93–95, 94*f*
 thermal conductivities, 95

Heat transfer modes:
 conduction, 93
 convection, 93
 forced convection, 93
 natural circulation (thermosyphoning, natural convection), 93
 radiation, 93

High-efficiency coalescer, 359–360, 360*f*

High-speed shut-off cock (HSSOC), 6

High-velocity oxygen fuel (HVOF), 36

Housing. *See* Bearing housing; Seal housing system

HRSG. *See* Heat recovery system generator

HSSOC. *See* High-speed shut-off cock

HVOF. *See* High-velocity oxygen fuel

Hydraulic balancing devices. *See* Impeller axial thrust, hydraulic balancing devices

Hydraulic unbalance, 602

I

Ideal gases. *See* Perfect gases

IGCC. *See* Integrated gasification combined cycles

IGCC plant, 108*f*
 reliability, availability and maintenance, 110

- IGCC plant considerations:
 - output enhancement, 109
 - size of IGCC, 109
 - turnkey cost, 108
 - IGCC plant emission reduction:
 - air pollutants, 109
 - carbon dioxide, 109
 - mercury, 109
 - nitrogen oxides (NO_x), 109
 - overview, 109
 - IGCC processes:
 - combustion of syngas, 108
 - cryogenic air separation, 108
 - gasification, 107
 - syngas cleanup, 108
 - Impellers. *See also* Centrifugal pump
 - impellers
 - axial-flow, 498f
 - centrifugal compressor, 381, 382, 383, 387f, 388f, 393, 394, 394f
 - centrifugal compressor stonewall and, 400f
 - compression ratio *vs.* number of, 377, 377f
 - Francis-vane radial double-suction closed, 498f
 - multistage centrifugal compressor, 425, 427f, 428, 428f, 429, 430
 - open, 497f, 498f
 - open mixed-flow, 498f
 - open with partial shroud, 497f, 498f
 - profile variations, 499f
 - pump characteristics with varied diameter of, 660f
 - semiopen, 498f
 - straight-vane radial single-suction closed, 497f
 - unbalance, 601–602
 - Impeller axial thrust:
 - in multistage pumps, 500, 501f
 - overview, 500, 500f
 - Impeller axial thrust, hydraulic balancing devices:
 - balancing disks, 502–503, 503f
 - balancing drums, 501–502, 502f, 503f
 - overview, 501
 - Impeller thrust:
 - centrifugal compressor, 396
 - centrifugal compressor interstage seals clearance and, 393, 394f
 - Impeller vane-shape classifications:
 - Francis-vane or screw-vane, 496
 - mixed-flow, 496, 498f
 - propeller or axial-flow, 496, 498f
 - straight-vane, 496, 497f, 498f
 - Impulse wheel-and-diaphragm construction, 48
 - Infrared analysis:
 - infrared spectrometry, 686–687
 - of used oil, 686–687
 - Inlet-air chilling, 122–123, 122f
 - Inlet-air pre-cooling, 119–120
 - Instant power stations (IPS), 128
 - Integrated gasification combined cycles (IGCC), 107. *See also specific IGCC topics*
 - Intercoolers and aftercoolers, 234–235
 - maintenance of reciprocating air compressor, 296
 - Intermediate pressure (IP), 14
 - Intermediate pressure regulation, 335
 - Internal irreversibilities, 193
 - International Standard Organization (ISO), 44, 119
 - International System (SI), 672
 - Interstage seals. *See also* Centrifugal compressor interstage seals
 - modern steam turbine, 49
 - IP. *See* Intermediate pressure
 - IPS. *See* Instant power stations
 - Irreversibility:
 - external, 193
 - internal, 193
 - ISO. *See* International Standard Organization
- L
- Labyrinth seals:
 - aluminum, 391, 391f, 392
 - centrifugal compressor, 391–392, 391f, 393–394, 395
 - damage to, 392, 393–394
 - fins, 392f, 393–394
 - fouled, 391, 391f
 - with hard materials, 392
 - multistage centrifugal compressor, 425, 428, 428f
 - rotary screw compressor, 333f, 334
 - worn, 391, 392f
 - Last-stage buckets (LSB), 12
 - continuously coupled, 51, 51f, 52f
 - Liners. *See* Cylinder liners

- Liquid entrainment:
 - centrifugal compressor, 412–413
 - overview, 412
 - prevention recommendations, 412–413
 - surge prevention systems and, 412
- Liquid/gas coalescers:
 - advantages, 361
 - efficiency change *versus* gas flow rate, 362
 - functions, 361
 - high-efficiency coalescer, 359–360, 360f
 - overview, 359–360, 361t
 - rating and sizing, 361–363, 362f
 - selection of, 361
 - undersizing problems, 362
- Liquid/gas separation:
 - formation of fine aerosols, 361, 362f
 - overview, 357
- Liquid/gas separation mechanisms:
 - categories, 358
 - centrifugal separation, 358
 - diffusional interception or Brownian Motion, 358
 - droplets and particles in, 358, 358f
 - gravity setting, 358
 - inertial impaction, 358, 359
- Liquid/gas separation technologies:
 - centrifugal separators, 359, 361t
 - equipment ratings and sizings, 361–363, 362f
 - equipment selection, 361, 361t
 - filter vane separators, 359, 361t
 - gravity separators, 359, 361t
 - liquid/gas coalescers, 359–360, 360f, 361–363, 361t, 362f
 - mist eliminators, 359, 361t
- Liquid ring compressors, 208, 209f, 230–231, 230f
- Liquid seals:
 - iso carbon seal, 447f
 - liquid bushing seals, 446–448, 448f
 - overview, 446, 447f
- LMTD. *See* Logarithmic mean temperature difference
- Lobe compressors. *See* Rotary lobe compressors
- Lobe pumps:
 - overview, 564
 - three-lobe, 564, 567f
 - three-lobe pump sectional diagram, 567f
 - two-lobe, 564
- Local instabilities, 423
- Logarithmic mean temperature difference (LMTD), 100
- Low-pressure (LP), 14
- Low-pressure cock (LPC), 6
- Low-pressure turbine exhaust hoods and inner casings:
 - borescope access ports, 55, 56f
 - inner casing, 55f
 - maintenance, 54–55
 - upper exhaust hood, 54f
- LP. *See* Low-pressure
- LPC. *See* Low-pressure cock
- LSB. *See* Last-stage buckets
- Lube oil sampling technique, 681
- Lubricant viscosity. *See* Viscosity
- Lubrication:
 - antifriction bearings, 677f, 678f, 679, 679f
 - bearing housing and, 675–676, 676f, 677f, 678f, 679, 679f
 - compressor cylinder, 282
 - elastohydraulic, 673
 - reciprocating air compressor, 292–293, 292f, 294
 - rod packings' cooling passages and, 282f
- Lubrication systems:
 - force-feed, 260, 261f
 - pressurized, 292–293, 292f
 - reciprocating compressor, 260, 261f
 - steam turbine, 59–60

M

- MacCoull's equation, 674
- Magnetic bearings:
 - axial inlet overhung compressor with, 465–466, 466f, 467f
 - axial stator, 462–463, 462f
 - background, 453–454
 - construction, 461f
 - control loop, 463, 463f
 - development efforts, 465
 - operating experience and benefits, 464–465
 - operating principles, 461–464, 461f, 462f, 463f
 - problems and solutions, 465
 - radial stator, 462, 462f
 - start-up procedure steps, 464
- Magnetic bearing integrated design:
 - overview, 467–468
 - Sulzer-Acec Hofin high-speed oil-free intelligent motor compressor, 468, 469f, 471
 - Sulzer Mopico motor pipeline compressor, 468, 468f, 469f, 470–471, 470f, 471f

Magnetic particles inspection (MPI), 42–43

Mean time between failure (MTBF), 46

Mean time to repair (MTTR), 46

Mechanical seals:

basic components, 517–518, 518f

face pressure, 519–520, 520f, 520t

hydraulic pressure on primary ring, 519, 519f

overview, 504, 504f, 517

power consumption, 521–522, 521t

pressure gradient factors for various services, 520t

pressure-velocity, 521

pressure wedge, 520, 520f

primary ring with hydropads for face lubrication, 529, 531f

seal balance, 518–519, 519f

seal face materials coefficients of friction, 521–522, 521t

seal leakage system and, 444–445, 445f

seal lubrication and seal leakage, 527, 529, 529f, 530f, 531, 531f

single inside pusher seals, 529f

stuffing box gland, 504, 504f

surface waviness, 529, 530f

Mechanical seal classification by arrangement:

double seals, 531, 533–534, 533f

externally mounted single seals, 531, 532, 532f

installations overview, 531

internally mounted single seals, 531, 532, 532f

multiple seals, 531, 533–534, 533f

package seals, 534, 535f

single seals, 531, 532, 532f

tandem seals, 531, 534, 534f

Mechanical seal classification by design:

balanced seals, 534, 535, 536f

bellows seals, 539–541, 540f, 541f

categories, 534

full-convolution bellows seal, 540, 540f

half-convolution bellows seal, 540, 540f

metal bellows seal, 541, 541f

multiple springs, 534, 535, 536, 537f

nonpusher secondary seals, 534, 539–541, 540f, 541f

O-ring, 536, 538f

pusher seal for slurry or sewage applications, 539, 539f

pusher secondary seals, 534, 536–539, 538f, 539f

rotating seals, 534, 535

single springs, 534, 535, 537f

Mechanical seal classification by design (*Cont.*):

stationary metal bellows seal, 541f

stationary seals, 534, 535, 537, 537f, 541

Teflon bellows seal, 540, 541f

unbalanced seals, 534, 535, 536f

V-ring, 536

wedge, 536, 538–539, 538f

Mechanical seal common failure modes:

coning (negative rotation), 552f

full contact pattern, 552f

high wear and grooving, 553f

high wear or thermally distressed surface, 553f

mechanical distortion, 552f

out-of-square mating ring, 553f

thermal distortion (positive rotation), 552f

thermally distressed surface patches, 553f

thermally distressed surface section, 553f

wide contact pattern, 553f

Mechanical seal gland plates:

basic designs, 543, 546, 547f

flush, 543, 547f

flush, vent and drain, 546, 547f

flush-and-quench, 546, 547f

overview, 543

piping arrangements and, 543

plain, 543, 547f

quench, or vent-and-drain, 546, 547f, 548f

restrictive devices, 546, 548f

Mechanical seal installation and

troubleshooting:

cleanliness in, 548

overview, 546

seal drawings and instructions review in, 548, 549f

seal leakage cause identification, 550, 550t–551t, 552f–553f

types of motion that influence seal performance, 546–548, 549f

typical installation reference dimensions, 549f

Mechanical seal leakage causes:

carbon dust on gland ring, 551t

mechanical seal common failure modes, 552f–553f

seal drips steadily, 550t

seal leaks, 551t

seal splits and sputters, 550t

seal squeals during operation, 551t

short seal life, 551t

- Mechanical seal materials of construction:
 - common, 543*t*
 - overview, 542–543
 - P-V in selection of, 542–543, 545*t*–546*t*
 - P-V limitations of seal face, 543, 545*t*–546*t*
 - seal face properties, 544*t*
 - temperature limits of secondary and static, 542, 545*t*
- Mechanical seal mating ring assemblies:
 - clamped-in, 542*f*
 - cup-mounted, 542*f*
 - floating, 542*f*
 - grooved o-ring, 542*f*
 - overview, 541, 542*f*
 - square section, 542*f*
- Mechanical seal temperature control:
 - axial-flow pumping ring, 527, 528*f*
 - balanced seal heat generation, 524*f*
 - cooling circulation, 526, 526*f*
 - external circulation plug port, 526, 526*f*
 - external cooling arrangement, 526, 527*f*
 - heat load from process, 525*f*
 - internal circulation plug port, 526, 526*f*
 - overview, 522–527
 - pump stuffing box cooling methods, 526, 526*f*
 - shutdown problems, 527
 - unbalanced seal heat generation, 523*f*
- Mechanically driven diaphragm pumps:
 - applications, 572–573
 - cross section of single-driven, 572*f*
 - engine-powered single-diaphragm, 572*f*
 - overview, 571–573
 - single-diaphragm, 571, 572*f*
- Metering pumps. *See also* Bellows-type metering pumps
 - diaphragm, 560, 562*f*, 569–570, 569*t*
 - diaphragm metering pump flow characteristics, 562*f*
 - operating limits and applications, 569*t*
- Mist eliminators, 359, 361*t*
- Modern steam turbine design features for reliability:
 - auxiliary systems, 50
 - bearings, 50
 - continuously coupled last-stage buckets, 51, 51*f*, 52*f*
 - controls and instrumentation, 50–51
 - impulse wheel-and-diaphragm construction, 48
 - industrial turbines special features, 51–52, 53*f*
 - Modern steam turbine design features for reliability (*Cont.*):
 - interstage sealing components, 49
 - turbine-rotor design, 48–49, 49*f*
 - vertical flex legs, 52, 53*f*
- Modern steam turbines auxiliary systems components:
 - coolers, 50
 - filters, 50
 - hydraulic oil pumps, 50
 - lube oil pumps, 50
 - solenoid valves, 50
- MPI. *See* Magnetic particles inspection
- MTBF. *See* Mean time between failure
- MTTR. *See* Mean time to repair
- Multistage centrifugal compressors:
 - description, 425, 426*f*, 427–430, 427*f*, 428*f*, 429*f*
 - discharge pressure, 425, 429
 - external connection of oil system, 429, 429*f*, 430
 - first-stage sectional view, 427*f*
 - gas flow in inlet volute, 425, 427*f*
 - horizontally split, 213, 215*f*, 430, 431*f*, 432*f*
 - integral-gear, 214, 217, 217*f*
 - petrochemical plant, 366*f*
 - pressure distribution around impeller, 425, 427*f*
 - sectional view schematic, 426*f*
 - with side loads refrigeration, 213, 216*f*, 217, 217*f*
 - single-phase, 430, 431*f*
 - straight-through with balancing piston, 385*f*
 - three-phase, 430, 432*f*
 - two-phase, 430, 431*f*
 - vertically or radially split (barrel), 214, 216*f*
- Multistage centrifugal compressor components:
 - balance drum, 425, 426*f*, 429
 - balancing line, 429, 429*f*, 430
 - diaphragm bundle, 425
 - diaphragms, 427, 428, 428*f*
 - diffuser, 427, 428, 428*f*, 429
 - discharge volute, 429, 429*f*
 - impellers, 425, 427*f*, 428, 428*f*, 429, 430
 - journal bearings, 425
 - labyrinth seals, 425, 428, 428*f*
 - major, 426*f*
 - oil film end seals, 425, 430
 - outer casing, 425
 - overview, 425, 426*f*
 - rotor hub, 425
 - shaft, 425, 429
 - thrust bearing, 425
 - thrust collar, 425

N

National Association of Corrosion Engineers (NACE), 542

National Electric Manufacturers Association (NEMA), 628

National Fire Protection Association (NFPA), 6

Natural circulation (thermosiphoning, natural convection), 93

Natural gas prices:
 combined-cycle and cogeneration plants economics and, 125–126
 projections 1995, 2000, 2010, 126*t*

NDE. *See* Non-destructive examination

NEMA. *See* National Electric Manufacturers Association

Net positive suction head (NPSH):
 computation of, 509–510, 509*f*
 overview, 509
 pump cavitation and, 509–510, 509*f*, 511*f*, 512*f*
 suction conditions to avoid cavitation, 510, 511*f*, 512*f*

Newtonian fluids:
 non-, 671, 674–675
 overview, 671, 674
 viscosity, 671, 674

NFPA. *See* National Fire Protection Association

Nitrogen oxides (NO_x), 109

Non-depreciable, 141

Non-destructive examination (NDE), 42

Non-exhaust cycle-steam conditions, 12–14

Non-Newtonian fluids:
 classes of, 674–675
 greases, 674
 oils at low temperatures, 675
 overview, 671, 674
 VI-improved oil, 674–675
 viscosity, 671, 674–675

Non-perfect gases:
 constants for some fluids, 184, 185*f*
 generalized compressibility factor chart, 184, 184*f*
 property relationships, 182, 184, 184*f*, 185*t*

Nonreverse ratchets, 614

Nonutility generator (NUG), 128

NO_x . *See* Nitrogen oxides

Nozzle boxes and diaphragms:
 fossil unit, 57*f*
 maintenance, 56

NPSH. *See* Net positive suction head

Nuclear Regulatory Commission (NRC), 622

NUG. *See* Nonutility generator

O

Oils. *See also* Used oil analysis
 ball bearings with oil rings, 678*f*, 679*f*
 bearing housing using lubrication by, 679, 679*f*
 compressor valves, 294–295
 at low temperatures, 675
 oil whirl as forcing frequency cause, 701
 VI-improved, 674–675

Oil film, 672
 end seals, 425, 430

Oil-flooded single-screw compressors, 331, 340*f*. *See also* Rotary screw liquid or oil-injected compressors
 capacity control, 341, 341*f*
 dirty gas application problems, 342–343
 gas flow, 339, 340*f*, 341
 overview, 339, 341–343
 process flow schematic, 341, 341*f*
 single-shaft, two-stage, 341, 342*f*
 slide pistons, 341, 341*f*
 two-stage version, 339, 340*f*, 341

Oil-injected compressors. *See* Oil-flooded single-screw compressors; Rotary screw liquid or oil-injected compressors

Oil system. *See also* Seal oil system
 multistage centrifugal compressor, 429, 429*f*, 430
 problems of compressor, 453–454

Once-through boilers:
 effect of steam conditions on heat rate, 81*f*
 flow diagram, 79*f*
 overview, 78–79
 with pulverized coal furnace, 80*f*

P

P-V. *See* Pressure *versus* volume

Packings:
 cartridges and arrangements, 281*f*
 cooling system for piston rod pressure, 282, 283*f*
 cylinder head, 297, 298*f*
 diaphragm shaft packing seals, 56–58
 distance pieces and, 284, 284*f*, 285*f*
 lubrication and cooling passages on rod, 282*f*
 maintenance of reciprocating air compressor, 297, 297*f*, 298*f*

- Packings (*Cont.*):
 oil-stop-head, 297, 297f
 overview, 281–282
 piston rods and, 281, 282, 282f, 283f, 284, 284f
 positive-pressure, 38, 38f, 39f
 reciprocating compressor, 281–282, 281f, 282f, 283f
- Partial pressure:
 gas, 188–189
 vapor pressure, 186–187
- Particle count analysis:
 modern particle counter, 687–688, 688f
 of used oil, 687–688, 688f
- Peak shaving, 1
- PEEK. *See* Polyether ether ketone
- Perfect gases, 179
 property relationships, 182, 183t
- Perfect gas laws:
 Amagat's law, 180
 Amonton's law, 180
 Avogadro's Law, 181
 Boyle's law (isothermal law), 180
 Charles' law, 180
 Dalton's law, 180
 heat and work, 181, 181t
 overview, 179
- Permanent viscosity loss (PVL), 675
- Permitting cost, 141
- Pistons:
 completion of delivery stroke in compressor operation, 176–177, 177f
 materials, 272
 reciprocating compressor, 272
- Piston and rider rings:
 band type, 273, 274f
 materials, 272
 PTFE, 272, 273, 274f
 reciprocating compressor, 272–273, 273f, 274f
 split type, 273
 types, 273
- Piston compressors. *See* Sliding crosshead piston compressors; Trunk piston compressors
- Piston pumps:
 basic design, 556f
 double-acting, 557, 557f
 horizontal, single-acting pump operating principle, 556f
 overview, 555, 556f, 557
 reciprocating pistons in, 555, 556f
 simplex pump liquid delivery rate, 556f
- Piston rings. *See also* Piston and rider rings
 maintenance of reciprocating air compressor, 296
- Piston rods:
 distance pieces and, 284, 284f
 overview, 280–281
 packings and, 281, 282, 282f, 283f, 284, 284f
 pressure packing cooling system, 282, 283f
 reciprocating compressor, 280–281, 281f
 rolled threads, 281, 281f
- Plug-type unloaders, 242–243, 244f, 245f
- Plunger pumps:
 capacities of horizontal, 661t–662t
 double-acting, 558, 558f, 560
 operating principle of horizontal double-acting, 558f
 operating principle of single-acting horizontal, 557f
 operating principle of triplex, 558f
 overview, 555, 557–560
 power-driven triplex, 559f
 single-acting, 557, 557f, 559
 triplex (three-plunger), 558–559, 558f, 559f
- Pneumatically powered diaphragm pumps:
 air-operated double-diaphragm, 573–574, 573f, 574f, 575f
 air-operated double-diaphragm pump performance chart, 574, 576f
 air-operated double-diaphragm pump with ball check valves, 574, 575f
 air-operated double-diaphragm pump with flap check valves, 573–574, 573f, 574f
 double-diaphragm, 573–574, 573f, 574f, 575f
 overview, 573–574
- Polyether ether ketone (PEEK), 277
- Polytropic efficiency, centrifugal compressor
 interstage seals and, 391, 392, 393, 394–395
- Polytropic efficiency, compressor, 193–194, 193f
- Polytropic exponent, 181
- Port-type unloaders, 242, 243–244, 246f, 247f
- Positive displacement compressors:
 applications, 204
 operation, 219, 221f
 overview, 203, 204, 219, 220f, 223
- Positive displacement compressor capacity
 control. *See also* Reciprocating compressor capacity control
 bypass, 241
 clearance pockets and, 244–246
 start/stop system, 240

- Positive displacement compressor capacity control (*Cont.*):
 - unloaders and, 241–244, 242*f*, 243*f*, 244*f*, 245*f*, 246*f*, 247*f*, 248*f*, 249*f*, 250*f*
 - variable-speed drive, 240–241
- Positive displacement compressor performance, 222–231. *See also* Reciprocating compressor performance
 - capacity control, 240–246, 242*f*, 243*f*, 244*f*, 245*f*, 246*f*, 247*f*, 248*f*, 249*f*, 250*f*
 - clearance pockets, 244–246, 248*f*, 249*f*, 250*f*
 - losses, 237
 - pressure ratio, 237
 - rotary screw compressors leakage, 249, 251
- Positive displacement compressor types, 220*f*, 223. *See also* Reciprocating compressors; Rotary compressors
- Positive displacement pumps, 555
- Positive displacement pump types. *See also* Reciprocating pumps; Rotary pumps
 - overview, 555
- Pour points, 674, 675
- Pour point depressants, 675
- Pre-C.O. operating cost, 141
- Pressure surge. *See also* Water hammer
 - water hammer as, 603
- Pressure *versus* volume (P-V), 175
- Pressurized lubrication system, 292*f*
 - maintenance, 292–293
- Product life cycle cost, 45
- Pseudo-critical conditions, gas mixture, 190–191, 190*f*
- PTFE. *See* Teflon [polytetrafluoroethylene]
- Pumps. *See also* Axial-flow pumps; Canned motor pumps
 - auxiliary lubricating, 305
 - characteristics with varied impeller diameter, 660*f*
 - compensating, 303, 305
 - hydraulic oil, 50
 - lube oil, 50
 - overview, 485, 486*f*
 - pressure and capacity ranges, 485, 486*f*
 - shaft-driven gear, 305
 - shutdown and water hammer, 614
 - vertical wet-pit condenser circulating, 678*f*, 679
 - vibration analysis, 513, 515
- Pump bedplates:
 - cast iron, 505*f*
 - overview, 504
 - structural steel, 506*f*
- Pump bidding and negotiation:
 - bid list, 631
 - cost, 631
 - economic life, 631
 - efficiency, 631
 - evaluation of bids, 631
 - guarantee/warranty, 631–632
 - public and private sector, 630–631
 - sample bid evaluation, 632*t*–635*t*
 - spare parts, 631
- Pump categories. *See also* Centrifugal pumps;
 - Positive displacement pumps
 - characteristics of modern, 658*t*
 - displacement (rotary), 485
 - dynamic (reciprocating), 485
 - modern classes and types, 658*f*
- Pump cavitation:
 - centrifugal pump, 509
 - impeller damaged by, 508, 508*f*
 - net positive suction head and, 509–510, 509*f*, 511*f*, 512*f*
 - overview, 508–509
- Pump and driver selection:
 - code requirements, 622
 - driver type, 623
 - fluid characteristics, 622–623
 - overview, 621–622
 - pump characteristics, 622
 - pump driver special considerations, 628–629
 - pump driver speed selection, 642–643, 642*f*
 - pump materials, 623
- Pump engineering of system requirements:
 - alternate modes of operation, 620
 - fluid type, 619
 - future system changes, 621
 - margins, 620–621
 - overview, 619
 - system-head curves, 619–620
 - wear, 621
- Pumps installed in parallel analysis:
 - calculations, 637–639
 - check valve locations to prevent reverse flow, 641*f*
 - parallel pumping check valve locations, 641*f*
 - single- and dual-parallel pump characteristic curves, 640*f*
 - system-head curve for parallel pumping, 641*f*
 - typical parallel pump characteristic curves, 640*f*

Pumps installed in series analysis:

- calculations, 637
- seriesed pump characteristic and system-head curves, 639f
- seriesed pump curve, 637

Pump maintenance. *See also* Pump troubleshooting

- annual inspection, 584
- centrifugal pump, 510
- complete overhaul, 584–585
- end play, 512, 513f
- gland register or stuffing box concentricity, 512, 515f
- overview, 583
- pump operation daily observations, 583
- pump trouble diagnoses, 586
- radial deflection, 512, 513f
- recommended, 511–512, 513f, 514f, 515f
- record of inspections and repairs, 585–586
- semiannual inspection, 583–584
- shaft run-out, 512, 514f
- spare and repair parts, 585
- stuffing box to shaft perpendicularity, 512, 514f

Pump materials:

- diaphragm pump, 576–577
- in pump and driver selection, 623

Pump mechanical seals. *See* Mechanical seals

Pump performance. *See also* Centrifugal pump

- performance characteristics
- testing, 628

Pump selection and procurement:

- bidding and negotiation, 630–632, 632t, 633t, 634t, 635t
- drawing and data requirements form, 629, 630f
- engineering of system requirements, 619–621
- overview, 619
- performance testing, 628
- pump and driver selection, 621–623
- pump drivers, 628–629
- pump specifications, 623–628, 624f, 625f
- quality assurance and quality control, 629
- sequence of steps in, 619
- special considerations, 628–629, 630f
- special control requirements, 629

Pump selection procedures:

- centrifugal pump composite rating chart, 659f
- centrifugal pump rating table, 660f

Pump selection procedures (*Cont.*):

- centrifugal pump selection chart, 657f
- drawing piping layout, 655, 656f
- horizontal plunger pump capacities, 661t–662t
- modern classes and types, 658f
- modern pump characteristics, 658t
- obtaining pumped liquid data, 656, 657f
- pipeline single-line diagram, 656f
- pump capacity determination, 655
- pump category and type selection, 657, 658f, 658t, 659, 659f, 660f, 660t, 661f, 661t–662t
- pump characteristics with varied impeller diameter, 660f
- pump total head determination, 656
- rotary pump capacity ranges, 661f
- selected pump evaluation, 660

Pump specifications:

- bidding documents, 626
- centrifugal pump data sheet, 624t
- codes and standards, 625–626
- commercial terms, 627–628
- rotary pump data sheet, 625t
- technical specification, 624t, 625t, 626–627
- types, 623–624

Pump total head calculation, 650–651, 654–655

- equivalent length of piping system fittings and valves, 652t
- pipe fittings resistance coefficients, 653f
- pipe friction loss for water, 655t
- typical pump suction and discharge piping arrangements, 650f

Pump troubleshooting. *See also* Pump maintenance

- centrifugal pumps, 586, 587t
- pump maintenance and, 583–586
- reciprocating pumps, 586, 589t
- rotary pumps, 586, 588t
- steam pumps, 586, 590t
- vibration diagnostics, 588–590, 591t–598t, 599–602, 601t

Pumping system calculations:

- centrifugal pump best operating speed selection, 648–649, 649t
- centrifugal pump capacity and efficiency determination, 646–647, 647f, 648f
- centrifugal pump selection using similarity or affinity laws, 644–646
- centrifugal pumps affinity laws, 643–644

Pumping system calculations (*Cont.*):
 pump characteristic curves for series installation, 638f
 pump driver speed selection, 642–643, 642f
 pump selection procedures, 655–657, 656f, 657f, 658f, 658t, 659–660, 659f, 660f, 660t, 661f, 661t–662t
 pump total head calculation, 650–651, 650f, 652t, 653f, 654–655, 655t
 pumps installed in parallel analysis, 637–639, 640f, 641, 641f
 pumps installed in series analysis, 637, 638f, 639f
 specific speed upper limits of centrifugal pumps, 642f
 specific speed upper limits of mixed-flow and axial-flow pumps, 647f
 PVL. *See* Permanent viscosity loss

Q

QA. *See* Quality assurance
 QC. *See* Quality control
 Quality assurance (QA), 629
 Quality control (QC), 629
 Quasi-flexible rotors, 407

R

Radiation, 93
 RAM. *See* Reliability, availability and maintenance
 RAM performance:
 cost/benefit analysis of high, 61
 value calculation, 61
 Reciprocating air compressor maintenance:
 air filters and suction lines, 289, 289f
 air receiver location and capacity, 289–290
 cleaning outside casing, 297
 foundation considerations, 287–289, 288f
 intercoolers and aftercoolers, 296
 location recommendations, 287
 lubrication, 292–293, 292f, 294
 non-lubricated cylinders, 293
 overview, 287
 packings, 297, 297f, 298f
 piston rings, 296
 service check chart, 298t–300t
 starting new compressors, 290–291
 troubleshooting and, 287–298, 298t–300t
 valves, 293–296, 294f

Reciprocating compressors. *See also* Diaphragm compressors
 balance opposed, 253, 254f, 255f
 basic components, 208, 209f
 bearings, 259–260
 bellows compressors, 227, 227f, 228f
 common characteristics, 224f
 compression cycle, 210
 compressor cooling, 269, 270f, 271–272, 271f, 272f
 connecting rods, 262–263, 262f
 crankshaft design, 256–257, 259f, 260f, 261f
 crossheads, 263–264, 263f
 cylinder lubrication, 282
 cylinders, 240, 264–265, 265f, 266f, 267f, 268–269, 268f, 269f, 270f, 282
 distance pieces, 282, 284, 284f, 285f, 286
 four-throw, 210, 212f
 frames, 264, 264f
 lubrication systems, 260, 261f
 nomenclature of, 211f
 operation, 175–176, 176f
 overview, 203, 208, 210, 220f, 223–224, 253
 packings, 281–282, 281f, 282f, 283f
 piston and rider rings, 272–273, 273f, 274f
 piston rods, 280–281, 281f
 pistons, 272
 sliding crosshead piston compressors, 225–226, 225f
 trunk piston compressors, 224–225, 225f, 226f, 268–269, 269f
 valves, 246–248, 250f, 274–275, 274f, 275f, 276f, 277, 277f, 278f, 279f, 280, 280f, 293–296, 294f
 vertically arranged, 253, 255f, 256f, 257f, 258f
 Reciprocating compressor capacity control:
 bypass, 241
 clearance pockets and, 244–246
 start/stop system, 240
 unloaders and, 241–244, 242f, 243f, 244f, 245f, 246f, 247f, 248f, 249f, 250f
 variable-speed drive, 240–241
 Reciprocating compressor performance:
 capacity control, 240–246, 242f, 243f, 244f, 245f, 246f, 247f, 248f, 249f, 250f
 clearance pockets, 244–246, 248f, 249f, 250f
 compressor valves, 246–248, 250f
 leakage, 248
 losses, 237
 rating, 237
 sizing, 237–240, 239f

- Reciprocating compressor selection:
 - compressor sizing and, 237–238
 - cost in, 238
 - determination of parameters in, 237, 238
 - efficiency in, 238
 - high-horsepower applications in, 238
 - importance of variables in, 238
 - low-horsepower applications in, 238
 - system reliability in, 238
- Reciprocating compressor sizing:
 - compressor selection and, 237–238
 - cylinder selection in, 240
 - discharge temperature in, 238–239, 239f
 - intercooling and, 238–239
 - stages in, 238–240, 239f
- Reciprocating pumps:
 - main characteristics, 555
 - overview, 555
- Reciprocating pump troubleshooting:
 - mechanical troubles, 589t
 - overview, 586
 - suction troubles, 589t
 - symptoms, 589t
 - system troubles, 589t
- Reciprocating pump types. *See also* Diaphragm pumps; Piston pumps; Plunger pumps
 - overview, 555
- Reheat cycle steam condition, 16–17
 - comparison of non-reheat and reheat expansions, 16f
 - three-pressure reheat combined cycle, 16, 17f
- Reliability, availability and maintenance (RAM), 61
- Reorder point (ROP), 585
- Reorder quantity (ROQ), 585
- Rods. *See* Connecting rods; Piston rods
- ROP. *See* Reorder point
- ROQ. *See* Reorder quantity
- Rotary compressors. *See also* Rotary lobe compressors
 - common characteristics, 228f
 - overview, 220f, 227–228
 - rotary liquid ring compressors, 208, 209f, 230–231, 230f
 - sliding rotary vane compressors, 206, 207f, 230, 230f, 235f
- Rotary lobe compressors (blowers). *See also* Straight lobe compressors
 - overview, 204–205, 206f, 229, 229f
- Rotary pumps, 555, 562
 - capacity ranges, 661f
 - data sheet, 625t
- Rotary pump troubleshooting:
 - overview, 586
 - suction troubles, 588t
 - symptoms, 588t
 - system troubles, 588t
- Rotary pump types. *See also* Cam pumps; Gear pumps; Lobe pumps; Screw pumps; Vane pumps
 - overview, 562
- Rotary screw compressors, 318f. *See also* Rotary screw liquid or oil-injected compressors
 - advantages, 338
 - applications, 318, 320, 321
 - bearings, 332, 332f
 - compressor seals, 332, 333f, 334
 - disadvantages, 338–339
 - dry compressors, 321, 328, 331
 - industry experience, 334–335, 335f, 336f, 337
 - labyrinth seals, 333f, 334
 - leakage, 249, 251
 - maintenance history, 337
 - mechanical construction, 332, 332f, 333f, 334
 - medium-sized package, 319f
 - noise-reducing equipment, 338, 339f
 - oil-free with integral step-up gears, 319f
 - overview, 207–208, 207f, 228–229, 229f
 - performance summary, 338–339, 339f
 - small packaged, 318f
 - sound levels with and without acoustic enclosures, 339f
 - twin-screw machines, 317
 - typical rotor combinations, 320f
- Rotary screw compressor capacity control:
 - bypass method, 329–331, 330f
 - compressor discharge unloading method, 331–332
 - diaphragm valve in, 330, 330f, 331
 - internal volume-regulating device, 331–332, 331f
 - overview, 328–332
 - suction throttling method, 331
 - variable-speed method, 328–329
- Rotary screw compressor operating principles:
 - API Standard 619 and, 326

Rotary screw compressor operating principles

(Cont.):

- compression process and ratios, 322–323, 323f, 324
- dynamic flow power loss formula, 325, 326f, 327f
- empirical loss factor, 325, 326f, 327f
- flow calculation, 324
- overview, 322–324
- P-V diagram of modern helical, 323f
- power calculation, 324–326, 326f, 327f
- temperature rise, 328
- VDI Specification 2045 and, 326

Rotary screw compressor operation:

- compression phase, 317, 320f
- discharge phase, 317, 320f
- suction phase, 317, 320f
- transfer phase, 317, 320f
- working phases, 317, 320f

Rotary screw liquid or oil-injected compressors. *See also* Oil-flooded single-screw compressors

- capacity control, 331–332, 331f
- coke gas plants use of, 334
- flow diagram of water-injected two-stage, 336f
- internal volume-regulating device, 331–332, 331f
- oil-injected, 328f, 329f, 331f
- operating principle, 329f
- overview, 321, 322
- temperature rise, 328
- water-injected two-stage screw compressor, 334–335, 335f, 336f, 337

Rotating stall, 422

Rotor:

- balancing in centrifugal compressors, 406–407, 407f
- dynamics in centrifugal compressors, 406
- hub of multistage centrifugal compressor, 425
- lateral critical speed mode shapes, 406
- two-phase centrifugal compressor with central double-flow, 430, 432f
- vibration analysis, 589, 590, 591t, 592t, 595t, 599

Rotor categories:

- quasi-flexible rotors, 407
- stiff shaft rotors, 407
- very flexible rotors, 407

S

Saturation temperature, 185

Saybolt Universal seconds (SUS), 672, 673f

SCC. *See* Self-cleaning coalescersSCL. *See* Surge control line

Scotch marine boiler, 64, 66, 66f

Screw compressors. *See* Rotary screw compressors

Screw pumps:

- cross-section of twin-screw, 566f
- overview, 563–564
- single-rotor, 563, 564
- three-rotor, 564, 566f
- twin-screw, 564, 565f
- twin-screw pump bodies and flow patterns, 565f

Seals. *See also* Compressor seals; Dry seals;

Mechanical seals

- diaphragm shaft packing seals, 56–58
- rub vibration analysis, 590, 591t
- steam turbine removable, 33f

Seal housing system. *See also* Bearing housing

- compressor, 442–443, 442f
- seal leakage, 444–445, 445f
- subsystem of seal oil system, 442–443, 442f

Seal leakage:

- causes of mechanical, 550, 550t–551t, 552f–553f
- mechanical seal lubrication and, 527, 529, 529f, 530f, 531, 531f

Seal leakage system:

- automatic drainer, 444
- degassing tank, 444
- overview, 443–444
- pump mechanical seal, 444–445, 445f
- seal housing system, 444–445, 445f
- seal liquid, 452
- supply system, 444
- vent system, 444

Seal-less pump motors. *See also* Canned motor pumps

- designs for centrifugal pumps, 581f
- overview, 582

Seal liquid leakage system, 452

Seal oil system:

- atmospheric draining subsystem, 443
- overview, 441, 442f
- seal housing subsystem, 442–443, 442f
- seal leakage subsystem, 443–445, 445f
- supply subsystem, 441

- Seal supply systems:
 contact type gas side seal, 450, 451*f*
 flow through atmospheric side bushing seal, 451
 flow through gas side contact seal, 451
 flow through seal chamber, 452
 shaft seal, 450, 451*f*
- Seal systems. *See* Compressor seal systems
- Sealing:
 elements of diaphragm compressors, 308
 mechanisms, 453
 modern steam turbine interstage, 49
 problem risk minimization in dry gas seal, 460–461
 steam turbine steam path, 56–58, 57*f*
- Self-cleaning coalescers (SCC):
 applications, 345
 filtration equipment, 345
 overview, 343–344, 344*f*
- Separating gases and solids from gases.
 See Liquid/gas separation
- Shear stability, 675
- Shell-and-tube heat exchangers:
 baffles, 102, 104*f*
 closed feedwater heaters, 105*f*, 106, 106*f*
 factors for two types of, 104*f*
 fluid paths variations, 103*f*
 overview 101–103, 104*f*, 101–103, 103*f*, 104*f*, 105*f*
 rolled tube joint for, 103, 105*f*
 typical parts of, 103, 105*f*
 U-tube bundle, 102, 105*f*
- Shell-type boilers. *See also* Fire-tube boilers
 accumulator, 64
 electric, 64
 overview, 64
- SI. *See* International System
- Sine waves, application to vibration, 691, 692*f*, 693*f*, 694*f*
- Single-shaft combined-cycle plants:
 advantages of, 117–118
 ambient temperature effect on performance, 114*f*
 arrangement of, 116, 116*f*, 117*f*
 environmental impact, 115
 overview, 111–112
 performance of, 112–114, 112*t*, 113*f*, 114*f*
 STAG product line ratings, 112*t*
- Single-shaft combined-cycle units:
 auxiliary steam supply, 116
 equipment configurations, 115–116, 115*f*
 heat rate variation with power output, 114*f*
 maintenance, 117
 non-heat steam cycle, 113*f*
 overview, 111–112
 starting systems, 116
 three-pressure reheat steam cycle, 113*f*
- Sliding crosshead piston compressors, 225–226, 225*f*
- Sliding rotary vane compressors, 206, 207*f*, 230, 230*f*
 inlet unloader for, 235*f*
- SLL. *See* Surge limit line
- Solid particle erosion (SPE), 34
- SPE damage:
 steam turbine advanced design features to minimize, 34–35, 35*f*, 38*f*
 steam turbine maintenance and, 34–35, 35*f*, 37*f*, 38*f*
- Specific gravity, gas, 188
- Specific heats, 189–190, 190*t*
- Specific humidity, 187
- Spectrographic analysis, emission, 684–686, 685*f*
- Spectrometers:
 direct-coupled plasma, 685, 685*f*
 double-beam infrared, 686–687
 emission, 684–686, 685*f*
 inductively coupled plasma, 685, 685*f*
- Spectrometry, infrared, 686–687
- SPT. *See* Standard pressure and temperature
- SSSF. *See* Steady-state steady-flow
- STAG. *See* Steam and gas
- STAG combined-cycle plants. *See also* Combined-cycle plants
 ACC in, 11
 overview, 11
 steam turbines in, 11
- STAG plants, 17–18. *See also* STAG combined-cycle plants
 approximate output, 13*t*
- STAG plants, steam turbine application:
 non-exhaust cycle-steam conditions, 12–14
 reheat cycle steam condition, 16–17, 16*f*, 17*f*
 STAG plants structure, 11, 12*t*, 13*t*
 steam turbine exhaust size selection, 12, 14*t*, 15*f*

- STAG plants, steam turbines performance, 17–18
 - advanced combined cycles, 21*t*, 22*t*–23*t*
 - non-reheat steam turbines, 18*t*, 19*t*, 20*t*
- STAG plants, structure, 11
 - GE gas turbine exhaust characteristics, 12*t*
 - STAG power plants approximate output, 13*t*
- STAG power plants. *See* STAG plants
- STAG steam turbine, casing arrangements, 23*f*
 - non-heat multi-shaft, 19, 23*f*, 24*f*, 25*f*
 - non-heat single shaft, 26, 26*f*
 - reheat multi-shaft, 26, 27*f*, 28*f*
 - reheat single shaft, 26, 29*f*
- Standard pressure and temperature (SPT), 188
- Standards of Hydraulic Institute, 622
- Starter motor, 4
- Steady-state conduction, 93–95, 94*f*
- Steady-state steady-flow (SSSF), 181
- Steam and gas (STAG), 11. *See also specific STAG topics*
- Steam and water injection, 123–124, 124*f*
- Steam drums:
 - overview, 74–75
 - separation, 74–75, 75*f*
- Steam generators. *See also* Boilers
 - auxiliary components, 63
 - components of, 63
 - economizers, 79–80
 - fire-tube boiler, 64, 65*f*, 66, 66*f*
 - industrial, 63, 64
 - once-through boilers, 78–79, 79*f*, 80*f*, 81*f*
 - overview, 63–64
 - schematic of modern, 70*f*
 - steam drums and steam drum separation, 74–75, 75*f*
 - utility, 63–64
 - water circulation, 72–74, 73*f*
 - water-tube boiler, 66, 67*f*, 68–72, 68*f*, 70*f*, 71*f*
- Steam generator air preheaters:
 - Ljungstrom, 82, 83*f*
 - overview, 80–82
 - recuperative, 81
 - regenerative, 82
 - tubular counterflow, 82*f*
- Steam generator control:
 - feedwater and drum-level control, 88, 88*f*
 - steam-pressure control, 88, 89*f*
 - steam-temperature control, 89–91, 90*f*, 91*f*
- Steam generator fans:
 - centrifugal, 83–85, 84*f*, 86*f*
 - control of, 85, 86*f*
 - forced-draft (FD) fan, 69, 83, 84, 84*f*
 - induced-draft (ID) fan, 69, 83
- Steam generator stacks:
 - dispersion, 87–88, 87*f*
 - driving pressure, 86–87, 87*t*
- Steam generator superheaters and reheaters:
 - arrangement of economizer, air preheater and, 77*f*
 - convection superheater, 76, 76*f*
 - differences and types of, 77, 77*f*, 78*f*
 - overview, 75–76
 - radiant superheater, 76–77, 76*f*
 - steam-temperature control of superheaters, 89, 90*f*, 91, 91*f*
- Steam hammer:
 - expansion joints damaged by, 616*f*
 - overview, 617
 - water hammer and, 617
- Steam path sealing features:
 - bucket tip spillstrings and diaphragm shaft packing seals, 56–58
 - maintenance, 56–58
 - removable seals, 57*f*
- Steam pump troubleshooting:
 - overview, 586
 - suction troubles, 590*t*
 - symptoms, 590*t*
 - system problems, 590*t*
- Steam-temperature control:
 - by attemperation and attemperators, 89–91, 90*f*, 91*f*
 - overview, 89
 - of superheaters, 89, 90*f*, 91, 91*f*
- Steam turbines:
 - removable seals, 33*f*
 - in STAG combined-cycle plants, 11
- Steam turbine advanced design features:
 - to minimize SPE damage, 34–35, 35*f*, 38*f*
 - positive-pressure packing, 38, 38*f*, 39*f*
 - to reduce reheat erosion, 36–37, 37*f*, 38*f*
 - to reduce start-up rubbing, 37–38, 38*f*, 39*f*
 - to reduce steam turbine damage, 34–38, 35*f*, 36*f*, 37*f*, 38*f*, 39*f*
 - reheat diaphragm, 36–37, 37*f*, 38*f*
 - steam turbine maintenance and, 34–38, 35*f*, 36*f*, 37*f*, 38*f*, 39*f*

- Steam turbine bearings:
 - journal bearings, 58, 59, 59f
 - modern, 50
 - thrust-bearings, 34f, 50, 58, 59f
 - tilted-pad journal bearings, 34f, 50, 58, 59, 59f
- Steam turbine damage:
 - advanced design features to reduce, 34–38, 35f, 36f, 37f, 38f, 39f
 - first reheat stage, 36–37, 36f
 - reheat erosion, 36–37, 36f, 37f
 - SPE damage, 34–35, 35f, 37f, 38f
 - start-up phase rubbing, 37–38
 - steam turbine maintenance and, 34–38, 35f, 36f, 37f, 38f, 39f
- Steam turbine design attributes enhancing maintainability:
 - bearings, 58–59, 59f
 - bolting, 60
 - low-pressure turbine exhaust hoods and inner casings, 54–55, 54f, 55f, 56f
 - lubrication system, 59–60
 - maintenance recommendations, 60
 - nozzle boxes and diaphragms, 56, 57f
 - overview, 53
 - primary steam valves, 58
 - rotors, 55–56
 - steam path sealing features, 56–58, 57f
 - turbine-generator control and supervisory systems, 60
 - turbine shells, 53, 54f
- Steam turbine design attributes enhancing reliability:
 - modern steam turbine design features, 48–52, 49f, 51f, 52f
 - overall mechanical design approach, 47–48, 47f, 48f
 - overview, 47
- Steam turbine design philosophy:
 - product life cycle cost minimization in, 45
 - for steam turbine reliability and maintainability, 45
- Steam turbine exhaust size selection, 12
 - combined-cycle steam turbines last stages, 14t
 - steam turbine wheel output and, 15f
- Steam turbine maintenance:
 - activities, 32–34, 33f, 34f
 - advanced design features and, 34–38, 35f, 36f, 37f, 38f, 39f
- Steam turbine maintenance (*Cont.*):
 - boroscopic inspection, 31, 32f
 - life cycle operating cost, 31
 - major cause of repair and, 31
 - SPE damage and, 34–35, 35f, 37f, 38f
 - steam turbine damage and, 34–38, 35f, 36f, 37f, 38f, 39f
 - steam turbine reliability, 31
- Steam turbine performance measures:
 - availability, 46
 - maintenance, 47
 - reliability, 46
- Steam turbine product structure:
 - casing arrangements, 19, 23f, 24f, 25f, 26, 26f, 27f, 28f, 29f
 - co-generation applications, 29, 30f
 - STAG plants steam turbines performance, 17–18, 18t, 19t, 20t, 21t, 22t–23t
- Steam turbine reliability and maintenance costs:
 - direct costs, 45
 - indirect costs, 45
- Steam turbine reliability and maintenance enhancement features:
 - cost/benefit analysis of high RAM performance, 61
 - design attributes enhancing maintainability, 53–60
 - design attributes enhancing reliability, 47–52, 53f
 - measures of reliability, availability and maintainability, 46–47
 - steam turbine design philosophy, 45
- Steam turbine selection for combined-cycle power systems:
 - application considerations, 11
 - overview, 11
 - steam turbine application to STAG plants, 11–17
 - steam turbine product structure, 17–30
- Stiff shaft rotors, 407
- Straight lobe compressors (blowers):
 - applications, 349, 351
 - bearings, 353
 - casing, 352
 - construction, 352–353
 - cross section through, 350f
 - higher compression ratios, 353
 - installation, 354–355, 354f
 - main components, 352

Straight lobe compressors (blowers) (*Cont.*):
 noise characteristics, 351
 operating characteristics, 349
 operating principle, 349–350, 349*f*, 350*f*
 power reduction, 353–354, 353*f*, 354*f*
 pulsation characteristics, 350*f*, 351
 rotors, 352
 single-stage compression power, 353, 353*f*
 staging, 353–354, 353*f*, 354*f*
 suction and discharge arrangement, 354–355, 354*f*
 timing gears, 352
 torque characteristics, 352
 two-lobe compressor pumping schematic, 350*f*
 two-stage compression, 353–354, 354*f*
 Sulzer-Acec Hofin high-speed oil-free intelligent motor compressor, 468, 469*f*, 471
 Sulzer Mopico motor pipeline compressor:
 application ranges, 470, 470*f*
 cost per installed horsepower, 471
 design advantages, 470
 dimensions, weight and simplified cross section, 469*f*, 470–471, 471*f*
 installation schematic, 470, 471*f*
 magnetic bearing integrated design, 468, 468*f*, 469*f*, 470–471, 470*f*, 471*f*
 overview, 468, 468*f*, 469*f*, 470
 variable frequency drive, 470
 Surge. *See also* Centrifugal compressor surge;
 Dynamic compressor surge; Pressure surge
 suppressors, 612
 Surge control line (SCL), 436*f*, 438
 Surge identification:
 centrifugal compressor, 412
 method, 412
 Surge limit line (SLL), 436, 436*f*
 Surge prevention. *See also* Surge protection
 system
 dynamic compressor, 438–440, 439*f*
 Surge prevention systems:
 bias station, 410
 centrifugal compressor, 408–411, 409*f*, 410*f*
 component functions, 409–412
 FCV anti-surge recycle valve, 411
 flow element, 409
 flow transmitter, 410
 LAG, 411
 liquid entrainment and, 412
 LX low-signal selector, 411

Surge prevention systems (*Cont.*):
 overview, 408–409, 409*f*
 radio station, 410
 surge control line, 410, 410*f*
 surge controller, 410, 410*f*
 typical, 409*f*
 Surge protection system. *See also* Surge prevention systems
 dynamic compressor single-parameter, 438, 439, 439*f*
 dynamic compressor standard two parameter, 439, 440*f*
 Surge tanks:
 one-way, 613
 overview, 614
 water hammer and, 613, 614
 water hammer and one-way, 613
 SUS. *See* Saybolt Universal seconds
 Symbols, of gas laws and compression principles, 175, 201–202

T

TAN. *See* Total acid number
 TBN. *See* Total base number
 TDC. *See* Top dead center
 Teflon [polytetrafluoroethylene (PTFE)], piston
 and rider rings, 272, 273, 274*f*
 Thermal conductivities, 95
 Thermodynamics:
 first law, 179
 second law, 179
 Thrust-reducing seals:
 axial inlet overhung compressor with, 465–466, 466*f*, 467*f*
 beam-type compressor, 465, 466–467, 467*f*
 beam-type thrust reducer with, 466, 467*f*
 magnetic bearings and, 465–466, 466*f*, 467*f*
 overhung axial inlet thrust reducer with, 465–466, 466*f*, 467*f*
 overhung compressor, 465–466, 466*f*, 467*f*
 overview, 465–466
 TMR. *See* Triple modular redundant
 Top dead center (TDC), 303
 Total acid number (TAN), 687
 Total base number (TBN), 687
 Transducers:
 acceleration, 695, 696*f*
 displacement transducer
 (proximity probe), 695
 velocity, 695, 696*f*

Transducer selection:

- frequency domain, 698, 699f
- guide of useful ranges, 697f
- overview, 697, 697f
- time domain, 697–698, 698f

Triple modular redundant (TMR), 50

- Trunk piston compressors, 224–225, 225f, 226f
- conventional and tandem cylinders, 268–269, 269f

Turbine-generator balancing:

- high-speed balancing, 41
- of high vibration, 42
- low-speed balancing, 41
- static balancing, 42
- of turbine rotors, 41–42

Turbine-generator maintenance:

- control and supervisory systems for, 60
- of electric runout, 43
- of rotor runout, 43
- of rotor-shaft vibration, 43
- of turbine blades, 43
- of turbine rotor bending, 43
- for turbine rotor or journal scratch or crack, 42–43

Turbine-generator vibration analysis, 42

Turbine-rotor design:

- modern features of, 48–49
- wheel-and-diaphragm design, 48–49, 49f

Turbine shells, 53, 54f

Turbocompressors, 365. *See also* Dynamic compressors**U**

Unloaders:

- finger-type unloaders (valve depressors), 242, 242f, 243f, 244
- plug-type unloaders, 242–243, 244f, 245f
- port-type unloaders, 242, 243–244, 246f, 247f
- positive displacement compressor capacity control and, 241–244, 242f, 243f, 244f, 245f, 246f, 247f, 248f, 249f, 250f
- reciprocating compressor capacity control and, 241–244, 242f, 243f, 244f, 245f, 246f, 247f, 248f, 249f, 250f
- sliding rotary vane compressor inlet, 235f

Unloading:

- blow-off control, 234
- bypass control, 234
- clearance pocket unloading, 234f
- compressor pocket unloading, 234, 234f

Unloading (*Cont.*):

- inlet throttling unloading devices, 234, 235f
- methods, 241–242
- suction valve unloading, 233, 234f
- system, 233–234, 234f, 235f

Unloading devices. *See also* Unloaders

- adjustable inlet guide vanes, 235f
- hand operated butterfly valve, 235f
- inlet throttling, 234, 235f
- inlet unloader for sliding vane compressor, 235f

Used oil analysis:

- chemical and physical tests, 681–688, 683f, 684f, 685f, 688f
- crackle test for water, 683, 683f
- emission spectrographic analysis, 684–686, 685f
- infrared analysis, 686–687
- maintenance and, 681–689
- overview, 689
- particle count, 687–688, 688f
- proper lube oil sampling technique, 681
- test description and significance, 681, 682t
- test schedule for used industrial oils, 682t
- total acid number, 687
- total base number, 687
- various degrees of water contamination, 682f
- viscosity, 683–684
- visual and sensory inspections, 681–683, 682f
- water content, 683, 683f, 684f
- water test apparatus using Karl Fischer method, 683, 683f

Utility interconnection cost, 141

Utility steam generators:

- supercritical once-through type, 63–64
- supercritical water-tube drum type, 63, 64

VValves. *See also* Compressor valves

- acceptable and unacceptable motion, 279f
- with center bolt, 274–275, 274f
- circular channel ring, 277
- configurations, 274, 274f, 275f, 276f, 277f
- damped plate, 275, 275f, 277
- deck-and-a-half, 275, 276f
- discharge, 276f
- double-deck, 275, 276f
- dynamics calculation, 278f
- dynamics diagram, 248, 250f
- hand operated butterfly, 235f

Valves (*Cont.*):

- losses in P-V diagrams, 280, 280*f*
- manual fixed-clearance pocket, 248*f*
- materials, 277
- overview, 274
- plate, 274*f*, 275, 275*f*
- poppet, 277, 277*f*, 278*f*
- primary steam, 58
- solenoid, 50
- straight channel, 277
- suction, 233, 234*f*, 276*f*
- vibration analysis, 595*t*, 600
- water hammer and controlled closure of, 612
- water hammer and pumps with no valves at pump, 607, 608*f*, 609*f*, 610*f*, 611*f*
- water hammer and quick-opening, slow-closing, 613

Valve depressors. *See* Finger-type unloaders

Vane pumps:

- overview, 565–566
- sliding, 568*f*

Vapor pressure:

- overview, 185
- partial, 186–187

VDI. *See* Verein Deutscher Ingenieure

Velocity transducers:

- LVT (contracting), 696*f*
- overview, 695
- seismic, 696*f*

Verein Deutscher Ingenieure (VDI; Society of German Engineers), Specification 2045, 326

VI. *See* Viscosity index

Vibration:

- application of sine waves to, 691, 692*f*, 693*f*, 694*f*
- causes, 699
- machine or multimass systems, 693, 694*f*
- overview, 691
- of piston driven by crankshaft, 691, 692*f*
- of spring mass system, 692*f*
- variation of acceleration with time, 691, 694*f*
- variation of velocity with time, 691, 693*f*

Vibration analysis:

- filtering use, 695
- forcing frequency causes, 699–702
- frequency domain, 698, 699*f*, 700*f*
- logarithms and decibels, 695
- machinery example, 699, 700*f*
- machinery understanding, 699

Vibration analysis (*Cont.*):

- resonance, 693
- time domain, 697–698, 698*f*
- transducer selection, 697–698, 697*f*, 698*f*, 699*f*
- vibration diagnostics chart, 704*t*–705*t*
- vibration instrumentation, 695, 696*f*
- vibration severity chart, 702, 702*f*

Vibration analysis case history (condensate pump misalignment):

- conclusion, 703
- corrective actions, 703
- final results, 703
- problem, 703
- test data and observations, 703
- vibration diagnostics chart, 704*t*–705*t*

Vibration analysis symptoms:

- amplitude response to speed variation during vibration-test runs, 593*t*–594*t*, 597*t*–598*t*
- axial rotor rub, 590, 591*t*
- bearing and support excited vibration, 590, 591*t*
- bearing cap data, velocity unfiltered, 601
- bearing cases, 592*t*, 599
- bearing damage, 590, 591*t*
- bearing liner, 592*t*, 599
- casing and support, 592*t*, 599
- casing distortion, 590, 591*t*
- casing structural resonance, 595*t*, 599–600
- clearance-induced vibrations, 596*t*, 601
- coupling critical, 595*t*, 599
- coupling inaccuracy or damage, 592*t*, 599
- critical speed, 596*t*, 600
- dry whirl, 596*t*, 601
- electrically excited vibration, 595*t*, 600
- foundation distortion, 590, 591*t*
- foundation structural resonance, 595*t*, 600
- friction-induced whirl, 596*t*, 600
- gear inaccuracy, 592*t*, 599
- harmonic resonance, 596*t*, 600
- initial unbalance, 588, 591*t*
- journal and bearing eccentricity, 590, 591*t*
- misalignment, 590, 591*t*
- oil whirl, 596*t*, 601
- overhang critical, 595*t*, 599
- overview, 588
- permanent bow or lost rotor parts (vaness), 589, 591*t*
- pipng forces, 590, 591*t*
- pressure pulsations, 595*t*, 600
- resonant vibration, 596*t*, 600

Vibration analysis symptoms (*Cont.*):

- resonant whirl, 596*t*, 601
- rotor (shrink fits), 592*t*, 599
- rotor and bearing system critical, 595*t*, 599
- seal rub, 590, 591*t*
- subharmonic resonance, 596*t*, 600
- supports structural resonance, 595*t*, 600
- temporary rotor bow, 589, 591*t*
- thrust bearing damage, 592*t*, 599
- torsional resonance, 596*t*, 601
- transient torsional, 596*t*, 601
- unequal bearing stiffness horizontal-vertical, 590, 591*t*, 599
- valve vibration, 595*t*, 600
- vibration transmission, 595*t*, 600

Vibration diagnostics:

- analysis symptoms, 588–590, 591*t*–598*t*, 599–601, 601*t*
- hydraulic unbalance, 602
- impeller unbalance, 601–602

Vibration instrumentation:

- acceleration transducers, 695, 696*f*
- displacement transducer (proximity probe), 695
- velocity transducers, 695, 696*f*

Viscosity:

- dynamic, 671, 672
- flow through pipes, 673
- greases, 674
- kinematic, 671, 672, 673*f*
- lubricant, 671–673
- Newtonian fluids, 671, 674
- non-Newtonian fluids, 671, 674–675
- oils at low temperatures, 675
- overview, 671, 672*f*
- oxidation reactions, 675
- physical reactions, 675
- pressure effect on, 674
- significance, 672–673
- SUS and, 672, 673*f*
- temperature effect, 674
- units, 671–672, 673*f*
- variation with temperature and pressure, 674
- variation with use, 675
- VI-improved oils, 674–675

Viscosity index (VI), 674

Volumetric efficiency, and compressor cylinder

- clearance, 198–201, 199*f*, 200*f*, 201*f*

Volumetric flow rate, compressor, 197–198

W

Water. *See also* Feedwater heaters

- barrier, 334
- content in used oil analysis, 683, 683*f*, 684*f*

Water cooling requirements:

- aftercoolers, 482*t*
- compressor, 481, 482*t*
- cylinder jackets, 482*t*
- intercooler, 482*t*

Water hammer:

- applications of, 607
- basic assumptions, 604
- effects in high- and low-head pumping systems, 605
- example, 614–617, 615*f*, 616*f*
- magnitude of pulse, 605–606
- mitigating measures to, 606
- normal pump shutdown, 614
- overview, 603
- pipeline profile, 615*f*
- possible causes of, 606
- as pressure surge, 603
- pressure wave velocity in steel pipes, 616*f*
- SI conversion factors, 604
- steam hammer and, 617
- variables nomenclature, 603–604

Water hammer, and pump motor failure:

- air chambers, 613–614
- controlled valve closure, 612
- nonreverse ratchets, 614
- one-way surge tanks, 613
- pumps with check valves, 610–612
- pumps with no valves at pump, 607, 608*f*, 609*f*, 610*f*, 611*f*
- quick-opening, slow-closing valves, 613
- surge suppressors, 612
- surge tanks, 613, 614
- water column separation, 612

Water hammer, and pumps with no valves at pump:

- downsurge at midlength, 608*f*
- downsurge at pump, 608*f*
- maximum reverse speed, 610*f*
- overview, 607
- radial-flow pump case results, 607, 608*f*, 609*f*, 610*f*, 611*f*
- time of flow reversal at pump, 610*f*
- time of maximum reverse pump speed, 611*f*
- time of zero pump speed, 611*f*
- upsurge at midlength, 609*f*
- upsurge at pump, 609*f*

Water injection:

- centrifugal compressor, 389–391
- steam and, 123–124, 124*f*

Water-tube boilers:

- bent-tube boiler, 68–69, 68*f*
- boiler walls, 70–71
- cross drum, 66, 67*f*, 68
- longitudinal drum, 66, 67*f*, 68
- modern, 69, 70*f*
- mud drum, 68
- overview, 66
- radiant boiler, 72
- recent developments, 69–72
- straight-tube boiler, 66, 67*f*, 68
- water tubes, 70–71, 71*f*

Water tubes:

- embedded in refractory, 70, 71*f*
- membrane, 71, 71*f*
- studded tubes, 71*f*
- tangent bare tubes, 70, 71*f*

Watermist, 6–7

WB. *See* Wet-bulb

Wet-bulb (WB), 122

WFA. *See* With freight allowed

Wind power turbine generators:

- Brushless Double-Fed Machine,
167–174
- double-feed generators, 167–174

With freight allowed (WFA), 627

Working capital, 141